W_r = humidity ratio of average room air, pounds of water per pound dry air.
G_z = moisture content of no-load zone air, grains per pound dry air.
W_z = humidity ratio of no-load zone air, pounds of water per pound dry air.
G_c = moisture content of cold supply air, grains per pound dry air.
G_o = moisture content of outside air, grains per pound dry air.
G_r = room latent load, grains per pound of dry air supplied.
G_w = moisture content of warm air, grains per pound dry air.
X_o = ratio of outdoor air to total air.
X_w = ratio of warm to total air $= (1 - R_L)(t_r - t_c)/(t_w - t_c)$
R_L = percentage of total internal sensible heat load divided by 100.
t_r = dry-bulb temperature of room air, Fahrenheit.
t_c = dry-bulb temperature of cold air, Fahrenheit.
t_w = dry-bulb temperature of warm air, Fahrenheit.

If the state of air for average room and no-load zone is determined for various loading conditions these could be plotted in manner shown on Fig. 7, giving performance of a particular dual-duct cycle through the entire range of operating conditions, from 100 percent to zero load. The points needed to represent the performance of the cycle can be calculated either from Equations 1 and 2 or can be determined by the trial and error method described for Fig. 6.* On Fig. 7, points 1 and 2 are plotted directly from Figs. 5 and 6. It should be noted that the temperature of warm air in Figs. 5 and 6 was kept constant at 86 F. If additional points for a system operating with 86 F warm air are determined for lower loading ranges curve "B" will be obtained. The corresponding points for no-load zones could be then calculated and shown by curve 2-2-X-Z. If we arbitrarily assumed different temperatures for the warm air an additional set of curves such as A, C, D and E could be obtained as shown on Fig. 7. These curves show at a glance the effect of the warm air temperatures on the resulting average room conditions. The effect is relatively small while the system is operating at high loads and is very pronounced when the internal sensible load on a system falls to a lower level. This fact gives a clue to how this system can be controlled to prevent rise of humidity in the conditioned spaces. Various methods of control are possible depending on the medium of heat available and on the desired degree of precision of control of relative humidities. As an extreme case, which would result in least variation in the state of the room air, the system could be operated from, say, 100 to 70 percent load at any warm air temperature, say 86 F, allowing a slight rise in room relative humidity, as shown by curve B. At 70 percent load, heat could be applied to the warm air coil raising its temperature to, say, 110 F. This shifts operation from curve B to curve E and keeps variations of humidities in the room air to a minimum. This method might be feasible and economical when an inexpensive source of reheat is available, such as the hot gas of the refrigerating machine.

On large comfort installations, the following method is suggested. Allow the system to operate without external reheat on a curve such as B up to a point such as X. At that point, apply reheat gradually to the warm air heating coil. This can be accomplished by installing a humidistat in the return duct to control steam or water flow to the heating coil and, in this manner, limit the rise in average room conditions as indicated by line X-Y. While the system as a whole will operate on line 1-1-X-Y, the conditions in the no-load zone could be calculated and shown by curve 2-2-X-V.

The majority of dual-duct systems for comfort application now operating are based on cycle of Fig. 1. Experience has shown that these systems give good results when designed and operated under the following set of conditions:

1. If located in moderately humid climates, where outdoor design conditions do not exceed say 78 F wet-bulb and 95 F dry-bulb.
2. When minimum outdoor air in the system does not exceed 35–40 percent of total.
3. When heat is available and applied properly for partial summer load operation.
4. When summer cold duct temperature does not exceed 55 F dry-bulb.

Cycle of Fig. 2

The arrangement of apparatus shown in Fig. 2 overcomes the objection of bypassing excessively humid outdoor air to the warm duct on partial cooling loads. The minimum outdoor air is normally precooled to a point which will not necessitate the application of reheat in the warm duct on high summer loads. Precooling with well water or with water leaving the main cooling coils is suitable for application to this cycle. Otherwise, the performance of this cycle is similar to that of the cycle illustrated in Fig. 1, but with a reduced rate of humidity build up on partial loads. The application of reheat in the heating coil still would be required at low summer partial loads.

This cycle may be justified on comfort installations under following set of conditions:

1. When installed in relatively humid climates.
2. When the minimum outdoor air exceeds 40 percent of total air handled by the system.
3. When humidity variations in the conditioned spaces must be restricted to a narrow range.

Cycle of Fig. 3

The two-fan system shown in Fig. 3 is economical to operate and will permit close control of room humidities in summer when less than half of the total air is handled by the warm duct. Each of the two supply fans shown are sized to handle approximately half of the total system air delivery. The bypass connections shown on the up-stream and the down-stream side of the supply fans do not require any flow control. The air flow through the bypass will always be determined by the thermal requirements of the conditioned spaces and by the temperatures maintained in the cold and warm ducts. While operating at high loads the system will be equivalent to a single duct system with face and bypass dampers at the cooling coils where only return air is bypassed around the cooling coils. On a further drop in internal sensible heat load, which will increase the flow of air in the warm duct over approximately half of the total system capacity, the performance of this cycle will resemble more and more that of the cycle of Fig. 1. The application of reheat to the warm duct coil still will be necessary to depress the relative humidities in the conditioned spaces during partial load operation coincident with high dewpoint temperatures of outdoor air.

The performance of this cycle could be best illustrated by plotting performance curves for the entire range of operation of the system, from 100 percent load to zero. Such curves as calculated from Equations 1 and 2 are shown on Fig. 8. The curves A, B, C, D and E show the resulting average room conditions which would exist in the system at different assumed warm duct temperatures for specified conditions of design. Note that the rate of build-up of humidity is relatively low while internal sensible heat load on the system decreases from 100 to approximately 50 percent. If it is assumed that at high loads 80 F will be maintained in the warm duct the corresponding conditions in no-load zones could be calculated from Equation 2 and represented by dotted curve 1-N. In the 100 to 50 percent range the no-load curve and the average room conditions as represented by the performance curve A will actually coincide. Beyond the 50 percent point, on further drop in internal sensible heat load, the no-load curve will show a slightly higher rate of build-up of humidities compared to the average room conditions. If it is assumed that at a point such as M, reheat is applied to the warm duct coil, gradually increasing warm duct temperatures, a further rise in average room humidities will stop and the room humidities will be maintained at a level shown by line M-S. The corresponding conditions which would exist in no-load zone could then be calculated and be represented by curve N-O.

In applying the cycle of Fig. 3, it should be kept in mind that some stratification of air in cold and warm chambers may be expected under some conditions of flow. For this reason care should be exercised in arranging the duct connections to cold and warm chambers to avoid unequal duct temperatures. It will also be a good precaution to use only averaging types of control instruments in cold and warm chambers. Another limitation of this cycle is the possibility of not meeting ventilating requirements in lightly loaded rooms or zones which would be fed predominantly by the warm duct.

An alternate arrangement of the cycle of Fig. 3, will be obtained if the bypass connection on the discharge side of the supply fans is omitted and if both supply fans are sized to handle approximately 100 percent of total system air. This arrangement will eliminate the possibility of excessive air stratification in cold and warm chambers. The disadvantages of the alternate arrangement will be:

a. Increased supply fan power requirements.
b. Increased complexity of controls due to necessity of controlling static pressures in the duct system.
c. Increased cost of apparatus rooms.

Cycle of Fig. 4

Fig. 4 shows an arrangement that will permit very close control of relative humidities in the conditioned spaces through the entire range of operation. However, the operating cost of the cycle, because of continuous demand for reheat, is relatively high. In addition, additional refrigeration must be provided in most installations to treat the air which will be bypassed to the warm air duct under conditions of maximum loading. The use of this cycle is not economically favorable for the majority of comfort installations and is restricted to special applications.

SYSTEM DESIGN

The basic calculations required in design of dual-duct systems are the same as those for any other system and are outlined in Chapters 25 and 26 of the 1961 GUIDE AND DATA BOOK. However, when calculating the air quantities to be handled by a dual-duct system the effect of the second stream of air should be evaluated. Since in the great majority of cases the temperature of the warm air during summer operation, and of cold air during winter operation, are different from the average room temperatures maintained in the conditioned spaces, the appropriate corrections should be made in basic air quantities due to *bypassed* air in the second duct which occurs at peak loads and is required to maintain constant circulation in all conditioned spaces.

All air quantities required for design of a dual-duct system of constant volume type can be determined from Equations 3 to 9.

$$Q_{c1} = H_{zs1}/1.08(t_r - t_c) \quad (3)$$

$$Q_{w1} = H_{zw1}/1.08(t_w - t_r) \quad (4)$$

$$Q_{v1} = Q_{o1}/X_o \quad (5)$$

$$Q = Q_{z1} + Q_{z2} + \cdots + Q_{zn} \quad (6)$$

Note: Q_{z1} is Q_{c1}, Q_{w1}, or Q_{v1}, whichever is the greatest.

$$Q = Q_c + Q_w \quad (7)$$

$$H_{ss} + 1.08\,Q_w(t_w - t_r) = 1.08\,Q_c(t_r - t_c) \quad (8)$$

$$H_{sw} + 1.08\,Q_c(t_r - t_c) = 1.08\,Q_w(t_w - t_r) \quad (9)$$

where

H_{ss} = total internal sensible heat load on a system during summer peak, Btu per hour.
H_{zs1} = internal sensible heat load of a particular zone (or room) during summer peak, Btu per hour.
H_{sw} = total internal sensible heat load on a system during winter peak, Btu per hour.
H_{zw1} = internal sensible heat load of a particular zone (or room) during winter peak, Btu per hour.
Q = total air handled by the supply fan, cubic feet per minute.
Q_c = total cold air required by the system during (summer or winter) peak, cubic feet per minute.
Q_w = total warm air required by the system during (summer or winter) peak operation, cubic feet per minute.
Q_{c1}, Q_{c2}, $\cdots$, Q_{cn} = cold air required in a particular zone at summer peak, when the zone receives only cold air, cubic feet per minute.
Q_{w1}, Q_{w2}, $\cdots$, Q_{wn} = warm air required in a particular zone at winter peak, when the zone receives only warm air, cubic feet per minute.
Q_{o1}, Q_{o2}, $\cdots$, Q_{on} = outdoor (ventilation) air required in a particular zone, cubic feet per minute.
Q_{v1}, Q_{v2}, $\cdots$, Q_{vn} = total air required by a particular zone to satisfy ventilation requirements, cubic feet per minute.
t_r = dry-bulb temperature of room air, Fahrenheit.
t_c = dry-bulb temperature of cold air, Fahrenheit.
t_w = dry-bulb temperature of warm air, Fahrenheit.

Since in all zones or rooms served by a constant volume dual-duct system, the air quantities will remain substantially constant during normal operation, the air quantities for each zone (or room) must be calculated from load consideration of the two peaks or ventilating requirements and the largest of the three used to establish the air quantity for that particular space. Equations 3, 4 and 5 can be used to determine this air quantity.

After the quantity of air is established for all individual spaces the total air to be handled by the supply fan will be determined by a simple summation of the air quantities assigned to each space. This is stated by Equation 6.

The solution of Equations 7 and 8 will establish the air quantities required in cold and warm ducts for summer peak

* Additional illustrations of application of Equations 1 and 2 to dual-duct cycles can be found in paper, Elements of dual duct design and performance by N. S. Shataloff (ASHRAE TRANSACTIONS, Vol. 62, 1956, p. 257).