BUILDING ENERGY MANAGEMENT AND REDESIGN RETROFIT (BEMARR) CONVERTING HIGH-VELOCITY, CONSTANT-VOLUME, DUAL-DUCT SYSTEM INTO LOW-VELOCITY, VARIABLE AIR VOLUME SYSTEM

CONTENTS — PAGE

1. GENERAL

1.01 This section provides information and guidelines for the conversion of a high-velocity, constant-volume, dual-duct, air conditioning system into a low-velocity, variable air volume system. System conversions of this type can significantly impact energy savings. The material used in this section has been extracted from the *Building Energy Management and Redesign Retrofit (BEMARR) Manual,* issued with GL 76-10-77 (EL-4857) dated October 7, 1976.

1.02 Whenever this section is reissued, the reason(s) for reissue will be listed in this paragraph.

1.03 One of the largest energy-consuming systems is a high-velocity, dual-duct, air conditioning system. It combines the least desirable features of all air conditioning and heating systems; for example:

- High-velocity air movement creates large static pressure losses which require large motors.
- The dual-duct system is basically a reheat system. Large quantities of energy are wasted in the reheat process.

1.04 A few simple changes can transform this system into a low-velocity, single-duct, cooling-only, variable air volume system. The annual savings in electrical energy consumption can range from 15 to 90 percent.

2. METHODOLOGY

2.01 The first requirement is to determine the energy profiles for each area or floor the system serves. Once actual needs are determined, the system can be evaluated.

2.02 For this example, it is assumed that the present dual-duct system uses a single-fan unit with the following characteristics:

Cubic feet per minute (CFM) = 26,000

Static pressure (SP) = 7 inches

Brake horsepower (bhp) = 42.5

2.03 The energy analysis indicates 15,600 CFM of circulated air is required to cool the spaces served by the system.

2.04 By applying the basic fan laws for this system, the pressure that will result from reducing the CFM quantity can be determined.

$$\text{Press}_1 = \text{Press}_2 \times \left(\frac{D_2}{D_1}\right)^4 \times \left(\frac{Q_1}{Q_2}\right)^2 \times \left(\frac{P_1}{P_2}\right)$$

where:

$\text{Press}_{1\,\&\,2}$ = system static or velocity press, in inches H_2O

$D_{1\,\&\,2}$ = fan wheel size, in inches

$Q_{1\,\&\,2}$ = air quantity, in CFM

$P_{1\,\&\,2}$ = air density, lb/ft^3

In this case:

$Press_1$ = unknown Q_1 = 15,600 CFM

$Press_2$ = 7 in. H_2O Q_2 = 26,000 CFM

$D_1 = D_2 = 33$ in.

and assume

$$P_1 = P_2 = 0.75 \text{ lb/ft}^3$$

Therefore, the equation becomes:

$$Press_1 = Press_2 \times \left(\frac{Q_1}{Q_2}\right)^2$$

$$Press_1 = 7 \text{ in.} \times \left(\frac{15{,}600}{26{,}000}\right)^2$$

$$= 7 \text{ in.} \times (0.6)^2$$

$$= 7 \text{ in.} \times 0.36$$

$$= 2.52 \text{ in.}$$

2.05 Using the existing system and reducing the circulated air quantity to 15,600 CFM will reduce system static pressure to 2.52 inches H_2O.

2.06 With this information, the fan manufacturer's ratings can be used to determine whether the fan can reliably operate at these new conditions. These stated conditions are within the operating range for this particular fan.

2.07 The required bhp at 15,600 CFM and 2.52 inches SP, based on the curve of the fan, is 9.7. Thus, the saving in bhp is:

$$42.5 - 9.7 = 32.8 \text{ bhp}$$

or

$$\frac{\text{bhp savings}}{\text{bhp installed}} \times 100\% = \%\text{ bhp savings}$$

$$\frac{32.8}{42.5} \times 100\% = 77\%$$

2.08 If the fan runs the same number of hours with this reduced load, there will be a 77 percent savings in the fan energy consumed. Because of the nature of variable air volume (VAV) (namely, the matching of the load requirements), the actual savings will be greater than 77 percent. (Refer to Section 760-550-209*.)

2.09 Given the above results and the fact that simultaneous and continuous hot and cold decks waste energy, the hot deck can be removed and/or abandoned (blanked off). The hot duct work can also be used as cold air supply duct work with some simple duct changes.

2.10 The mixing boxes need not be changed if they can modulate the air without any problem.

2.11 The fan should be fitted with inlet vanes to modulate the air quantity and maintain a minimum SP. The manufacturer must be consulted to find out how to do this as well as what the minimum fan CFM is at the static pressure required in the system. In this example, at 2.52 inches SP, the minimum CFM is 8500. Certain boxes within the system must be set to have a minimum flow through them so their additive effect is 8500 CFM, or a bypass duct must be installed between the discharge of the fan and suction sides to handle 8500 CFM.

2.12 Heating devices should then be placed only in areas that actually require heating to maintain 65°F during design conditions. A load-profile graph will identify these areas and show the heat that must be added. (See Section 760-550-210*.)

2.13 When economically possible, the heating device should be interlocked to the air terminal device so that the air terminal device is at its minimum cooling position before the heater is energized.

*Check Divisional Index 760 for availability.

3. ECONOMIC ANALYSIS

3.01 The cost of local heaters and duct modifications must be weighed against the savings in energy costs using life-cycle costing.

3.02 A detailed approach to determine bhp savings is described in Section 760-550-209*.

*Check Divisional Index 760 for availability.

4. BIBLIOGRAPHY

4.01 Information used in this section is based on the following reference:

1. *ASHRAE Equipment Handbook,* American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1975.