ENERGY
MANAGEMENT
SERIES

10

FOR INDUSTRY
COMMERCE
AND INSTITUTIONS

Heating Ventilating and Air Conditioning

Energy, Mines and
Resources Canada

Énergie, Mines et
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Canada

PREFACE

Much has been learned about the art and science of managing energy during the past decade. Today, energy management is a seriously applied discipline within the management process of most successful companies.

Initially, in the early 1970's, energy conservation programs were established to alleviate threatened shortages and Canada's dependency on off-shore oil supplies. However, dramatic price increases quickly added a new meaning to the term "energy conservation" — reduce energy costs!

Many industrial, commercial and institutional organizations met the challenge and reduced energy costs by up to 50%. Improved energy use efficiency was achieved by such steps as employee awareness programs, improved maintenance procedures, by simply eliminating waste, as well as by undertaking projects to upgrade or improve facilities and equipment.

In order to obtain additional energy savings at this juncture a greater knowledge and understanding of technical theory and its application is required in addition to energy efficiency equipment itself.

At the request of the Canadian Industry Program for Energy Conservation, the Commercial and Institutional Task Force Program and related trade associations, the Industrial Energy Division of the Department of Energy, Mines and Resources Canada, has prepared a series of energy management and technical manuals.

The purpose of these manuals is to help managers and operating personnel recognize energy management opportunities within their organizations. They provide the practitioner with mathematical equations, general information on proven techniques and technology, together with examples on how to save energy.

For further information concerning the manuals listed below or regarding material used at seminars/workshops including actual case studies, please write to:

Industrial Energy Division
Energy Conservation Branch
Department of Energy, Mines and Resources
580 Booth Street
Ottawa, Ontario
K1A 0E4

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Architectural Considerations
Process Furnaces, Dryers and Kilns.

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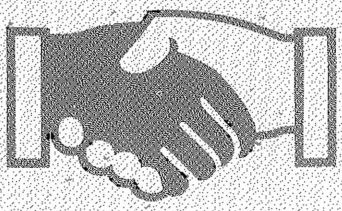


Purpose

For centuries buildings served as basic shelters to protect occupants from the extremes of the outdoor environment. Control of the indoor environment was chiefly dependent upon open fires for heat, and natural air circulation for ventilation.

The emerging technology of the twentieth century allowed the development of Heating, Ventilating and Air-Conditioning (HVAC) systems capable of maintaining fully controlled indoor environments. Based on apparently abundant low cost energy, systems were designed to meet a wide range of demands, but with little concern about energy efficiency.

Current technology has demonstrated that HVAC systems can provide safe, healthful and comfortable environments and operate at low energy consumption.

By applying the available technology to manage energy in existing buildings, dramatic cost savings can be achieved. Even in newly constructed buildings significant *Energy Management Opportunities* can be found by operating staff who are adequately informed about the building systems and their functions.

This module is intended to assist building owners and operating staff in several ways.

- Introduce the purposes and functions of heating, ventilating and air-conditioning systems.
- Describe various system types.
- Define methods of determining the approximate energy consumption.
- Provide methods of estimating potential energy cost savings.
- Provide a set of worksheets which can be used to establish both energy and cost saving potential.
- Provide a list of typical Energy Management Opportunities.

It must be understood that this is not a design manual, but a presentation of information and calculation tools which can be used to identify Energy Management Opportunities, and to estimate potential energy and cost savings.

Contents

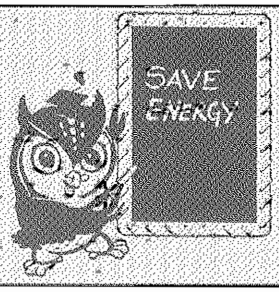
The material is subdivided into the following major headings.

The *Fundamentals* section outlines basic HVAC principles and provides simplified equations for estimating the energy requirements. Schematic diagrams illustrate the principles involved, and worked examples demonstrate application of the equations.

The *Equipment/Systems* section describes the major system components and discusses their characteristics with respect to energy consumption.

The *Energy Management Opportunities* section provides a suggested list of topics for consideration. Fundamental equations are used in worksheets to produce sample calculations of energy saving, cost saving and simple payback.

FUNDAMENTALS



Purposes of HVAC Systems

The purpose of an HVAC system is to provide the heating, cooling and ventilation requirements of a building over a range of ambient conditions specific to the building location. A system must be designed to cope with the maximum value of each of these requirements. The degree to which an HVAC system fails to match the requirements and overheats, overcools or overventilates the building space determines the amount of energy being wasted.

Particular systems may have one or more purposes.

- To maintain *comfort* by controlling temperature and humidity within acceptable limits.
- To maintain *air quality* within acceptable limits of carbon dioxide, oxygen and odor content.
- To remove *airborne contaminants* produced by processes and occupants.
- To remove *internal heat gain* produced by processes, building services and occupants.
- To provide *special environment control* for equipment and processes.

Energy Consumption by HVAC Systems

Energy consumption in HVAC systems is affected by the following.

- Building enclosure heat loss and heat gain.
- Heat loss and gain owing to infiltration of outdoor air, and exfiltration of indoor air.
- Heating and cooling of ventilating air.
- Amount of heat produced by internal sources.
- Fan energy for circulation of conditioning air.
- Pump energy for circulation of heating and cooling liquid.
- Distribution system loss.

Heat exchange in building systems is normally defined under two categories.

- *Sensible heat* which is the heat required to raise or lower the temperature of a substance such as air or water.
- *Latent heat* which is the heat required to cause a change of state of a substance, such as the conversion of water to ice (latent heat of fusion), or water to vapor (latent heat of vaporization).

Heating, ventilating and air-conditioning of a building involves sensible heat exchange to control the space temperature, and latent heat exchange to add or remove water vapor for control of humidity.

Building Enclosure Heat Loss and Heat Gain

Cold weather heat loss through the roof, walls and floor components of a building is mainly a function of temperature difference and the thermal conductivity of each component.

Heat transfer through a building enclosure component can be calculated by the equation:

$$Q = A \times U \times (T1 - T2) \times 3.6$$

Where, Q = Heat flow (kJ/h)

A = Area of enclosure component (m^2)

U = Coefficient of heat transfer of the component [$W/(m^2 \cdot °C)$]

$T1$ = Warm side temperature (°C)

T2 = cold side temperature (°C)

1 Wh = 3.6 kJ

The *coefficient of heat transfer* (U) is the heat transmission in a unit of time through a unit area of a particular body or assembly divided by the difference between the environmental temperatures on either side of the body or assembly [$W/(m^2 \cdot °C)$]. Approximate coefficients of heat transfer for typical types of building enclosure components are given in Table 1.

The determination of annual heat loss involves a totalization of the heat flows at various outdoor temperatures. In a diagnostic audit of building energy consumption by a consultant or professional this value is normally calculated using a microcomputer. This module incorporates manual approximation methods using *degree days*. The procedure is based on the assumption that, on a long-term average, solar and internal gains will offset heat loss when the mean daily outdoor temperature is 18°C, and the energy consumption will be proportional to the difference between the mean daily temperature and 18°C. Average heating degree days above and below 18°C, and below 0°C for selected Canadian locations are listed in Table 2. A complete listing is available from the Climatalogical Services Division, Atmospheric Environment Service, Environment Canada.

Degree days below 18°C for various Canadian locations are also published in The Supplement to the National Building Code of Canada (NBC). The values may vary slightly from one source to the other owing to differences in the recording period and the specific location of measurements on which the data is based.

It should be noted that a listing of Fahrenheit degree days above or below 65°F for a particular location cannot be directly compared to a listing of Celsius degree days above or below 18°C owing to the difference between the two reference temperatures. 18°C is equivalent to 64.4°F and 65°F is equivalent to 18.333°C. However, for estimating purposes, the use of the common conversion factor for Fahrenheit to Celsius degrees will yield values that compare within five per cent.

Total annual heat flow based on degree days can be estimated by the equation:

$$AH = \frac{Q \times DD \times 24}{(T1 - T2) \times 1000}$$

Where, AH = Annual heat flow (MJ)

Q = Maximum heat flow rate (kJ/h)

DD = Degree days above or below 18°C

24 = Hours per day

(T1 - T2) = Temperature difference for which Q was calculated (°C)

1000 kJ = 1 MJ

In building enclosure heating load applications, heat retention by the building structure results in the actual load being somewhat less than the theoretical heat flow rate. To account for this effect a factor is applied to the effective hours per day of the equation. For the purpose of approximating annual loads, a value of 18 hours per day can be used for most buildings.

The equation for annual building enclosure heating load then becomes:

$$AH = \frac{Qh \times DDh \times 18}{(T1 - T2) \times 1000}$$

Where, AH = Annual heating load (MJ)

Qh = Maximum heat loss rate (kJ/h)

DDh = Heating degree days below 18°C

The cost of the annual heating energy is:

$$\text{Cost} = \frac{\text{AH x Cf}}{\text{HV x Ef}}$$

Where, Cf = Unit fuel cost ($/unit)

HV = Fuel heat value (MJ/unit)

Ef = System efficiency expressed as a decimal fraction.

A detailed procedure for the determination of maximum warm weather heat gains through the enclosure of an air-conditioned building is provided in the Fundamentals Handbook of the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE). The procedure involves the determination of "effective temperature differences" and direct gain through windows caused by solar radiation. The procedure is not included in this module because of the amount of tabulated data required. The resulting values are affected by the building location, the sky clarity at that location, and the exposure angle to the sun at various times of day. Table 3 lists typical maximum summer heat gain rates for a few building components. It can be seen that maximum external heat gains occur at different times of day for various components and exposures. It can also be seen that the significant summer heat gains in a building which is insulated for winter conditions, occur through the windows and roof.

An approximation of the annual heat gain through a building enclosure can be determined from the degree day equation:

$$\text{AC} = \frac{\text{Qc x DDc x 24}}{(\text{T1 - T2}) \text{ x } 1000}$$

Where, AC = Annual heat gain (MJ)

Qc = Maximum heat gain (kJ/h)

DDc = Cooling degree days above 18°C

T1 = Outdoor design temperature (°C)

T2 = Indoor design temperature (°C)

1000 kJ = 1 MJ

The energy required by a refrigeration system to absorb the heat gain varies with the type and size of system. A value of 80 kWh/GJ can be used to approximate the energy consumption owing to a building cooling load.

For individuals wishing to have more information on procedures, a detailed description of refrigeration system performance is provided in Refrigeration and Heat Pumps, Module 11.

The equation for annual cooling cost is:

$$\text{Cost} = \frac{\text{AC x RE x Unit Energy Cost}}{1000}$$

Where, Cost = Annual refrigeration energy cost ($)

RE = Refrigeration energy consumption per unit of cooling load (kWh/GJ)
(If unknown use 80 kWh/GJ)

1000 MJ = 1 GJ

Example Building Enclosure Heat Loss Calculation

Consider an exterior corner room on an intermediate floor of a multistorey building in Windsor, Ontario.

- Room dimensions = 5 m by 4 m
- Wall height (floor-to-floor) = 4 m
- Two windows at 4 m^2 each, Ag = 8 m^2
- Gross wall area = 36 m^2
- Net wall area, Aw = 36 - 8 = 28 m^2
- Indoor design temperature, T1 = 22°C
- Outdoor design temperature, T2 = -18°C
- Degree days below 18°C, DDh = 3622 (Table 2)
- Masonry cavity wall with 100 mm insulation, Uw = 0.3 W/(m^2·°C) (Table 1)
- Double insulating glass, Ug = 3.3 (Table 1)

Wall heat loss, Qw = Aw x Uw x (T1 - T2) x 3.6 = 28 x 0.3 x [22-(-18)] x 3.6 = 1210 kJ/h

Window heat loss, Qg = Ag x Ug x (T1 - T2) x 3.6 = 8 x 3.3 x [22-(-18)] x 3.6 = 3802 kJ/h

Total room heat loss through building enclosure = 1210 + 3802 = 5012 kJ/h

$$\text{Annual heat flow, AH} = \frac{\text{Qh x DDh x 18}}{(\text{T1} - \text{T2}) \text{ x } 1000} = \frac{5012 \text{ x } 3622 \text{ x } 18}{40 \text{ x } 1000}$$

$$= 8169 \text{ MJ}$$

For a number 2 oil-fired heating system with 70 per cent efficiency and a fuel cost (Cf) of $0.40/L:

Fuel heat value, HV = 38.68 (MJ/L) (Appendix C)

System Efficiency, Ef = 0.70

$$\text{Annual heating energy cost} = \frac{\text{AH x Cf}}{\text{HV x Ef}} = \frac{8169 \text{ x } 0.40}{38.68 \text{ x } 0.70}$$

$$= \$121$$

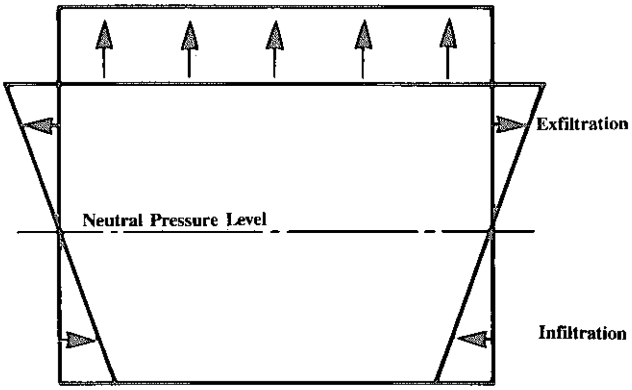
Infiltration Heat Loss and Heat Gain

Air leakage occurs in every building because of *stack effect* caused by the temperature difference between the building interior and outdoor air, and wind pressures. These forces create air flow through building leakage paths such as doorways, cracks around windows, joints between walls and roofs and joints between other building components. Outward leakage (exfiltration) of conditioned air occurs where the interior building pressure is higher than the outdoor pressure. The lost air must be replaced by simultaneous inward leakage (infiltration) of outdoor air at another location in the building.

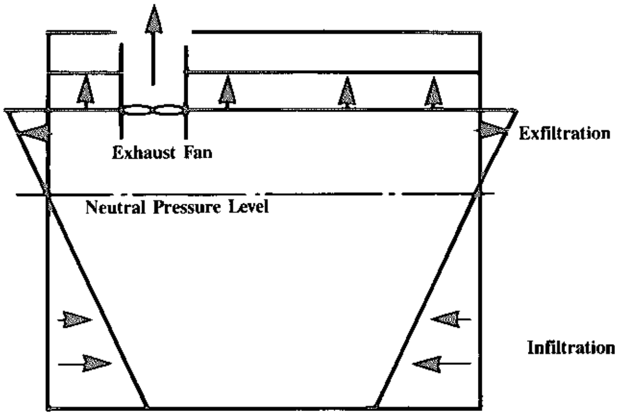
In multistoried buildings the stack effect results in infiltration at the lower storeys and exfiltration at the higher storeys. A neutral pressure zone occurs at the cross-over level. Figure 1 illustrates the pressure relationship owing to stack effect.

In some buildings ventilation and make-up air required to replenish air expelled by the exhaust systems depends on natural infiltration from building leakage and openable windows. In cold weather with windows closed, operation of the exhaust systems causes a negative pressure in the building. The negative pressure increases infiltration rates at building leakage points, which often results in cold drafts at the points of maximum leakage and contributes to deterioration of the building components owing to condensation. The condition is also evidenced by a rush of cold air into the building when doors to the outside are opened. Figure 2 illustrates the pressure relationships caused by stack effect combined with mechanical exhaust systems.

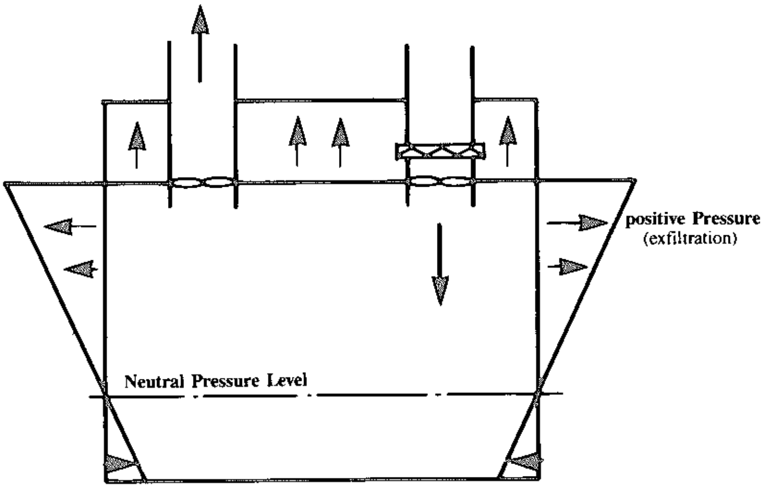
Some buildings are equipped with a make-up air system sized to deliver heated outdoor air to replace the exhaust air and maintain a positive building pressure. The positive pressure improves the comfort level by reducing the infiltration rate caused by stack effect and wind pressure (Figure 3). With no make-up air system the added load must be handled by the local heating and cooling systems. A make-up air system provides controlled heating and cooling from a central location, and provides an opportunity for heat recovery from exhaust air.



Building Stack Effect
Figure 1



Building Stack Effect With Powered Exhaust
Figure 2



Building Stack Effect With Powered Make-up
Figure 3

The Fundamentals Handbook by ASHRAE outlines several optional methods for estimating the infiltration rate. The subject is also discussed in Architectural Considerations, Module 18. An approximation may be achieved by assuming a value of air changes per hour based on experience with various building exposures and configurations. Typical values are given in Table 4.

Air flow by infiltration and exfiltration can be approximated by the equation:

$$fa = \frac{V \times CH}{3.6}$$

Where, fa = Air flow rate (L/s)

V = Room volume (m^3)

CH = Air changes per hour

$$3.6 = \frac{3600 \text{ s/h}}{1000 \text{ L/m}^3}$$

Energy calculations are based on conditioning the infiltration air to the building design temperature and humidity levels.

The *sensible* heat exchange with infiltration air may be approximated by the equation:

$$Qs = fa \times (T1 - T2) \times 4.345$$

Where, Qs = Sensible heat flow (kJ/h)

fa = Rate of air flow (L/s)

T1 = Warmer temperature (°C)

T2 = Cooler temperature (°C)

4.345 = a factor which accounts for the specific heat of dry air and conversion to common units.

The factor 4.345 would increase slightly for air containing water vapor, but this value is considered sufficiently accurate for estimating purposes.

For a heating load the equation for annual sensible heat is:

$$AHs = \frac{Qs \times DDh \times 24}{(T1 - T2) \times 1000}$$

Where, AHs = Annual sensible heat energy (MJ)

DDh = Annual heating degree days below 18°C

1000 kJ = 1 MJ

The latent heat exchange with infiltration air may be approximated by the equation:

$$QL = fa \times (H1 - H2) \times 10.84$$

Where, QL = Latent heat flow (kJ/h)

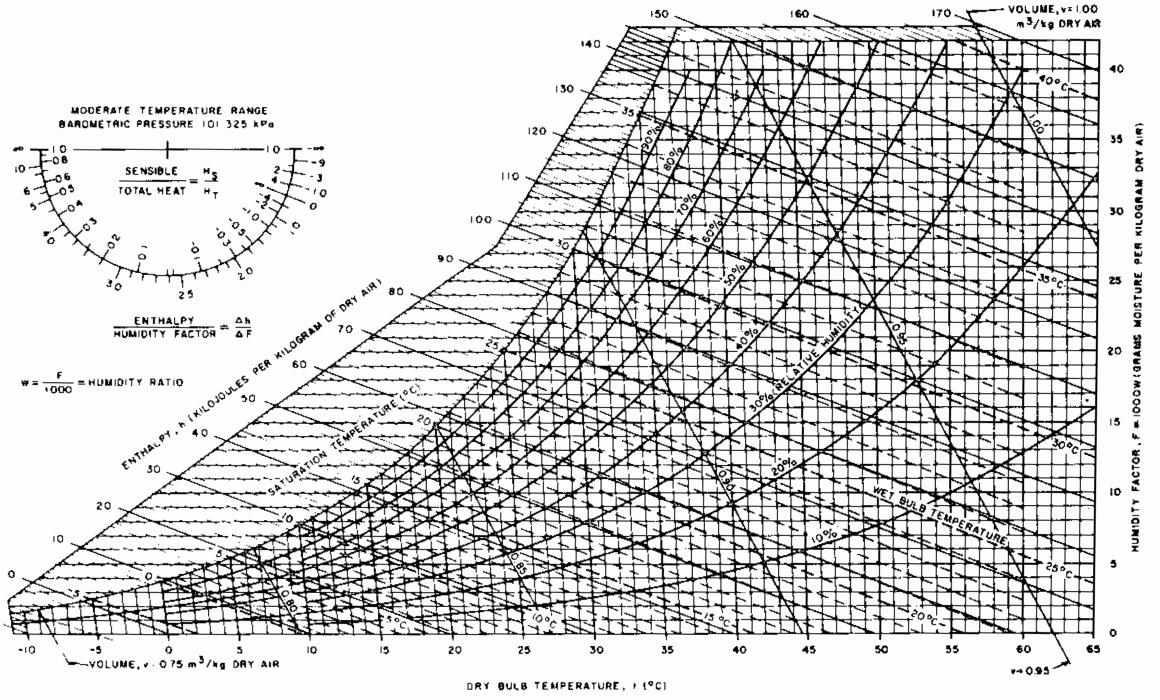
fa = Rate of air flow (L/s)

H1 = Higher "humidity factor" (g water/kg dry air)

H2 = Lower "humidity factor" (g water/kg dry air)

10.84 = Factor which accounts for the latent heat of vaporization of water and conversion to common units.

The *humidity factor* of air at various temperature and relative humidity levels is given by the psychrometric chart of Figure 4. To read the chart locate the "dry bulb temperature", proceed vertically to the intersection with the known "relative humidity" or "wet bulb temperature" line, and draw a line horizontally to the "humidity factor" value at the right hand side.



Psychrometric Chart
Figure 4

Courtesy of ASHRAE

The annual heat energy required for humidifying infiltration air varies with geographic location and the design condition. An approximate value can be determined by assuming the humidification load proportional to one-half the annual degree days.

The equation to calculate the annual heat required for humidification is:

$$AHL = \frac{QL \text{ x } (0.5 \text{ x } DDh) \text{ x } 24}{(T1 - T2) \text{ x } 1000}$$

$$= \frac{QL \text{ x } DDh \text{ x } 12}{(T1 - T2) \text{ x } 1000}$$

Where, AHL = Annual humidification heat energy (MJ)

DDh = Annual heating degree days below 18°C.

The annual cooling load owing to outdoor air conditions can also be estimated by the degree day method. The procedure is based on the same assumption as for heating; on a long-term average, the heat gain owing to outdoor conditions will be proportional to the difference between the mean daily temperature and 18°C. For outdoor air cooling loads, both the sensible and latent loads can be considered proportional to degree days.

Annual cooling heat gain based on degree days can be estimated by the equation:

$$AC = \frac{Qt \text{ x } DDc \text{ x } 24}{(T1 - T2) \text{ x } 1000}$$

Where, AC = Annual heat gain (MJ)

Qt = Total maximum cooling load (kJ/h)

DDc = Annual cooling degree days above 18°C

(T1-T2) = Temperature difference between indoor and outdoor air on which Qt was calculated (°C)

1000 kJ = 1 MJ

Example of Infiltration Heat Loss Calculation

Consider the room described in the previous Example Building Enclosure Heat Loss Calculation, with an infiltration rate of ½ air change per hour:

- Ceiling height = 3 m
- Room volume = 3 x 5 x 4 = 60 m^3
- Indoor relative humidity = 40%
- Indoor air humidity factor, H1 = 6.6 (from Figure 4)
- Outdoor relative humidity = 90%
- Outdoor air humidity factor, H2 = 1.4 (from Figure 4 by extrapolation)

$$\text{Infiltration air flow, fa} = \frac{V \text{ x } CH}{3.6} = \frac{60 \text{ x } 0.5}{3.6}$$

$$= 8.33 \text{ L/s}$$

Sensible heat loss, Qs = fa x (T1 - T2) x 4.345 = 8.33 x [22-(-18)] x 4.345 = 1448 kJ/h

Latent heat loss, QL = fa x (H1 - H2) x 10.84 = 8.33 x (6.6 - 1.4) x 10.84 = 470 kJ/h

Total infiltration heat loss = 1448 + 470 = 1918 kJ/h

Example of Infiltration Heat Gain Calculation (Cooling Load)

Consider the previously described example as a summer air-conditioning load:

- Indoor design temperature, T2 = 24°C
- Outdoor temperature, Tl = 31°C
- Indoor design relative humidity, RH = 60%
- Indoor air humidity factor, H2 = 11.1 (Figure 4)
- Outdoor wet bulb temperature = 24°C
- Outdoor air humidity factor, Hl = 15.9 (Figure 4)
- Infiltration air flow, fa = 8.33 L/s (1/2 air change/h)
- Degree days above 18°C, DDc = 391 (Table 2)

Sensible cooling load, Qs = fa x (Tl - T2) x 4.345 = 8.33 x (31 - 24) x 4.345 = 253 kJ/h

Latent cooling load, QL = fa x (Hl - H2) x 10.84 = 8.33 x (15.9 - 11.1) x 10.84 = 433 kJ/h

Total infiltration cooling load, Qt = 253 + 433 = 686 kJ/h

$$\text{Annual cooling load, AC} = \frac{\text{Qt x DDc x 24}}{\text{(Tl - T2) x 1000}} = \frac{\text{686 x 391 x 24}}{\text{7 x 1000}}$$

$$= 920 \text{ MJ}$$

Annual air-conditioning energy cost resulting from infiltration, based on a refrigeration consumption (RE) of 80 kWh/GJ at $0.05/kWh is:

$$\text{Cost} = \frac{\text{AC x RE x Unit Energy Cost}}{1000} = \frac{\text{920 x 80 x 0.05}}{1000}$$

$$= \$3.68/\text{yr}$$

Heating and Cooling of Ventilation Air

Most buildings require some circulation of outdoor air for ventilation purposes in addition to the intermittent effects of infiltration. If ventilation is achieved by opened windows or forced infiltration from the operation of exhaust systems, the effect is difficult to control and may result in local heating or cooling loads which exceed the capacity of the building HVAC system. Various ventilation systems and their characteristics are discussed later in this module.

The equations previously given for heat exchange with infiltration air can also be applied to estimate the heating and cooling loads for ventilation air.

Cooling Required by Internal Heat Sources

Internal heat sources in a building include sensible heat from lights, people and process equipment, and latent heat in the form of water vapor from people and processes. Such heat gains to the internal space are normally offset by supplying cool, dry air capable of absorbing the gains without exceeding the room design conditions. The various systems used to achieve this effect are outlined later in this module.

The instantaneous heat gain from lights and other electrically powered devices is represented by the electric power supplied, and can be calculated by the equation:

Qi = W x 3.6

Where, Qi = Instantaneous heat gain (kJ/h)

W = Power (W)

1 W = 3.6 kJ/h

Heat gain from process equipment can be estimated from the energy input rate to the process. Equipment installed under an effective exhaust hood will give off negligible latent and convection heat to the room. However, even with an effective exhaust hood, the radiant heat gain to the room from high temperature surfaces such as cooking appliances can represent over 30 per cent of the energy input.

Heat gain from people depends on the rate of activity, modes of dress and environmental conditions. Typical rates of heat gain for various conditions are provided in Table 5. The rates provided are averaged values based on normal mixes of men, women and children, and conditions normally encountered for the activities listed.

The effect of sensible heat gain on the cooling load of a space is not instantaneous because of thermal storage effects of the building structure and contents. This effect is analyzed in detail in the ASHRAE Fundamentals Handbook where factors are given for various heat sources and their duration times. For estimating purposes, a *factor of duration* of 0.8 can be applied to most sensible heat gains having a duration of less than 8 hours per day. A factor of 1.0 should be used for load durations in excess of 8 hours per day. For intermittent operation a *factor of usage* representing the average minutes of use per hour is also applied.

An estimate of the sensible cooling load caused by an internal heat gain can be determined by the equation:

$$Qs = \frac{Qi \text{ x } fd \text{ x } fu}{60}$$

Where, Qs = Sensible cooling load (kJ/h)

Qi = Instantaneous heat gain (kJ/h)

fd = Factor of duration (1.0 if greater than 8 h/day, or 0.8 if less than 8 h/day)

fu = Factor of utilization (average minutes of operation per hour)

60 = Minutes per hour

The total cooling load caused by internal heat gain is:

$$Qt = Qs + QL$$

Where, Qt = Total cooling load (kJ/h)

Qs = Sensible cooling load (kJ/h)

QL = Latent cooling load (kJ/h)

The annual cooling load caused by internal heat gain is determined by the number of days during which the *daily mean temperature* is above the *balance temperature* for the building. The balance temperature is the outside temperature at which the building heat loss equals the heat gain.

An estimate of the annual cooling load can be determined by the equation:

$$ACi = \frac{Qt \text{ x } N \text{ x } h}{1000}$$

Where, ACi = Annual internal cooling load (MJ)

Qt = Total cooling load (kJ/h)

N = Number of months during which the *monthly mean temperature* is above 18°C

h = Average hours of operation per month

1000 kJ = 1 MJ

The monthly mean temperatures from which to determine N for a particular location are available from the local weather offices of Environment Canada.

Example Calculation of Cooling Load

Consider an *interior zone* of 200 square metres in a Fredericton office building, containing 20 persons for 7½ hours per day, 5 days per week (162.5 hours per month), and office equipment with total connected power load of 3000 watts and average usage of 30 minutes per hour:

- Space contains 80 light fixtures of 88 watts per fixture
- Ventilation air supply = 6 litres per second per person
- Indoor design condition, T2 = 24°C at 60% RH)
- Degree days above 18°C, DDc = 143 (Table 2)
- Months during which monthly mean temperature is above 18°C, N = 2 (Environment Canada)
- Cooling design outdoor dry bulb temperature, T1 = 29°C (National Building Code)
- Cooling design outdoor wet bulb temperature = 21°C (National Building Code)
- Factor of duration, fd (people, lights and equipment) = 0.8
- Factors of utilization:

  fu (people and lights) = 60 minutes per hour

  fu (equipment) = 30 minutes per hour

- Sensible heat gain per person (Table 5) = 240 kJ/h
- Instantaneous sensible heat gains:

  Qi (people) = 240 x 20 = 4800 kJ/h

  Qi (lights) = W x 3.6 = (80 x 88) x 3.6 = 25 344 kJ/h

  Qi (equipment) = W x 3.6 = 3000 x 3.6 = 10 800 kJ/h

- Sensible cooling loads:

$$\text{Qs (people)} = \frac{\text{Qi x fd x fu}}{60} = \frac{4800 \text{ x } 0.8 \text{ x } 60}{60} = 3840 \text{ kJ/h}$$

$$\text{Qs (lights)} = \frac{\text{Qi x fd x fu}}{60} = \frac{25\ 344 \text{ x } 0.8 \text{ x } 60}{60} = 20\ 275 \text{ kJ/h}$$

$$\text{Qs (equipment)} = \frac{\text{Qi x fd x fu}}{60} = \frac{10\ 800 \text{ x } 0.8 \text{ x } 30}{60} = 4320 \text{ kJ/h}$$

- Latent heat gain per person (Table 5) = 200 kJ/h
- Latent cooling loads:

  QL (people) = 200 x 20 = 4000 kJ/h

- Total cooling load from internal heat gains:

  Qt (internal) = Qs (people) + QL (people) + Qs (lights) + Qs (equipment)

  = 3840 + 4000 + 20 275 + 4320

  = 32 435 kJ/h

- Ventilation air flow, fa = 20 x 6 = 120 L/s
- Indoor air humidity factor, H2 = 11.1 (Figure 4)
- Outdoor air humidity factor, H1 = 12.2 (Figure 4)

Ventilation air sensible cooling:

$$Qsv = fa \times (T1 - T2) \times 4.345 = 120 \times (29-24) \times 4.345 = 2607 \text{ kJ/h}$$

Ventilating air latent cooling:

$$QLv = fa \times (H1 - H2) \times 10.84 = 120 \times (12.2 - 11.1) \times 10.84 = 1431 \text{ kJ/h}$$

Total ventilation cooling:

$$Qv = Qsv + QLv = 2607 + 1431 = 4038 \text{ kJ/h}$$

Annual internal cooling:

$$ACi = \frac{Qt \text{ (internal)} \times N \times h}{1000} = \frac{32\ 435 \times 2 \times 162.5}{1000} = 10\ 541 \text{ MJ/yr}$$

Annual ventilation cooling:

$$ACv = \frac{Qv \times DDc \times 7.5}{(T1 - T2) \times 1000} = \frac{4038 \times 143 \times 7.5}{(29 - 24) \times 1000} = 866 \text{ MJ/yr}$$

Annual cooling gain :

$$AC = 10\ 541 + 866 = 11\ 407 \text{ MJ}$$

Annual air-conditioning energy cost, based on 80 kWh/GJ and $0.05/kWh, can be determined by the equation:

$$\text{Cost} = \frac{AC \times RE \times \text{Unit Energy Cost}}{1000} = \frac{11\ 407 \times 80 \times 0.05}{1000}$$

$$= \$46/\text{yr}$$

Fan and Pump Energy

Power to move an air or liquid stream against a given pressure can be calculated by the equation:

$$W = \frac{f \times DP}{1000 \times Ef}$$

Where, W = Pumping power (kW)

f = Flow rate (L/s)

DP = Total pressure differential across the fan or pump (kPa)

Ef = Overall efficiency of the pump and motor expressed as a decimal fraction

1000 = a factor which accounts for conversion to common units.

Annual pumping energy cost can be determined by the equation:

$$\text{Cost} = W \times t \times Ce$$

Where, Cost = Annual cost ($)

t = Annual operating time (h)

Ce = Electrical energy cost ($/kWh)

Fan and pump efficiencies are discussed in Fans and Pumps, Module 13. For approximation purposes in HVAC systems, typical efficiency factors would be 0.75 for fans and 0.50 for pumps.

The equation for approximating fan power becomes:

$$Wf = \frac{fa \times DP}{750}$$

Where, Wf = Fan power (kW)

fa = Air flow rate (L/s)

The equation for approximating pump power becomes:

$$Wp = \frac{fw \times DP}{500}$$

Where, Wp = Pump power (kW)

fw = Liquid flow rate (L/s)

Example of Pump Energy Calculation

A heating system pump operates continuously circulating 100 L/s against a total system pressure drop of 310 kPa. For ten weeks per year heating is not required and the pump can be switched off. At an electrical energy cost (Ce) of $0.05/kWh, the estimated annual pump energy cost saving is determined by:

$$\text{Pump power, } Wp = \frac{fw \times DP}{500} = \frac{100 \times 310}{500} = 62 \text{ kW}$$

$$\text{Annual cost saving} = Wp \times t \times Ce = 62 \times (10 \times 7 \times 24) \times 0.05 = \$5{,}208$$

Heating Systems

Systems commonly used to provide heating energy in a building include hot water, steam, electric, forced air, and solar.

Hot Water Heating

Hot water heating systems deliver heat energy from a boiler or heat exchanger to the terminal heating units by circulating water through a piping system. Circulation is usually maintained by an electrically driven circulating pump or pumps. The rate of circulation is designed to provide the maximum required heat transmission with approximately an 11°C temperature difference between the supply and the return at the boiler or heat exchanger. The required supply water temperature is determined by the output capacity of the terminal heating units. Most systems require 80 to 110°C supply water. Systems designed to utilize condenser heat from refrigeration systems operate with a supply water temperature as low as 38°C.

Heat transmission by water at normal heating system temperatures may be approximated by the equation:

$$Q = fw \times (T1 - T2) \times 15$$

Where, Q = Total heat load (MJ/h)

fw = Water flow (L/s)

T1 = Heating water supply temperature (°C)

T2 = Heating water return temperature (°C)

15 = A multiplier which accounts for the specific heat of water and conversion to common units.

System efficiency is affected by three factors.

- Boiler or heat exchanger efficiency.
- Heat loss from the piping system.
- Pumping energy required to maintain the water flow.

The *boiler efficiency* of a hot water boiler is a function of combustion efficiency and boiler shell heat losses. Refer to Combustion, Module 5 and Boiler Plant Systems, Module 6 for additional details.

The heat input to a boiler is determined by the quantity of fuel consumed and the heat value of the fuel (Appendix C). Operating efficiency can be over 80 per cent for a well maintained boiler at full load to less than 60 per cent for a poorly maintained boiler at part load.

The fuel consumption by a boiler can be determined by the equation:

$$\text{Fuel consumption} = \frac{Q}{HV \times Ef}$$

Where, Fuel consumption = fuel input (L/h)

Q = Total heat transmitted (MJ/h)

HV = Fuel heat value (MJ/L)

Ef = Boiler efficiency expressed as a decimal fraction.

The efficiency of a *heat exchanger* is mainly a function of shell losses since the efficiency of the heat source system is normally considered separately. Heat exchanger efficiency of 100 per cent can be used for estimating purposes.

Heat loss from the piping system is a function of the water temperature and the amount of pipe insulation. Details on heat loss from pipes at various temperatures and insulation levels is provided in Process Insulation, Module 1. However, average piping heat loss of 3 per cent of the peak heat delivery rate can be used for estimating purposes.

The following is an example calculation of the required heat input to a hot water heating system.

Water flow rate	6.5 L/s
Return water temperature	80°C
Supply water temperature	90°C

Boiler: oil-fired, #2 fuel, 75% boiler efficiency

Fuel heat value = 38.68 MJ/L (Appendix C)

Heat flow, $Q = fw \times (T1 - T2) \times 15 = 6.5 \times (90 - 80) \times 15 = 975$ MJ/h

$$\text{Fuel consumption} = \frac{Q}{HV \times Ef} = \frac{975}{38.68 \times 0.75} = 33.6 \text{ L/h}$$

Pumping Energy can be estimated by the equation previously given under "Energy Consumption in HVAC Systems".

System efficiency is determined by the equation:

$$Ef = \frac{Qh}{Qb + [(Wp + Wb) \times 3.6]}$$

Where, Ef = Overall system efficiency expressed as a decimal fraction

Qh = Heat output by the terminal heating units (MJ/h)

Qb = Heat input to the boiler or heat exchanger (MJ/h)

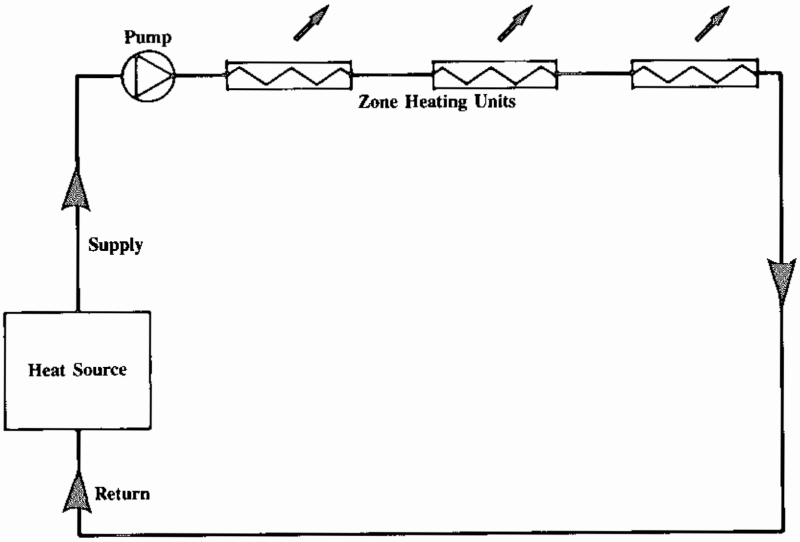
Wp = Power input to the circulating pump (kW)

Wb = Power input to the boiler components (kW)

1kW = 3.6 MJ/h

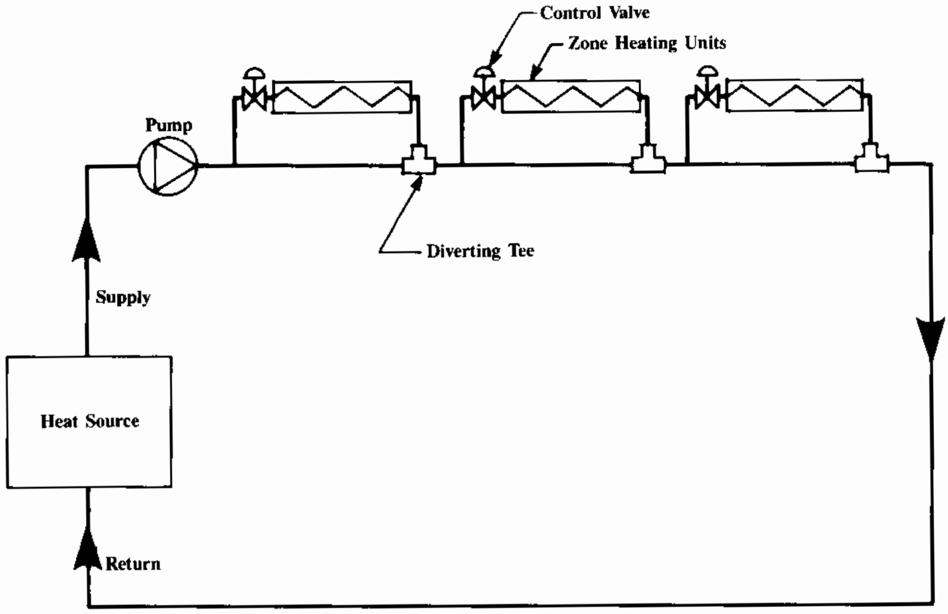
The following are typical hot water heating system piping arrangements.

The *series loop* arrangement (Figure 5) is a low cost method of connecting a series of heating units to the heat source, but provides no opportunity for individual control of the heat output from each heating unit. The heat output of small systems can be controlled by starting and stopping the pump in response to a building thermostat. The heat output of larger systems is usually controlled by regulating the supply water temperature in relation to outdoor air temperature or a selected building space temperature. If the individual heating unit outputs are not matched precisely with the requirements of each space, fluctuations of local space temperatures and possible conflict with air-conditioning systems serving the same space could occur. Successive heating units in the loop receive a lower supply water temperature because of the heat removed by preceding units.



Series Loop Piping
Figure 5

The *single pipe loop with diverting tees* system (Figure 6) is an adaptation of the series loop system, and provides the opportunity for control of the heat output from each heating unit. Systems may have control valves to regulate individual heating unit flows, and more than one heating unit may be connected to one diverting tee branch. Like the series loop system, each succeeding heating unit receives a lower supply water temperature than the preceding units.

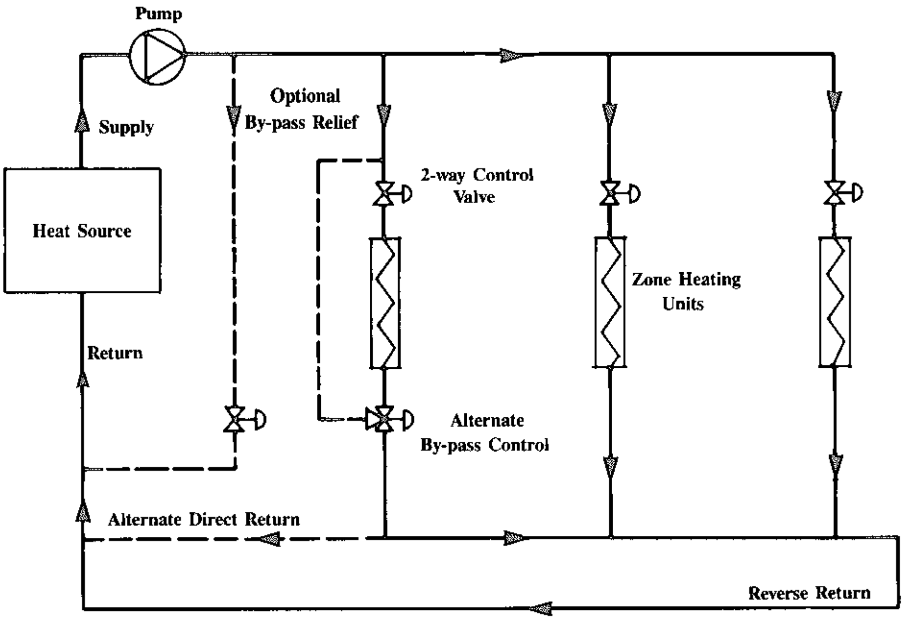


Single Pipe Loop With Diverting Tees
Figure 6

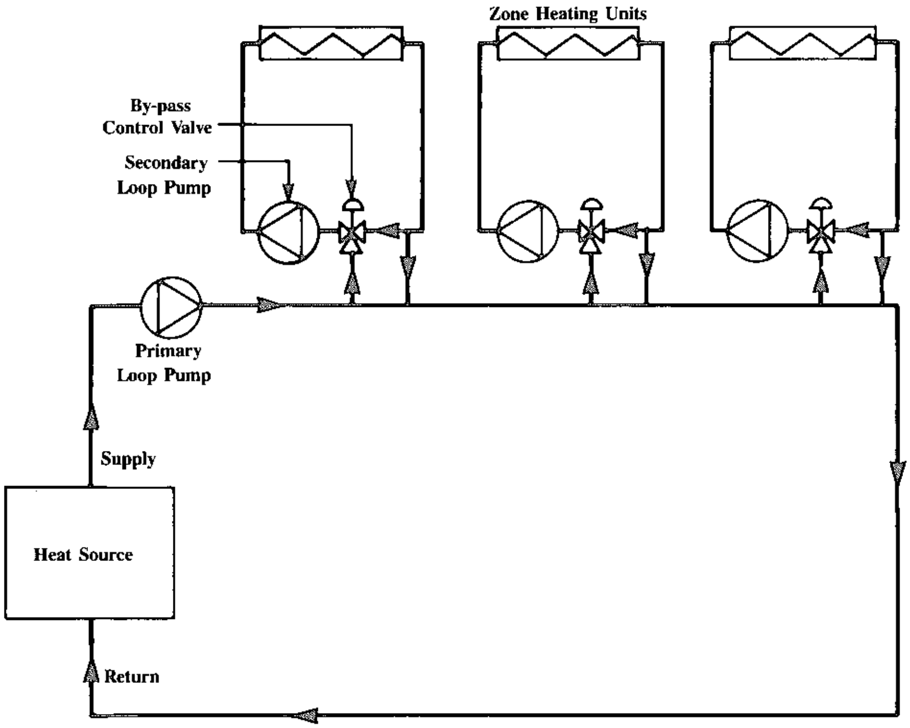
The *two pipe parallel connection* system (Figure 7) is the most commonly used hot water heating arrangement, and provides the opportunity for individual control of heating units. Where two-way control valves are used at the heating units, some method of regulating the circulation flow rate is usually employed. The circulating pump speed may be regulated, or a pressure relief by-pass may be used to permit constant pump operation. An alternate three-way by-pass control at each heating unit allows a constant water circulation rate and fast heating response at the heating units. The *reverse return* arrangement provides equal-length piping to all heating units for ease of system balancing. Small systems often use the alternate *direct return* arrangement for economy of piping.

In a *primary-secondary pumped system* (Figure 8) the primary loop flow is unaffected by variations in the secondary loop flow. This system allows stable control of secondary loop water temperature below that of the primary loop. For widely distributed systems it allows the use of smaller piping with higher than normal temperature drop in the primary loop. It also allows individual secondary loops to be shut off when not in use. This system may be found in combination with any of the previous piping systems and may be used to add new secondary loops to an existing system.

Information on other hot water heating systems for specialized industrial and institutional applications, utilizing high pressure, high temperature water or high temperature organic fluids, is available in the ASHRAE Systems Handbook and Module 8. Steam and condensate systems.



Two Pipe Parallel Connection
Figure 7



Primary-Secondary Pumped System
Figure 8

Steam Heating

A steam heating system uses the vapor phase of water to transport heat from a boiler to the terminal heating devices. Steam is propelled through the supply piping system by the pressure generated at the boiler. The steam condenses in terminal heating devices, giving up its latent heat of vaporization, and the condensate is returned to the boiler. The condensate may return to the vicinity of the boiler by gravity flow, but it must be pumped into the boiler to complete the circuit. The characteristics and details are described in Steam and Condensate Systems, Module 8.

Steam systems have advantages and disadvantages compared to water systems.

Advantages include:

- A higher heat transfer rate per unit of surface area is produced at the terminal device, allowing the use of smaller devices.
- Smaller piping can be used for a given rate of energy transmission.
- Less pumping energy is required than for a hot water system.
- The system flow is self-regulating, being dependent upon the condensation rate (heat output) at the terminal devices and condensation (heat loss) in the supply piping.
- Steam can be used directly for humidification of building air.

Disadvantages include:

- Condensate may be released from the terminal heating devices at a temperature higher than the *saturation temperature* at the return system pressure. This results in the generation of *flash steam* which is often discharged outdoors from vented condensate receivers, resulting in a significant energy loss.
- Because the saturation temperature of steam at atmospheric pressure is 100°C, a partial vacuum is necessary to operate at a lower steam temperature.
- System maintenance cost is high owing to the number of distribution system components such as steam traps and condensate pumps.

For normal HVAC system steam pressures up to 200 kPa(gauge), the heat flow to a steam heated device can be approximated by the equation:

$$Q = fs \times 2.2$$

Where, Q = Heat flow (MJ/h)

fs = Steam flow (kg/h)

2.2 MJ = 1 kg of steam

System efficiency is affected by several factors.

- Boiler efficiency.
- Heat loss from the steam and condensate piping.
- Flash steam loss in the return system.
- Condensate loss.
- Steam or condensate leaks.
- Condensate and feedwater pump energy.

Boiler efficiency is similar for both steam and hot water boilers. However, a typical steam boiler plant has other losses which affect the overall plant efficiency: condensate receiving tanks and condensate treatment equipment have surface heat losses, and chemical treatment of boiler feedwater requires *boiler blowdown*. These losses may reduce the overall plant efficiency by 1 to 5 per cent below that of an equivalent hot water plant. A typical annual efficiency range for steam boiler plants is 60 to 80 per cent. Careful attention to minimizing heat losses from boiler plant equipment can result in a steam boiler plant achieving the same efficiency as an equivalent hot water boiler plant.

Heat loss from steam and condensate piping is a function of the amount of pipe insulation, the length of the piping system, the pressure of the steam and the temperature of the return condensate. A detailed description and analysis of pipe insulation is given in Process Insulation, Module 1. For estimating purposes in this module, pipe heat loss can be taken as 3 per cent of the peak heat output of the system.

Flash steam loss in the return system is affected by the operating temperature of the terminal heating units, and the effective operation of the steam traps to prevent bypassing of steam to the condensate return system. Steam and Condensate Systems, Module 8, identifies the flash loss from condensate at various temperatures when released to atmospheric pressure.

Condensate loss results from certain system conditions and methods of using steam.

- The direct use of steam for humidification.
- Heating of water by direct injection of steam.
- Overflow at undersized condensate receivers.
- Steam and condensate system leaks.

The loss of steam and condensate from direct steam use, leaks and overflow requires the heating of an equivalent amount of cold water make-up at the boiler plant. An increased make-up rate requires increased chemical treatment and a higher blow-down rate at the boiler.

Steam leaks represent a total loss of both condensate and heat energy of the steam. A relatively small opening can result in very significant annual losses because it leaks continuously. Steam and Condensate Systems, Module 8, lists typical steam leakage rates for a range of opening sizes.

Condensate and feedwater pumping energy can be estimated using the general pump energy equation previously given for hot water heating pumps.

The characteristics of steam terminal devices are discussed in more detail in Heating and Cooling Equipment (Steam and Water), Module 9.

System efficiency is given by the equation:

$$Ef = \frac{Qh}{Qb + [(Wp + Wb) \times 3.6]}$$

Where, Ef = System efficiency expressed as a decimal fraction

Qh = Heat output by the terminal heating device (MJ/h)

Qb = Heat input to the boiler (MJ/h)

Wp = Power input to condensate and feedwater pumps (kW)

Wb = Power input to the boiler (kW)

1 Kw = 3.6 MJ/hr

Electric Heating

Electric power allows a simple and efficient system of energy distribution within a building. It is used in areas where electrical energy costs are competitive with the cost of fossil fuel energy. Electricity can be used as the energy source for steam or hot water boilers, but the distribution losses of such systems reduce the efficiency advantage. Heat is most commonly generated at the terminal device by resistance elements which operate at 100 per cent efficiency. Control is achieved by switching off the entire element, by staged switching of multiple elements, or by high speed solid state switching of the alternating current in such a way as to modulate the power input to the element.

The characteristics and details of electric power systems are addressed in Electrical, Module 3.

Terminal heating devices utilize several types of electric heating elements.

- Direct resistance element in contact with the air or fluid.
- Embedded resistance element to produce low temperature *radiant heat.*
- Exposed infrared-producing element to produce high temperature radiant heat.
- Submerged electrodes using water as the resistance medium.

The heat output by an electric heating element can be determined by the equation:

$$Q = Wi \times 3.6$$

Where, Q = Heat Output (MJ/h)

Wi = Power input (kW)

1kW = 3.6 MJ/h

Overall system efficiency is affected only by transformer energy loss and small resistance losses in the distribution wiring.

Forced Air Heating

Forced air heating systems use air as the transport medium between the heat source and the space to be heated. The heat source may be direct fired with fossil fuel or other combustible materials, electrically heated, supplied with heat rejected by processes such as refrigeration systems, or a combination of two or more of these sources. Forced air heating systems are found in single zone applications such as small commercial facilities, supermarkets, and repair garages. In these applications the air system is often the only source of space heating.

The air circulation system provides an opportunity to perform several other functions in addition to heating or cooling.

- Filtering of the air stream for removal of air borne dust and other contaminants.
- Humidification.
- Controlled injection of outdoor air for ventilation.
- Air redistribution.

The total heat energy required for a forced air heating system includes the sensible heat required to raise the air temperature, and the latent heat required to provide humidification.

The efficiency of a forced air heating system is affected by several factors.

- Efficiency of the forced air heat source unit.
- Heat loss and air leakage from ducts located outside the heated space.
- Fan energy required to maintain the air flow.
- The effectiveness of the air distribution in the heated space.

The system efficiency can be determined by the equation:

$$Ef = \frac{Qr + Qv}{Qf + (Wf \times 3.6)}$$

Where, Ef = System efficiency expressed as a decimal fraction

Qr = Heat input required at the occupied level of the space being heated (MJ/h)

Qv = Heat required to heat the ventilating air to room temperature (MJ/h)

Qf = Heat input to the warm air source (MJ/h)

Wf = Power input to the circulating fan (kW)

1 kW = 3.6 MJ/h

Solar Heating

Under Canadian climatic conditions solar heating is used mainly as a supplementary heat source to reduce the consumption of fossil fuels or electricity.

Solar systems fall under two general classifications.

- *Active systems* collect solar heat into an air or liquid system and transport it to storage or to a terminal heating device.
- *Passive systems* collect solar heat directly into the space or process to be heated.

Active solar heating system components and system arrangements are described later in the Equipment/Systems section.

Active solar air-heating systems usually circulate the heating air directly through a solar collector. A thermal storage medium is often provided to store heat collected during peak sunshine hours for use during other heating periods.

Active solar water-heating systems include direct and indirect systems. Direct systems circulate the heated water through the solar collector. Indirect systems circulate a secondary fluid, such as an antifreeze solution, between the solar collector and a heat exchanger. In all cases care must be taken to ensure that circulation through the collector takes place only when the collector temperature is above that of the receiving water system.

Passive systems involve the arrangement of building windows and shading to allow maximum solar radiation to the building interior during the low sun angles of cold weather months,while shading out the sun during the high sun angles of summer. Large building masses, such as heavy masonry walls and concrete floor slabs, are located to absorb solar heat and minimize heating requirements during hours when the sun is not shining.

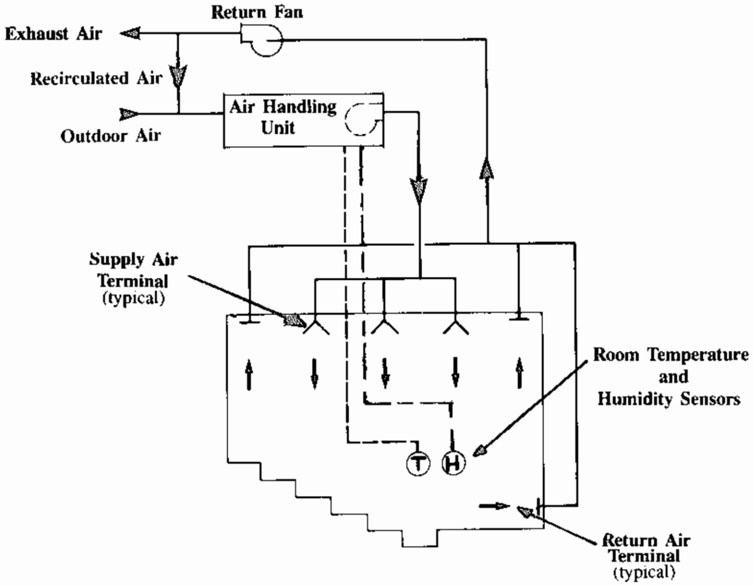
Central Heating, Ventilating and Air-Conditioning Systems

Central HVAC systems provide a conditioned air supply through a ducted distribution system to control the interior environment of a building. A particular system may supply 100 per cent outdoor air with all return air exhausted, or it may operate with a portion of the return air recirculated through the system.

Single Zone HVAC System

A single zone HVAC system (Figure 9) is used to serve a single room, or a group of rooms with similar heating and cooling requirements. All conditioning of the air supply is done at the central air handling unit. Control of the supply air condition may be based on room sensors, or on a predetermined schedule of supply air temperature and humidity. Single zone systems are common for large individual rooms such as retail stores, theatres, meeting halls and gymnasia, or as a ventilating air supply to multiple rooms equipped with other means of local zone control. The system allows the use of outdoor air for *free cooling* and provides good room temperature and humidity control when room sensors are used.

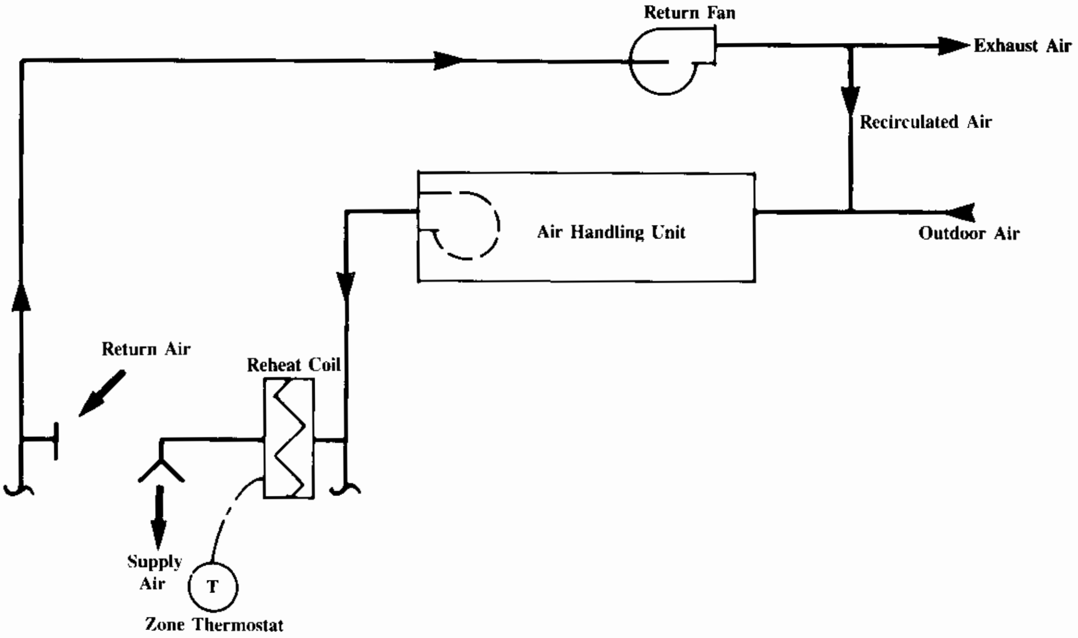
In a building containing a separately controlled heating system, simultaneous operation of the heating and cooling systems can occur and waste energy dollars.



Single Zone Hvac System
Figure 9

Terminal Reheat HVAC System

A terminal reheat HVAC system (Figure 10) provides a central air supply to multiple rooms at a temperature to suit the room requiring maximum cooling. A small heating coil in the air supply to each room or zone reheats the air in response to a room thermostat. The room thermostat may also control perimeter heating in sequence or in parallel with the reheat coil. This system provides good room temperature control, but results in simultaneous cooling at the central air handling unit and heating at the reheat coils. The system is used in office buildings and institutional facilities, and in buildings requiring control of pressure relationships between rooms, such as hospitals and laboratories. Some systems use refrigeration condenser heat in the reheat coils to reduce the energy cost impact of simultaneous heating and cooling.



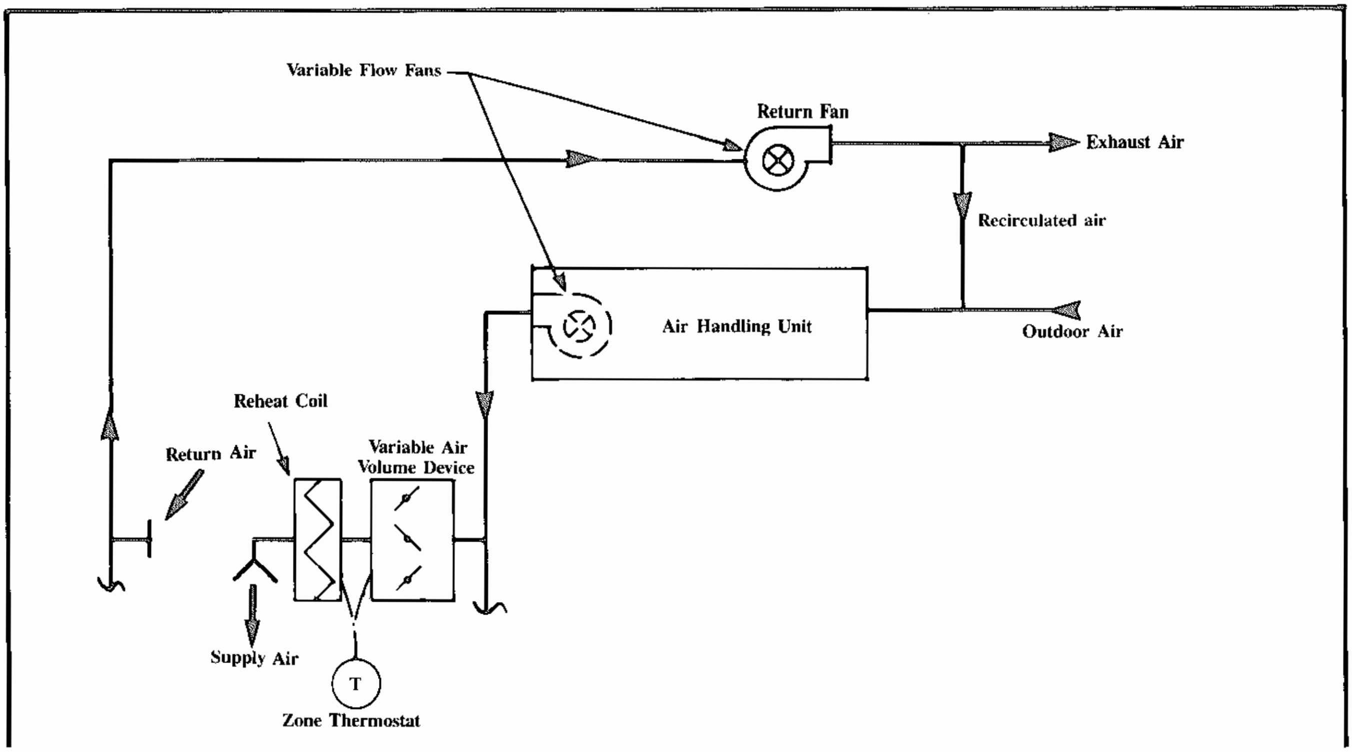
Terminal Reheat HVAC System
Figure 10

Variable Air Volume HVAC System

A basic variable air volume (VAV) system consists of a central air handling unit supplying conditioned air to multiple rooms or zones with a variable air volume control device on each zone supply duct. The central supply air is conditioned to meet the requirement of the zone requiring the most cooling. The VAV devices regulate the air flow in response to zone thermostats. This system is commonly used to serve interior zones where no heating is required, and provides good room temperature control when the supply air temperature is properly regulated. The supply and return air flows are modulated to match the total air flow required by the VAV decices. The supply air pressure required by the VAV decives results in additional fan power at maximum air flow. However, under normal operation the reduced air flow results in a lower average fan power than required by a constant volume system.

Variable Air Volume System With Terminal Reheat

VAV systems that serve building zones requiring heating use a reheat coil in each VAV device (Figure 11). The reheat coil operates in sequence with the VAV device to provide heating at minimum air flow. Such systems reduce the simultaneous heating and cooling loads, but require relatively higher temperature reheat than constant volume reheat systems.

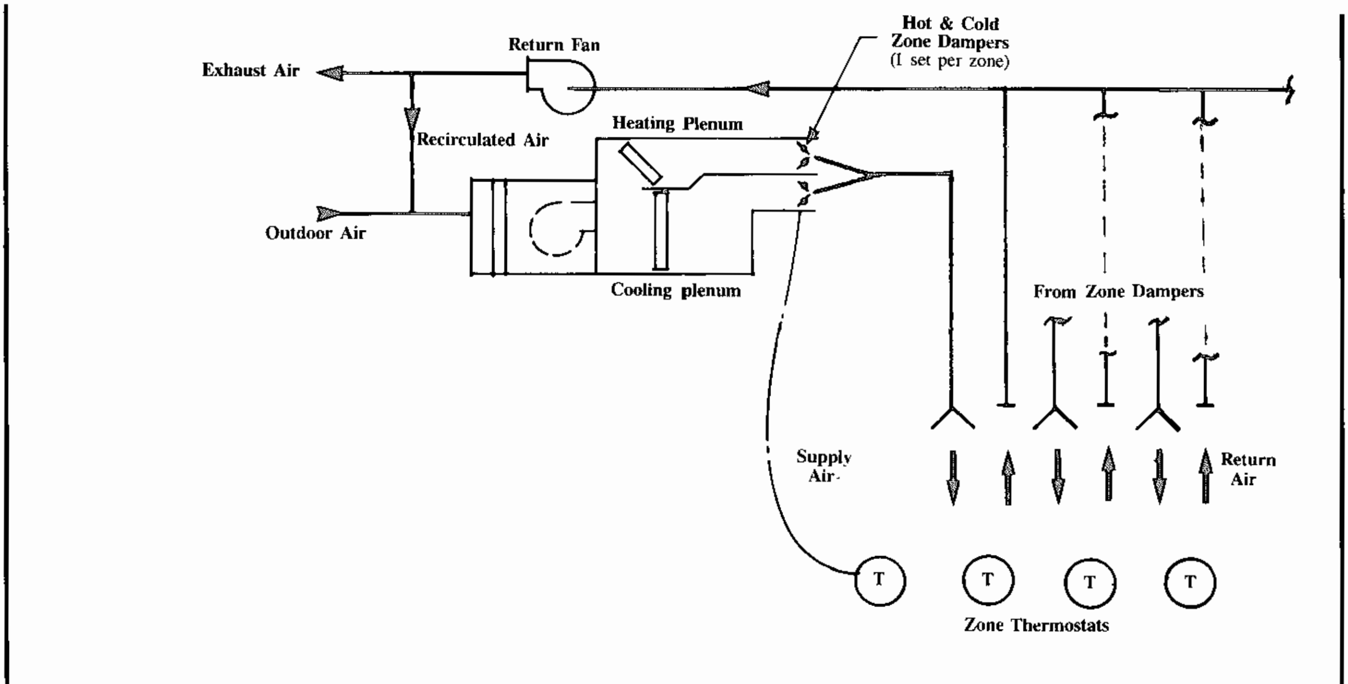


Variable Air Volume With Terminal Reheat HVAC System
Figure 11

Multizone HVAC System

A multizone system (Figure 12) consists of a central air handling unit with parallel heating and cooling plenums. Individual dampers provide a controlled mixture of air from the heating and cooling plenums to each building zone.

Individual ducts are provided to each zone from the mixing dampers, and each set of dampers is regulated by the respective zone thermostat. Careful control of hot and cold plenum temperatures is required to minimize simultaneous heating and cooling.

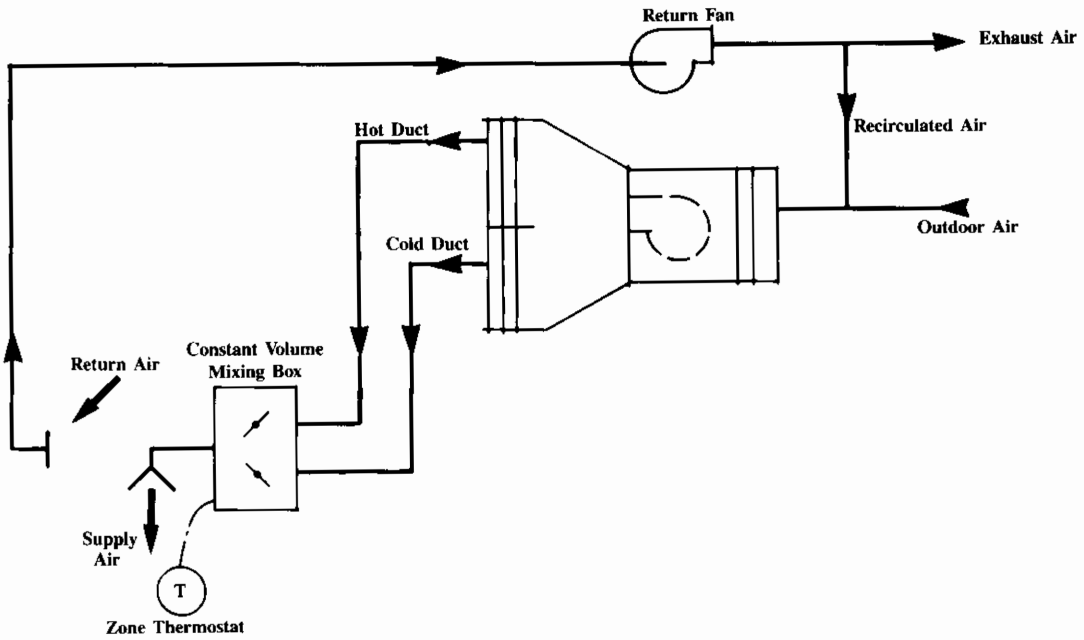


Multizone HVAC System
Figure 12

Dual Duct HVAC System

A dual duct system (Figure 13) consists of a central hot and cold plenum type air handling unit supplying heated and cooled air through parallel duct distribution systems. Devices at each zone provide a controlled mixing of air from the hot and cold duct systems in the proportions required to satisfy the zone thermostat. These systems usually involve high velocity duct design and a high fan power requirement to provide for the variation in air flow in each duct system. Each zone mixing device incorporates a regulating device for maintaining constant air flow at varying inlet pressures.

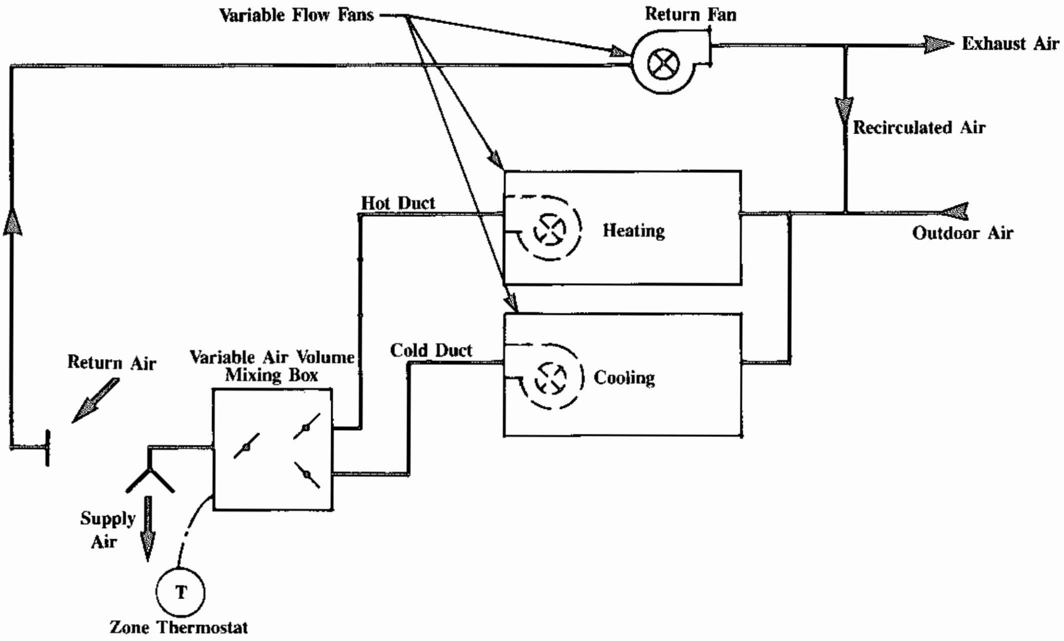
Dual duct systems provide maximum flexibility for zone changes through the addition or relocation of mixing devices. They are used in large commercial buildings where flexibility in zone control is a prime consideration. Hot and cold duct temperatures must be optimized to avoid high energy costs.



Dual Duct HVAC System
Figure 13

Dual Duct Variable Air Volume System

A dual duct, variable air volume system (Figure 14) employs a variable air flow device in each zone mixing device to automatically reduce the air flow during low heating and cooling load periods. The central air handling system may use separate fan units for each of the hot and cold duct systems to allow fan output modulation to match the varying air flow requirements. A dual duct VAV system requires significantly less fan power than a constant volume dual duct system, but higher hot duct temperatures are usually necessary.



Dual Duct, Dual Fan Variable Air Volume System
Figure 14

Packaged Unitary Systems

Unitary systems for heating and cooling employ a packaged local air handling unit in each room or zone. Energy for heating and cooling is distributed to the packaged units in form of hot water, chilled water, electric power, or natural or manufactured gas.

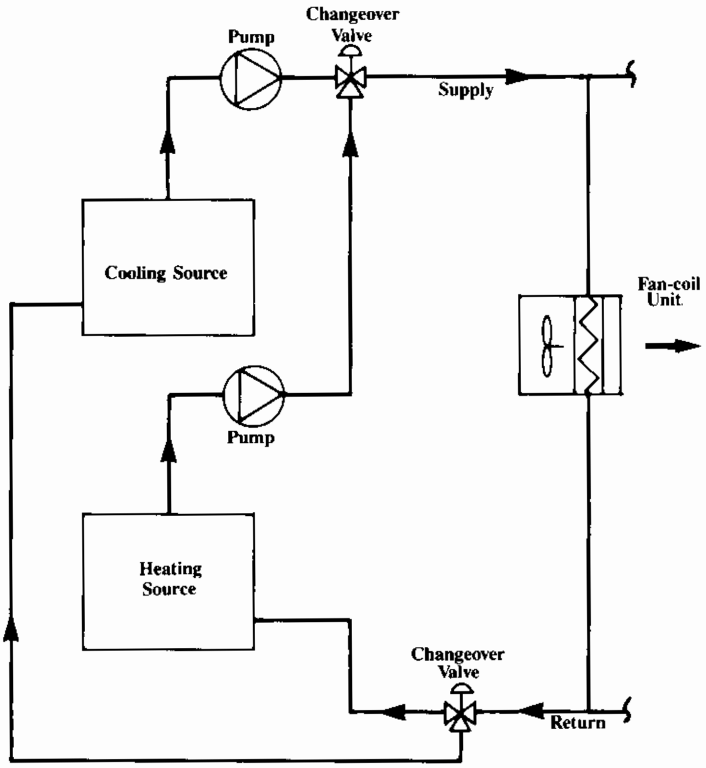
Fan-Coil Unit

A fan-coil unit consists of a cabinet containing a small fan and one or two coils, and provision for low efficiency filters. Units located at an exterior wall may be provided with an outside air intake. The coil is supplied with hot water for heating and chilled water for cooling. The supply systems for heating water and chilled water may be one of three common arrangements.

- Two-pipe system with a seasonal change over from heating to chilled water and vice versa.
- Three-pipe system with both hot and chilled water supply and a common return piping system.
- Four-pipe system with separate supply and return piping for hot and chilled water. Four-pipe systems may employ two coils in the terminal unit: one for heating and one for cooling.

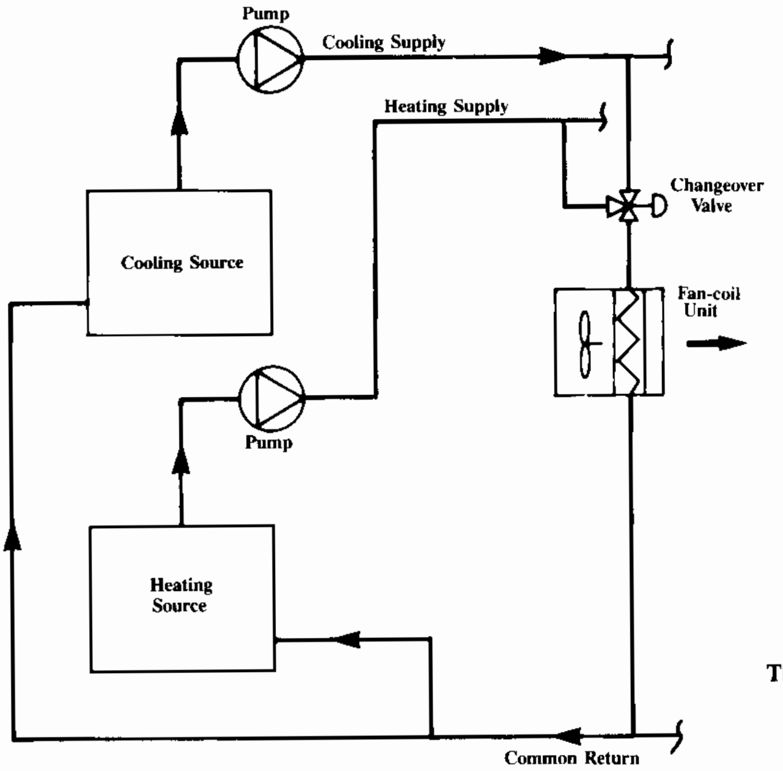
Fan-coil systems are used in office buildings, hotels and schools. They allow shutdown of individual zones when not occupied, and they have a relatively low capital cost. System efficiency depends on piping losses, control of outdoor air intake, and minimizing of cross-over flow between hot and chilled water in single-coil terminal systems.

Figures 15, 16 and 17 are schematics of typical fan-coil unit systems.



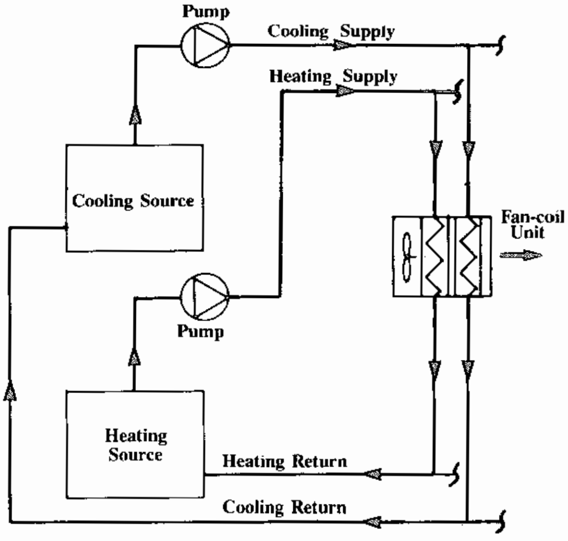
Two-pipe Fan-coil Unit System

Figure 15



Three-pipe Fan-coil Unit System

Figure 16



Four-pipe Fan-coil Unit System
Figure 17

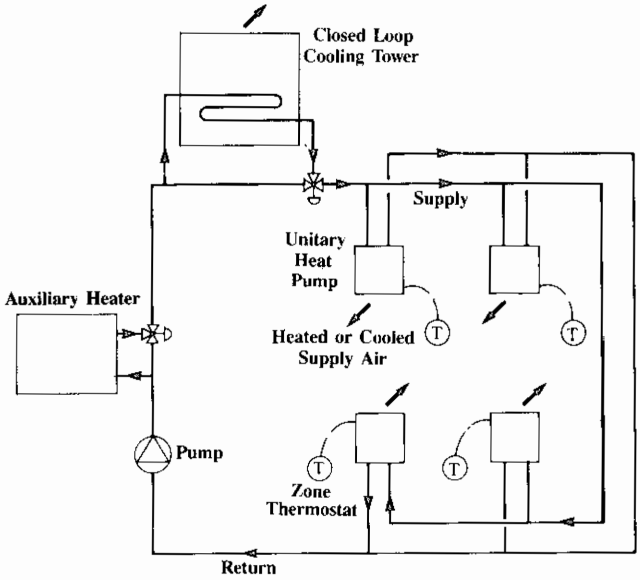
Unitary Air-Conditioner

A unitary air-conditioner system consists of a packaged air-conditioning unit located within each zone. The unit may be mounted on an exterior wall to allow the refrigeration system heat to be rejected directly outdoors. Some systems may utilize a central *cooling tower* water system for heat rejection. Zone heating may be provided by a separate heating device or by an electric heater included in the unit. Unitary air-conditioners are used in motels, hotels, schools and small office buildings. Capital cost for unitary systems is low, but the noise level and maintenance costs are high.

Unitary Heat Pump

A unitary heat pump system consists of a packaged air-conditioning unit with a reversible refrigeration system capable of providing heating and cooling. A complete description is provided in Refrigeration and Heat Pumps, Module 11.

Water-to-air unitary heat pumps (Figure 18) may be located at the perimeter wall, in the ceiling space, or in a fan room. Each unit contains a fan, a refrigerant coil in the air stream, and a heat exchanger connected to a central circulating water loop. On a requirement for cooling, the heat pump cools the air and rejects the condenser heat to the water loop. On a requirement for heating, the refrigeration system is reversed to heat the air by extracting heat from the water loop. The system averages the heating and cooling requirements of all zones, by transfering heat between zones.



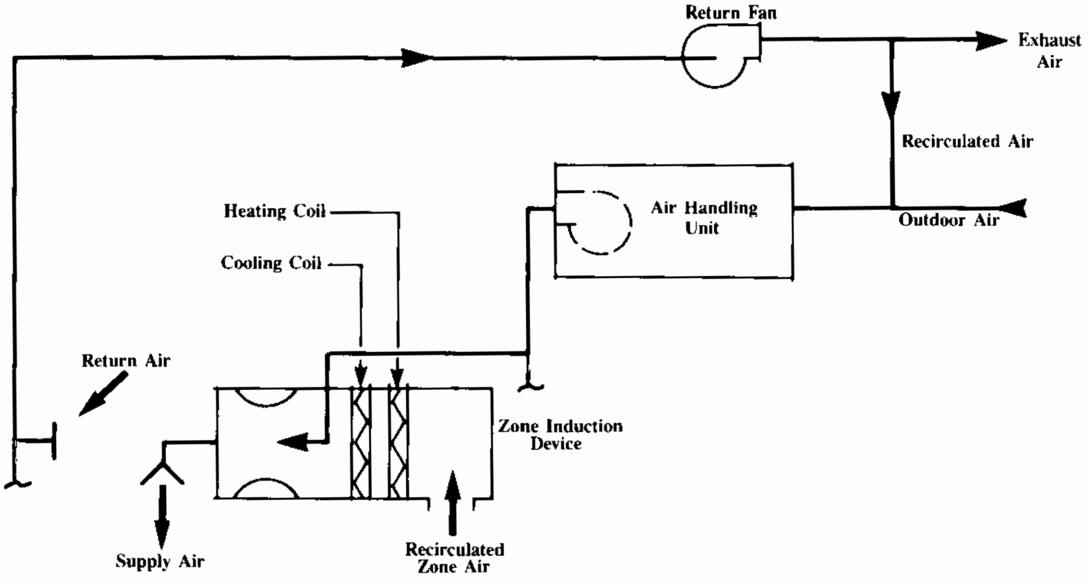
Water-to-air Unitary Heat Pump System
Figure 18

Water-to-air unitary heat pumps are used in office buildings, schools, motels, and institutional buildings. They allow complete shut-down of zones not in use, and make optimum use of internal building heat gains.

Air-to-air unit heat pumps may be mounted in a perimeter wall, or on a roof immediately above the building zone. On a requirement for heating they extract heat from the outdoor air and reject it to the zone. The heating capacity is reduced by a drop in outdoor air temperature, requiring supplementary electric or gas heating for low temperature operation.

Induction HVAC System

An induction system (Figure 19) is a combination central and unitary system. A low volume, high pressure, conditioned central air supply is used in an *aspirator* arrangement to induce room air circulation through terminal units for local heating or cooling in the zones. The central air supply is used to provide ventilation and limited humidity control. The terminal units are supplied with heated or chilled water from piped systems similar to those discussed for unitary fan-coil systems. The system is found in high-rise office buildings, and some hotels and hospitals.



Induction HVAC System
Figure 19

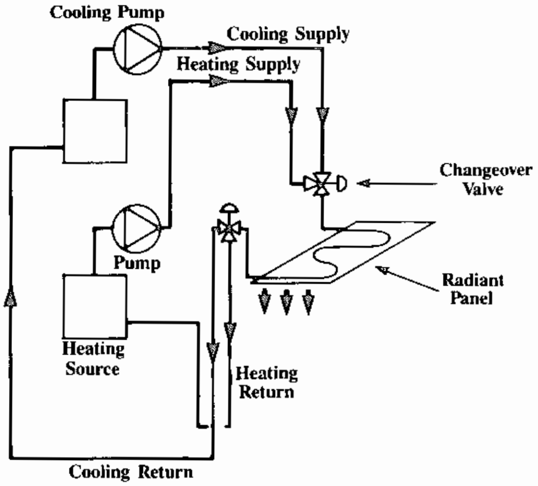
Panel Heating and Cooling Systems

Panel systems provide most of their heating and cooling effect through radiation heat exchange with room surfaces and occupants. Depending on the amount of competing radiation from other sources such as hot equipment, or cold windows and walls, this system can provide comfortable conditions with less air temperature treatment than an all-air system.

Panel systems use pipes or electrical heating cables embedded in the floor or ceiling, or prefabricated unit panels mounted in the walls or ceiling. Most systems utilize a relatively low temperature differential between the panels and the room. To avoid condensation on the panels, dehumidification must be provided by a separate air system. Water piping may be a two-pipe, three-pipe or four-pipe system as described for packaged unitary systems.

Panel systems are used for supplementary heating and cooling to reduce the capacity required of central HVAC. They are used in hospitals, schools, office buildings, and in industrial applications for spot heating or cooling. Panel heating may be used to achieve smooth, unobstructed room surfaces.

Figure 20 is a schematic of a typical heating and cooling system using unit panels and a four-pipe water system.



Panel Heating And Cooling System
Figure 20

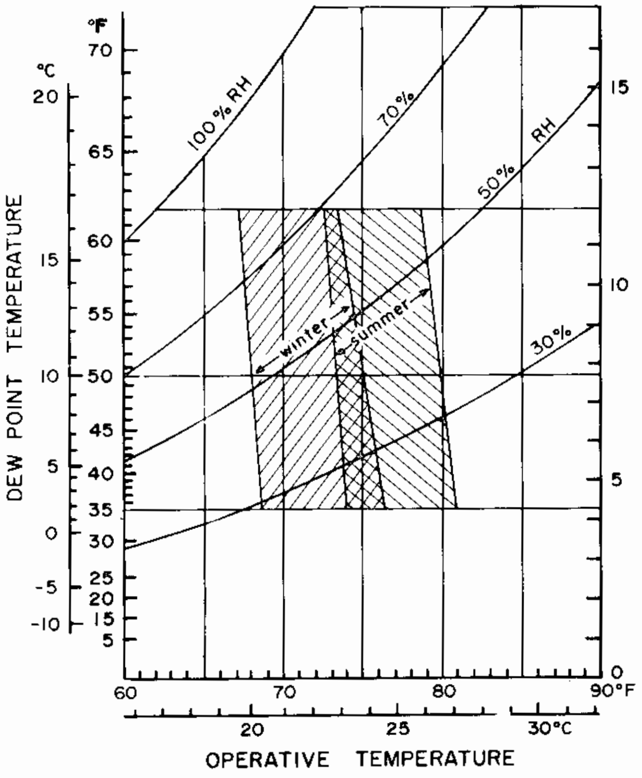
Air Distribution and Control

Proper air distribution and control is very important for successful HVAC performance, and can have a significant effect on energy cost.

Comfort Standards

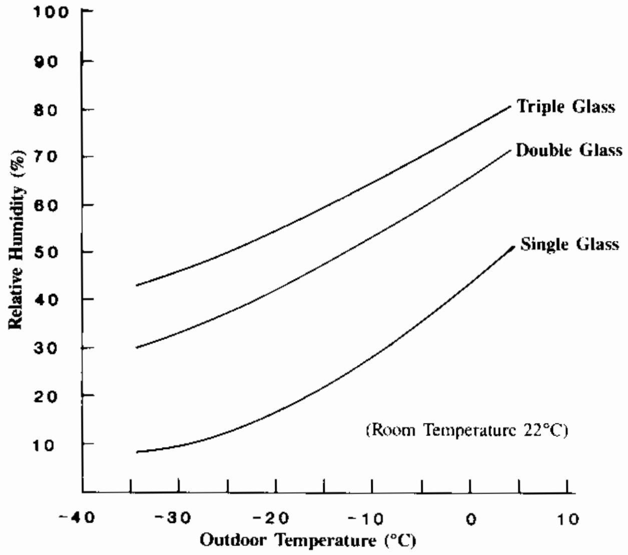
Primary factors which affect human comfort are temperature, humidity, air motion and air quality (odor, dust or toxic gases). Secondary factors include clothing worn by occupants, activity, the presence of warm or cool radiating surfaces, and variations in individual preference.

ASHRAE has defined a *comfort envelope* of room conditions (Figure 21). The difference between summer and winter ranges are based on normal seasonal clothing habits of the occupants. A detailed description of these factors is given in ASHRAE Standard 55-1981.



Comfort Envelope
Figure 21

Courtesy of ASHRAE



Condensation On Windows
Figure 22

The level of humidity for comfort may be limited in winter by potential condensation on the building structure, principally at windows. Figure 22 is a plot of the humidity levels at which condensation may occur on windows at various outdoor temperatures.

Health Standards

Most regions have health and occupational environment standards defined by regulation. Such standards normally define:

- Minimum ventilation rate using outdoor air in L/s for each person or in L/s per square metre of floor area.
- Maximum and minimum temperature and humidity levels.
- Maximum allowable concentrations of airborne contaminants.

ASHRAE Standard 62-1981 lists recommended standards for air quality and ventilation rates for a range of applications.

Some buildings contain sensing systems to detect unacceptable levels of contaminants and initiate increased ventilation rates. Examples include carbon monoxide sensors in parking garages and carbon dioxide sensors in large assembly buildings.

Room Diffusion and Stratification

Conditioned air should be supplied to a room and diffused in such a way as to minimize air velocity around the occupants. Acceptable values for maximum velocity at the occupant level are 0.15 m/s in winter and 0.25 m/s in summer. An exception to these limits would occur where higher velocities are used for spot cooling.

Natural convection tends to cause warm air to stratify near the ceiling. Such an effect can occur where heating is supplied from the ceiling level, or in buildings with high infiltration and poorly insulated walls. Diffuser settings to overcome cold-weather stratification may result in excessive air velocities at the occupant level during warm weather.

Cold-weather stratification is common in areas with high ceilings, such as automotive service garages. Air temperature at the ceiling can be more than 15°C higher than the thermostat setting at occupant level, resulting in high heat loss through the roof. Slow-speed propeller fans suspended from the ceiling may be used to overcome stratification by creating air circulation between the ceiling level and the occupant level. Other destratification devices include floor-to-ceiling ducted fan systems and high velocity induction air jets.

Natural convection forces may be used to advantage in cooling applications by exhausting hot air accumulated at high level from heat producing equipment, while providing spot cooling to equipment operators at a lower level.

Energy Audit Methods

Energy Management Opportunities exist in HVAC systems in industrial, commercial or institutional facilities. Many of these opportunities are recognizable during a *walk through audit* of the facility. This audit is usually more meaningful if a "fresh pair of eyes" generally familiar with energy management is involved.

Typical energy saving items noted during a walk through audit might include; damaged or improperly adjusted thermostats, loose damper linkages, exhaust systems running when not required, dirty filters in air handling systems, and worn or damaged drive belts. Alert management, operating staff and good maintenance procedures can with a little effort, reduce energy usage and save money.

Not all items noted in the walk-through audit are as easy to analyze as those previously described. For example it may be noted that an air handling system serving an office building is operated 24 hours per day to serve a continuously staffed computer facility, even though most of the building is occupied only 8 to 9 hours per day. The apparent solution is the installation of a separate local air system to serve the computer facility. This leads to certain key questions.

- What size local air system is required?
- Should the local system be a unitary type, stand-alone system, or should it be connected to the central heating and chilled water plant?
- What are the implications to other parts of the building of shutting down the central air system during the unoccupied periods?
- How long will it take for the energy cost saving to pay back the capital cost?
- Are there other solutions, such as applying variable air volume to the central system, which might achieve slightly less saving, but have faster payback?

This example requires a *diagnostic audit* to mathematically establish the reduction in energy consumption and the potential cost saving. With the cost saving plus the estimated cost to supply and install the modified system, simple payback calculations can establish the financial viability of the opportunity.

The implementation of Energy Management Opportunities can be divided into three categories.

- *Housekeeping* refers to an energy management action that is *repeated on a regular basis but never less than once a year.* Examples are changing of air filters, calibration of control systems, steam trap maintenance, and other preventive maintenance procedures.
- *Low Cost* refers to an energy management action that is *done once and for which the cost is not great.* Examples are the installation of time clocks to shut down systems when not required, and the relocation of zone thermostats to prevent overheating or overcooling.
- *Retrofit* refers to an energy management action that is *done once and for which the cost is significant.* Examples are the conversion of a constant volume air system to variable air volume, and the installation of heat recovery systems.

It must be noted that the division between low cost and retrofit is normally a function of the size and the type of the organization.

Summary

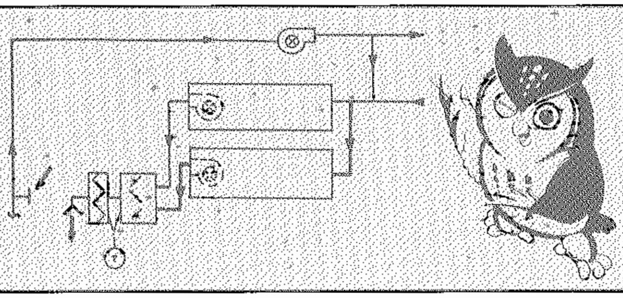
Numerous energy and cost saving opportunities exist in heating, ventilating and air-conditioning systems. Alert personnel, with an awareness of energy management techniques, can easily learn to recognize these opportunities and reap dividends.

The energy consumption by building HVAC systems is affected by several factors.

- Heat loss and gain through the building enclosure.
- Uncontrolled ventilation caused by infiltration.
- Internal heat gain from equipment and processes.
- Outdoor air to replenish exhaust air.
- The efficiency at which the HVAC system delivers the necessary heating and cooling energy.

The subject of energy management must be approached with an open mind to expose previously-accepted, but inefficient practices. The opportunities listed in the Energy Management Opportunities section of this module may generate similar or additional ideas that are specific to the facility. Energy management awareness on the part of management and operating staff, combined with an active energy management program, can pay dividends in energy and cost reduction.

EQUIPMENT SYSTEMS



This section describes heating ventilating and air-conditioning equipment which may be encountered in the Industrial, Commercial and Institutional sectors.

Boilers and Furnaces

Boilers and forced air furnaces deliver heat energy from a fuel to the water, steam or air of a distribution system. The operating cost is affected by several factors.

- Combustion efficiency.
- Efficiency of heat exchange to the distribution medium including heat loss to stack gases and heat loss to the boiler or furnace room.
- Energy required for fuel feeding systems and combustion air fans.

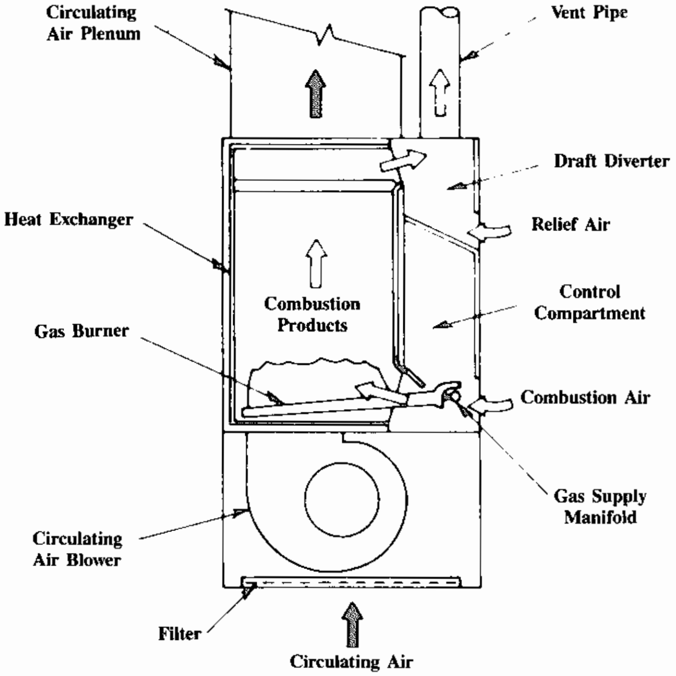
Boilers

Boilers are described in Boiler Plant Systems, Module 6.

Furnaces

Air heating furnaces (Figure 23) are used to heat and distribute air in single zone heating systems, and are available in sizes up to about 4700 MJ/h. Output is normally regulated by starting and stopping the burner. Periodic cleaning of the combustion side of heat exchange surfaces and fine tuning of the burner are important for maintaining optimum efficiency. An average annual efficiency of 75% is a realistic target.

Combustion systems are described in more detail in Combustion, Module 5.



Air Heating Furnace
Figure 23

Courtesy of ASHRAE

Refrigeration

Refrigerated cooling is provided to an HVAC system by one of two systems.

Direct expansion systems are used where accurate control of the cooling air temperature is not required, and where the refrigeration compressors can be located close to the air system.

Chilled liquid systems are used where it is necessary to convey cooling from a central chiller to multiple air handling systems and other terminal cooling devices.

The common types of refrigeration systems for HVAC purposes use electrically driven compressors, or an absorption process with a heat energy source.

The principles and equipment of refrigerated cooling are described in Refrigeration and Heat Pumps, Module 11.

Electrically Driven Chillers

Electrically driven systems require an average overall energy input of approximately 80 kWh per GJ of cooling including an allowance for pumps and cooling tower fan energy. Energy consumption increases with an increase in the difference between condenser temperature and chilled water temperature.

Absorption Chillers

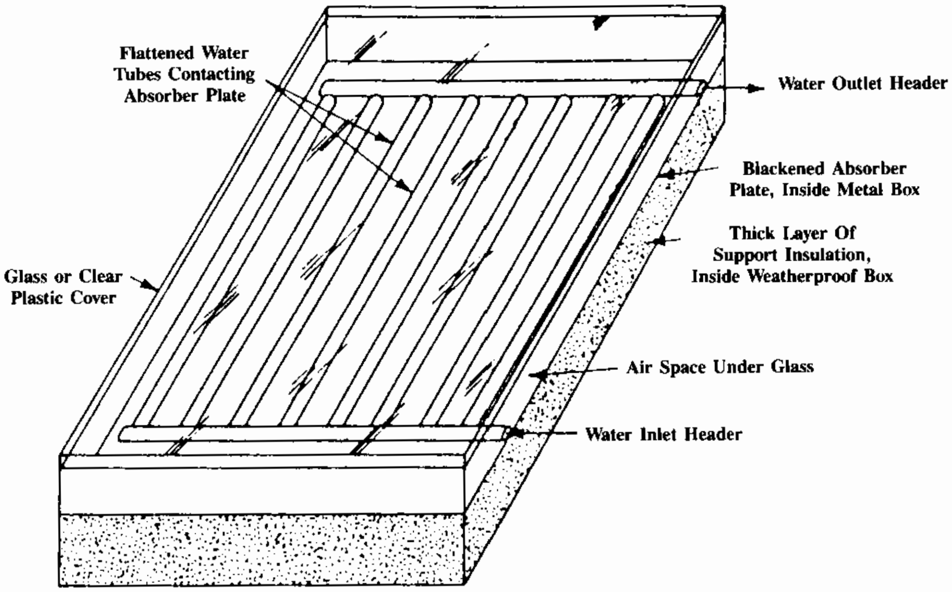
Absorption systems use heat energy to create cooling. Absorption chillers require an energy input of approximately 1.4 MJ per MJ of cooling. Because of the high energy consumption, absorption systems are normally used only where a low cost heat source is available.

Solar Collectors

Flat Plate Collectors

Flat plate collectors (Figure 24) are designed to collect solar radiation directly into a working fluid such as water, an antifreeze solution or air. They are normally used for water heating and space heating at fluid temperatures of up to 90°C.

Efficiency is affected by transmission and radiation losses to outside air. Typical efficiencies of flat plate collectors are 50 to 60 per cent for summer operation and 35 to 50 per cent for winter operation.

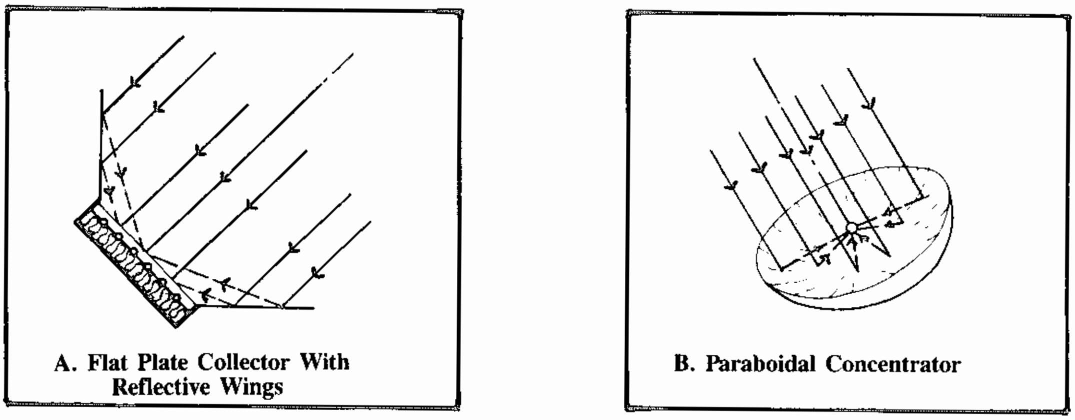


Flat Plate Solar Collector
Figure 24

Courtesy of ASHRAE

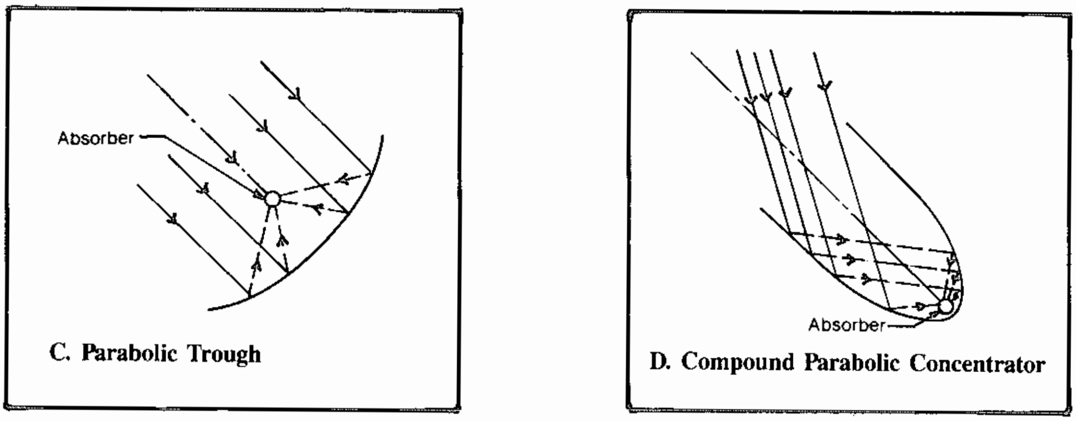
Concentrating Collectors

Concentrating collectors (Figure 25) can produce working fluid temperatures above 150°C, and may be used as a heat source for absorption cooling. For optimum performance concentrating collectors must be motorized to keep the concentrator directed toward the sun.



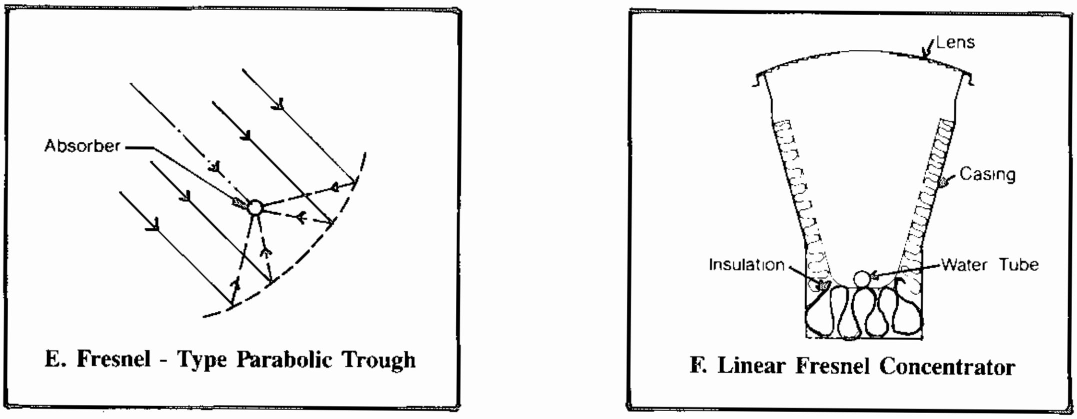
A. Flat Plate Collector With Reflective Wings

B. Paraboidal Concentrator



C. Parabolic Trough

D. Compound Parabolic Concentrator



E. Fresnel - Type Parabolic Trough

F. Linear Fresnel Concentrator

Concentrating Solar Collectors
Figure 25

Courtesy of ASHRAE

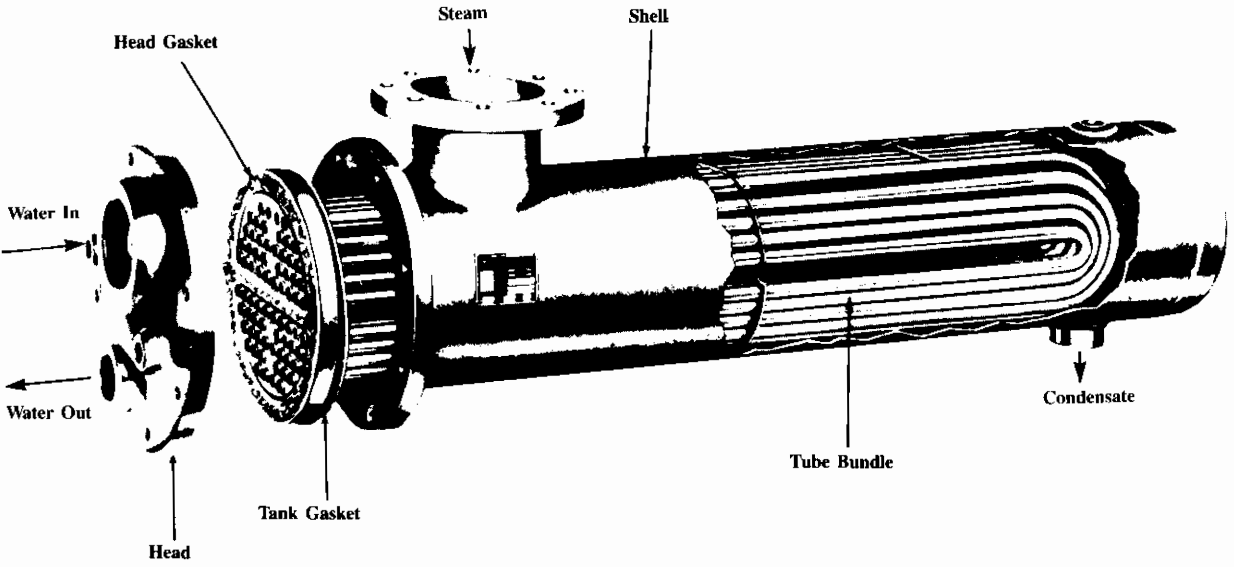
Heat Exchangers

Steam-to-Water

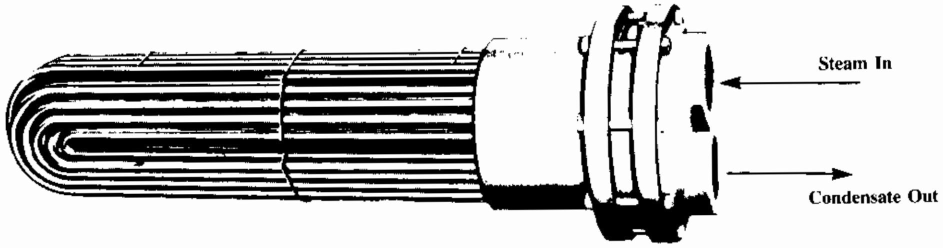
Steam-to-water heat exchangers are used in HVAC systems in two configurations: Shell and tube (Figure26) and U-tube immersion type (Figure 27).

When steam is supplied to a shell and tube heat exchanger care must be taken to ensure free condensate drainage. Periodic cleaning of heat exchange surfaces is required to maintain the heat exchange rate.

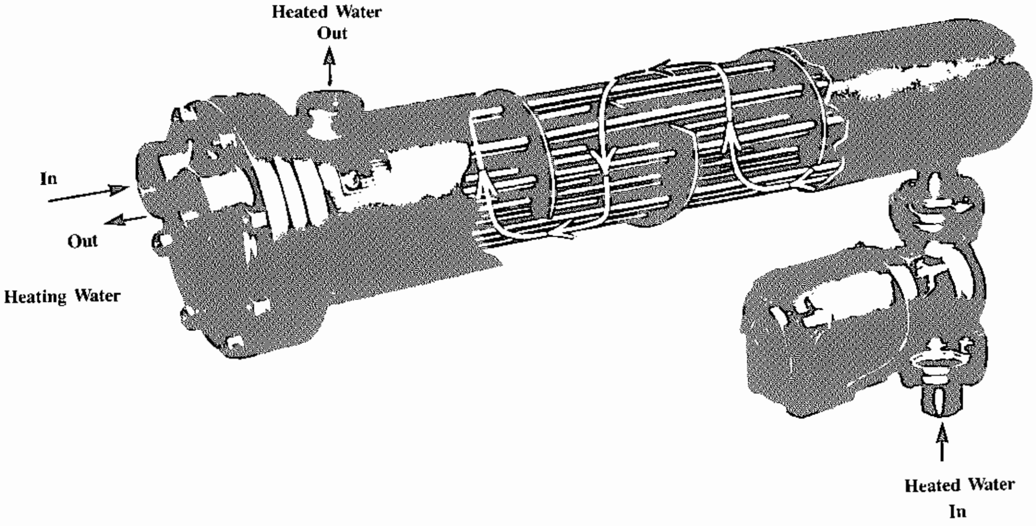
Heating and Cooling Equipment (Steam and Water), Module 9 describes steam heaters and heat exchangers.



Shell And Tube Steam Heat Exchanger
Figure 26



U-tube Immersion Steam Heat Exchanger
Figure 27



Shell And Tube Water Heat Exchanger
Figure 28

Water-to-Water

Water-to-water and steam-to-water heat exchangers are used in similar configurations. Shell and tube type water-to-water heat exchangers (Figure 28) contain baffles in the shell to ensure turbulent flow around the tubes for effective heat transfer.

Plate type water-to-water heat exchangers (Figure 29) are used to achieve a low temperature difference between the heating, and the heated water. Such exchangers are used frequently in cooling water applications, and in heat recovery systems.

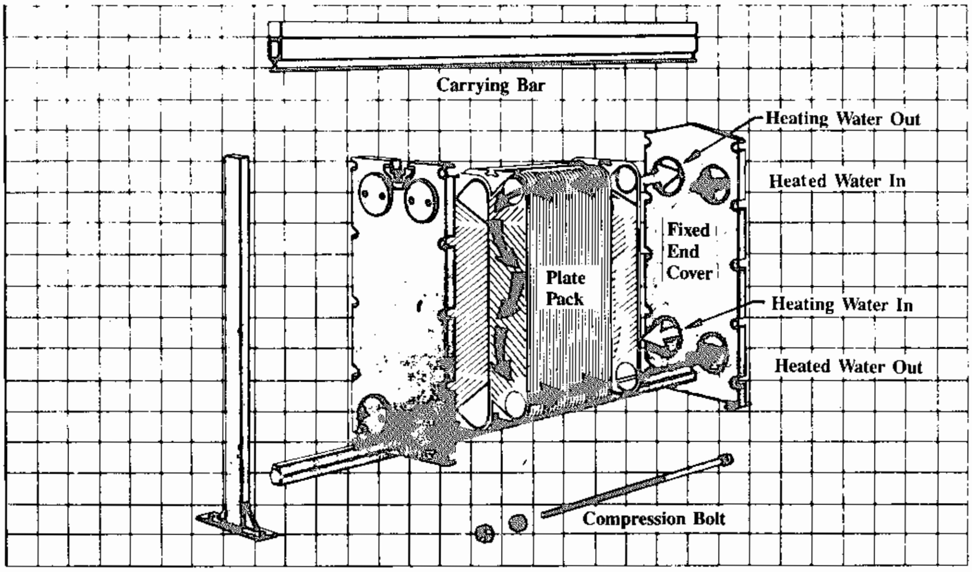


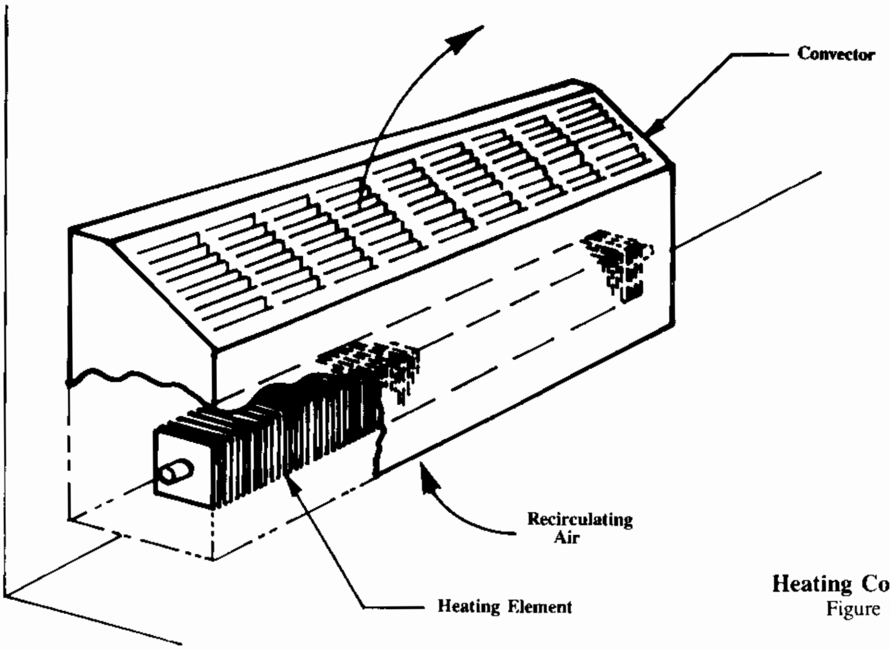
Plate Type Heat Exchanger
Figure 29

Space Heating Units

Convectors

A heating convector consists of a finned-tube heating element in a vertical enclosure (Figure 30) designed to act as a chimney and provide air flow over the element. The convection air flow and heat output is affected by the height of the enclosure: the higher the enclosure the greater the heat output. Manual dampers in the enclosure can be used to reduce the heat output by up to 80 per cent. Convectors are used in a variety of configurations for steam, hot water and electric heating.

Convectors at an exterior wall may have a high heat loss to outdoors if the wall is poorly insulated or sealed against air leakage. Periodic cleaning of the finned element is required to maintain the heating output.



Heating Convector
Figure 30

Fan-Coil Heaters

Fan-coil heaters contain a finned-tube heating element and a fan to provide forced air circulation over the heating element (Figure 31). This arrangement provides a high heating output and requires less space than a convector of comparable output.

Fan-coil heaters are used in areas such as entranceways and vestibules which require high heating rates.

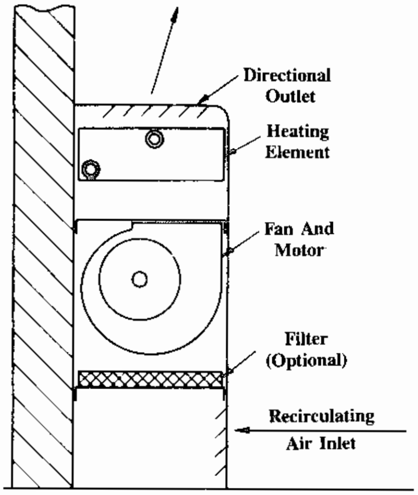
The heat output from steam and water heated units may be controlled by starting and stopping the fan without interrupting the steam or water flow. This results in a continuous heat output caused by convection when the fan is off.

For optimum performance, filters should be replaced regularly and the heating element and fan cleaned periodically.

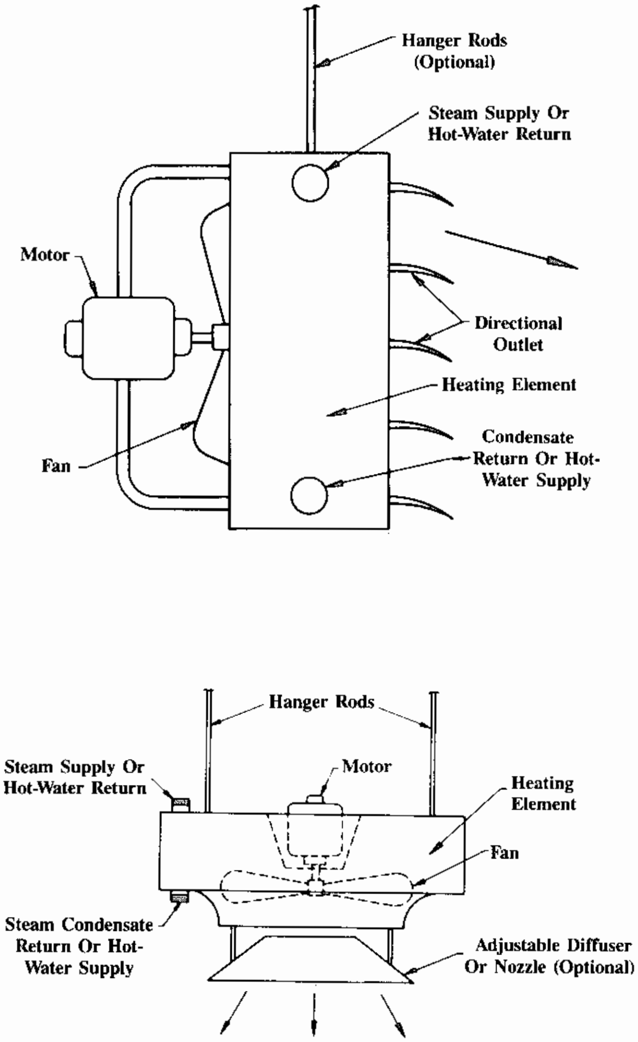
Unit Heaters

Unit heaters provide a low cost source of heat for spaces in which noise levels and high air motion are not a principal concern. Typical applications include industrial plants, service garages and warehouses. Unit heaters are available in a variety of configurations (Figure 32), and may utilize steam, water, electricity, oil or gas as the heating source.

Steam and water heated units may be controlled by starting and stopping the fan without interrupting the steam or water flow. When the fan is off a significant continuous heat output occurs. Since unit heaters are usually mounted near the ceiling, the continuous heat output can go unnoticed during long periods when no heat is required.



Fan-coil Heater
Figure 31



Unit Heaters
Figure 32

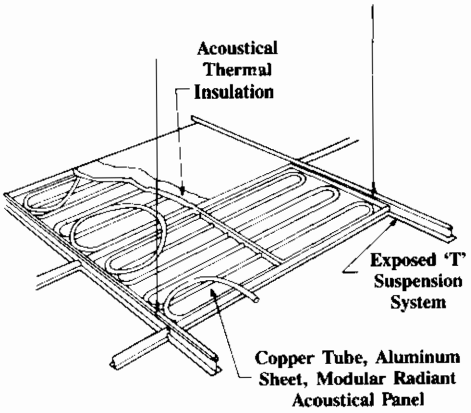
Courtesy of ASHRAE

Radiant Heaters

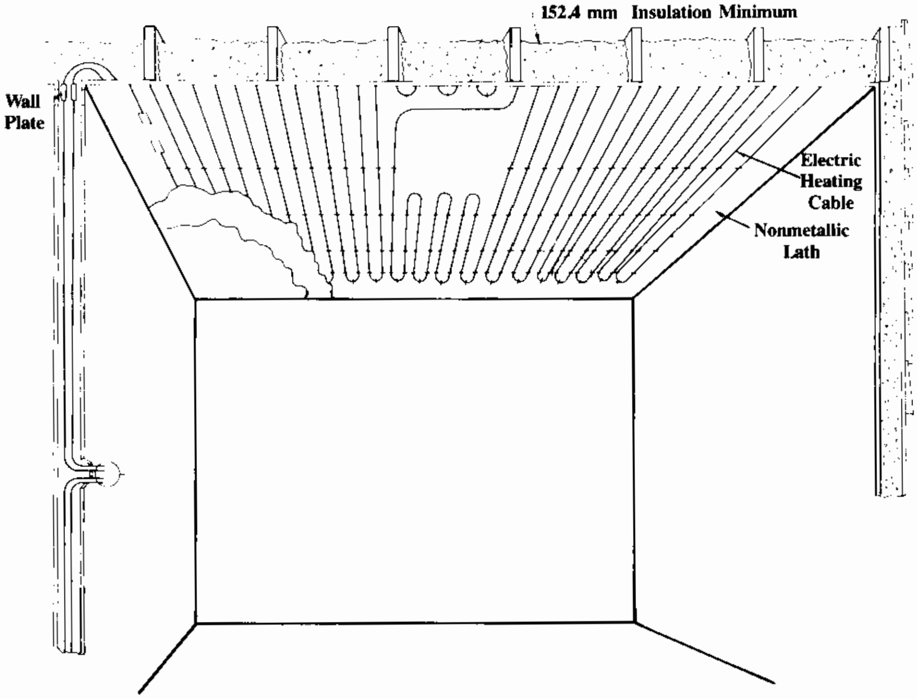
Radiant heaters are available in a variety of configurations.

- Low temperature water heated panels.
- Low temperature electric panels.
- Electric infrared heaters.
- Gas fired infrared heaters.

Low temperature panel heaters (Figures 33 and 34) operate at a temperature of 30 to 90°C. Insulation must be provided behind the panels to minimize heat loss from the space. Painting of low temperature radiant panels can seriously affect the output rate.



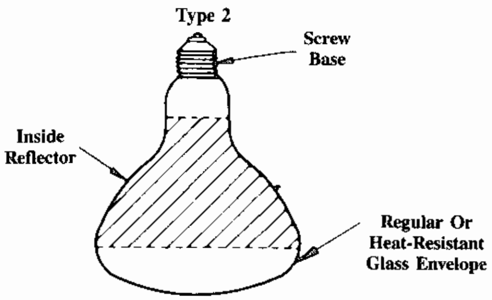
Low Temperature Water Heated Panel
Figure 33



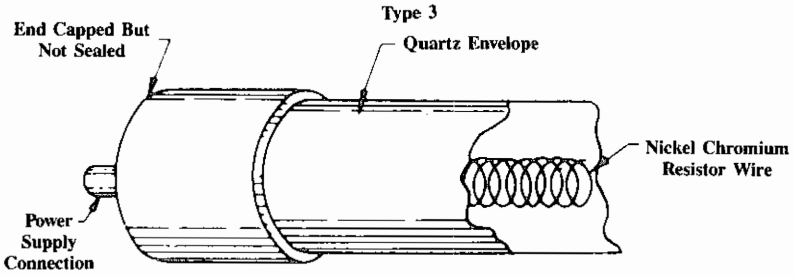
Low Temperature Electric Panels
Figure 34

Courtesy of ASHRAE

Infrared heaters (Figures 35 and 36) are used for spot heating where it is impractical to heat the air in the space. Examples include entrance canopies, loading docks and unheated industrial plant areas. Infrared heaters with manual control are often left operating when not required, resulting in a waste of energy.

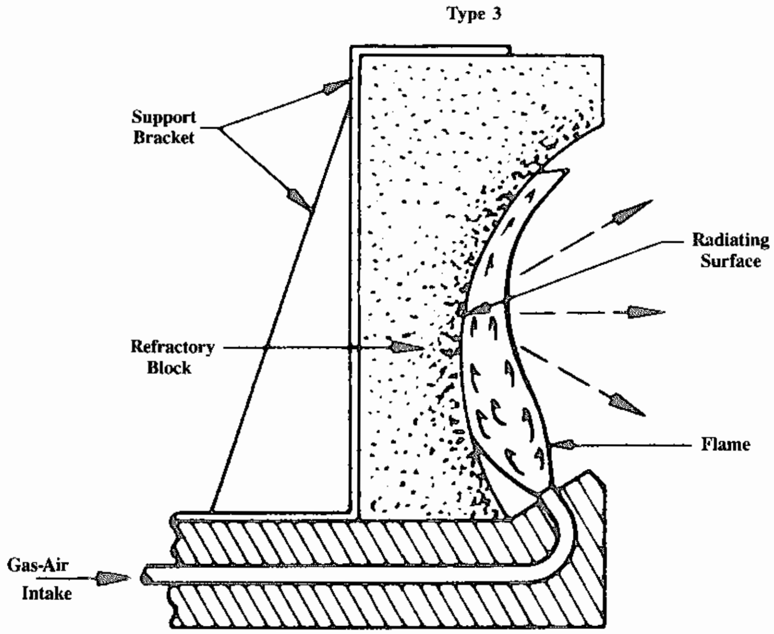


Reflector Lamp Element



Quartz Tube Element

Electric Infrared Heaters
Figure 35



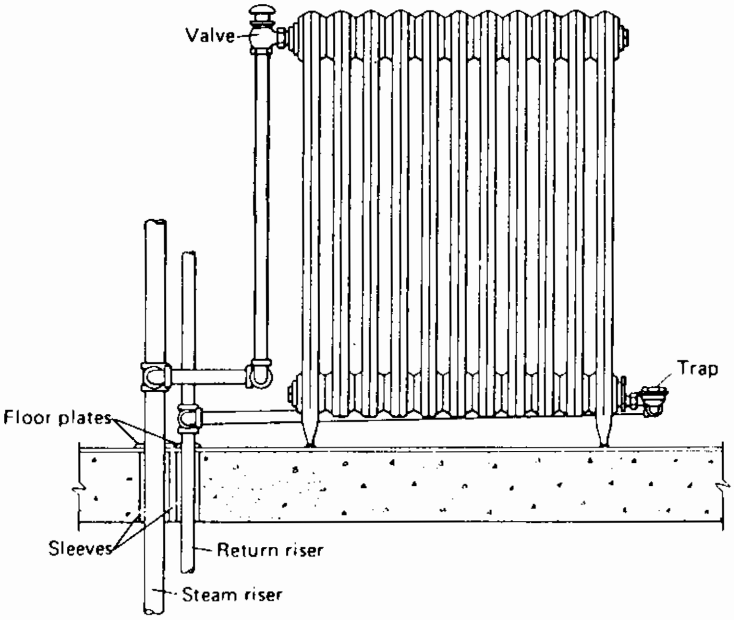
Direct-fired Refractory I-R Burner

Gas Fired Infrared Heater
Figure 36

Courtesy of ASHRAE

Cast Iron Radiators

Many buildings erected before 1940 contain sectional cast iron radiators (Figure 37) for heating. The systems were designed to operate with low temperature water or steam at less than 35 kPa (gauge). When served by an automatic boiler plant with a higher heating water temperature or steam pressure, such units are difficult to control. The heat storage capacity and slow response of cast iron radiators often results in uncomfortable fluctuations in space temperatures and overheating during mild outdoor temperatures.



Sectional Cast Iron Radiator

Central Air Systems

Single Duct Air Handling Unit

A single duct air handling unit (Figure 38) provides a supply of tempered and conditioned ventilating air. A typical unit has several features.

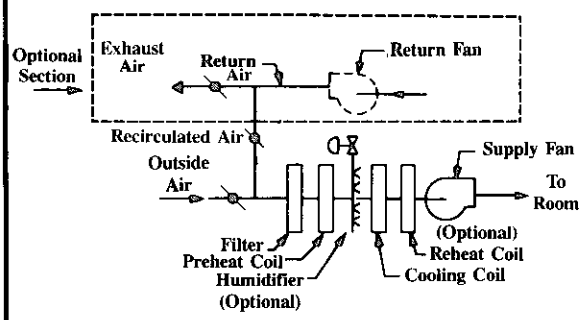
- Exhausts return air through a fan (optional in small systems), capable of overcoming the return duct system pressure drop.
- Takes in outdoor air.
- Controls mixing of return air and outdoor air.
- Filters the mixed air.
- Preheats the mixed air to an acceptable minimum supply air temperature.
- Humidifies the preheated air to maintain an aceptable minimum relative humidity in the space.
- Cools and dehumidifies the supply air to limit the maximum relative humidity in the space.
- Delivers the conditioned air to the space through a fan capable of overcoming the pressure drop of the air handling unit and the supply duct system.

A single duct air handling unit for conditioned make-up of 100 per cent exhausted air may consist of an outdoor air intake, filters, preheat coil and supply fan.

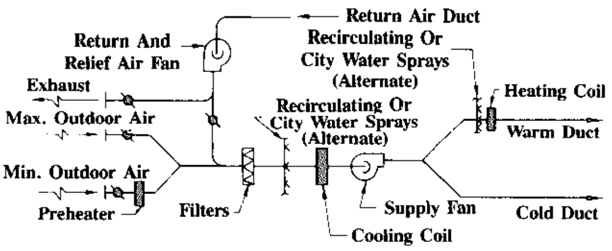
A reheat coil is used on systems requiring low upper-limit humidity control, or where the space has a high internal latent heat gain. For office space the inherent dehumidification resulting from meeting the sensible cooling load will maintain an acceptable level of relative humidity.

Efficient energy performance is affected by several factors.

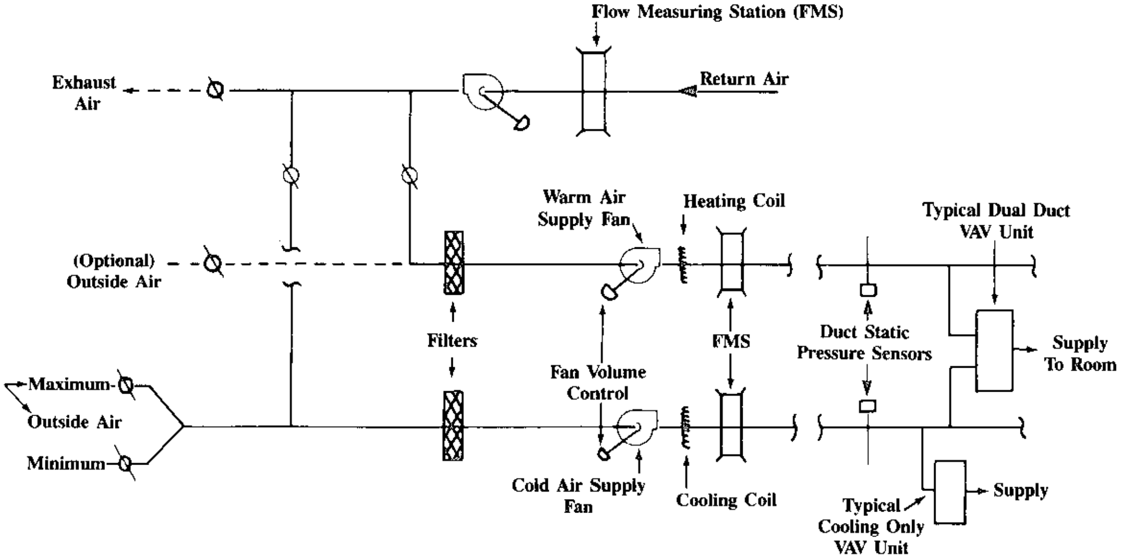
- System control to prevent simultaneous heating and cooling.
- Optimum mixing of return air and outdoor air to minimize both the heating and cooling loads.
- Regular cleaning of filters and coils to minimize fan power and optimize the air handling efficiency of the system.



Single Duct Air Handling Unit
Figure 38



Single Fan Dual Duct Unit
Figure 39



Dual Fan Dual Duct Unit
Figure 40

Courtesy of ASHRAE

Dual Duct Air Handling Unit

A dual duct unit simultaneously provides both cooling and heating air through parallel ducts to zone terminal devices which mix the air streams to suit the zone requirements.

A *single fan dual duct* unit (Figure 39) is a common arrangement for small systems. Return and outdoor air are mixed, filtered, preheated and humidified as in a single duct unit. Ideally, the hot air stream is heated to the temperature required by the zone requiring the most heating and the cold air stream is cooled to the temperature required by the zone requiring the most cooling. In many installations the hot and cold ducts are either operated at constant temperatures to suit the peak requirements, or the temperatures are varied on a fixed schedule based on outdoor air temperature. Optional *load analyzer controls* are available which monitor the signals from zone thermostats and provide optimum control of the hot and cold duct temperatures.

A constant flow supply fan must be capable of delivering the maximum design air flow through each duct for peak heating and peak cooling load conditions. With the air flow normally shared between the two ducts this results in higher than necessary duct pressures and a waste of fan power. Air flow monitoring at the supply fan, combined with fan volume control, allows modulation of the fan power to match the total system air flow requirement.

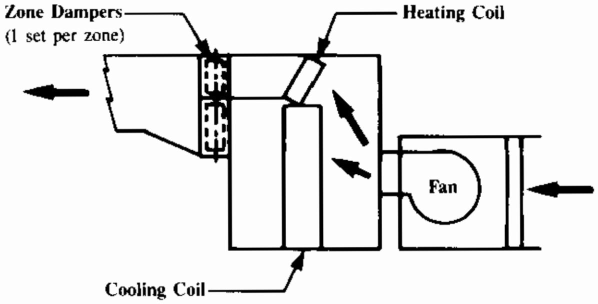
A *dual fan dual duct* unit (Figure 40) uses a separate air handling unit for each duct and requires complete duplication of controls. Depending on the minimum outdoor air requirement, the cold duct unit may require a preheat coil to maintain an acceptable minimum supply air temperature.

Multizone Air Handling Unit

A multizone air handlng unit (Figure 41) is similar to a single fan dual duct system, except the mixing of the air streams is done at the central air unit. A separate supply duct is provided from the air handling unit to each zone. A preheat coil is not required when the required outdoor air intake allows an acceptable minimum mixed air temperature. A return air fan is not required on systems with low return air pressure loss.

Zone damper leakage on a multizone air handling unit may require an increased temperature differential between the hot and cold plenums to satisfy the zone requirements.

Hot and cold plenum temperatures are based on maximum space heating and cooling loads similar to a dual duct system. Load analyzer controls can be used to reset the temperatures to optimum levels.

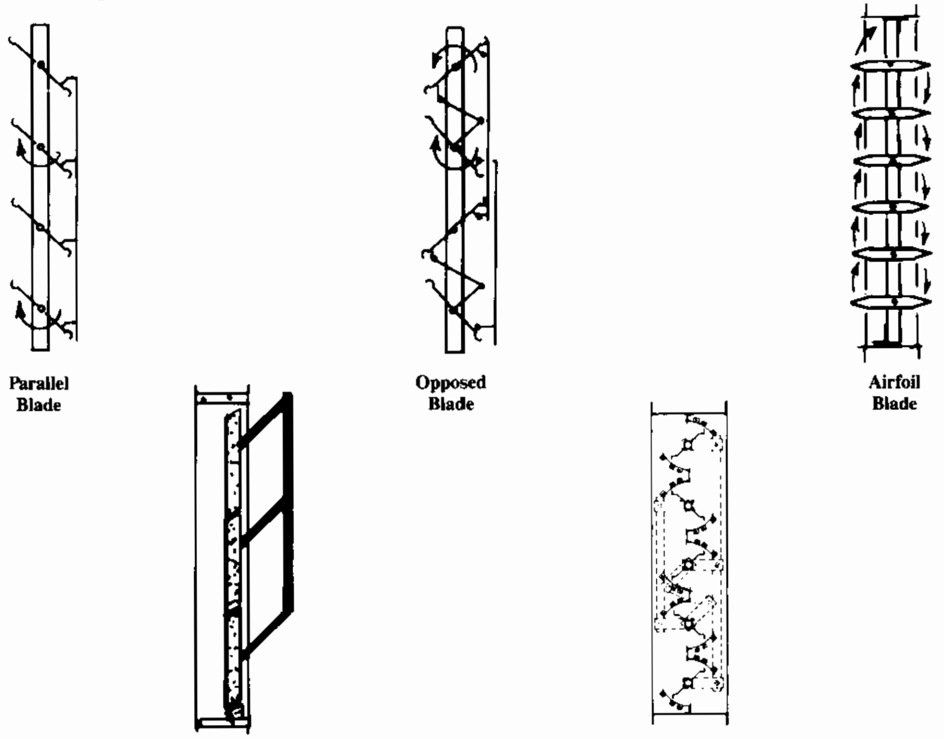


Multizone Air Handling Unit
Figure 41

Dampers

To efficiently control air flow, dampers require regular maintenance. Incorrectly adjusted or loose damper linkages account for substantial energy loss resulting from air leakage and destabilizing effects on the automatic controls.

Dampers are available in various configurations (Figure 42). All dampers that require tight shut-off should be equipped with flexible edge seals and effective linkages to ensure that all damper blades close completely. In some locations protection against cold weather ice formation on damper blades, in the form of electric heating elements or special damper materials, may be required.



Parallel Blade

Opposed Blade

Airfoil Blade

Insulated Blade

Linear Flow Characteristics

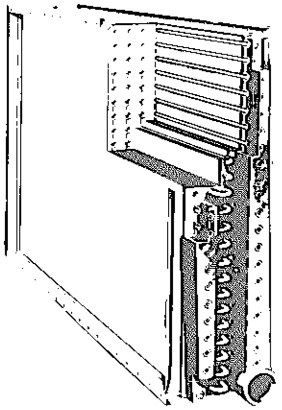
Dampers
Figure 42

Air Heating Coils

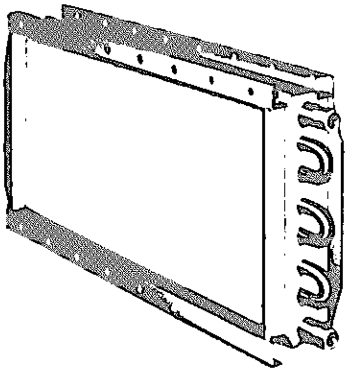
Air heating coils are normally of finned-tube construction to offer the maximum possible heat exchange area to the air stream. Large coils are normally multiple tube type with inlet and outlet headers (Figure 43), while small coils may have a continuous serpentine tube (Figure 44).

A *steam* coil may have inner distribution tubes (Figure 45) to reduce temperature variation across the coil face. A steam coil that is exposed to air temperatures below 0°C may be subject to freezing at the condensate outlet if the steam supply is modulated. For such applications a number of coils are mounted in series, or a single coil is mounted with face and by-pass dampers (Figure 46).

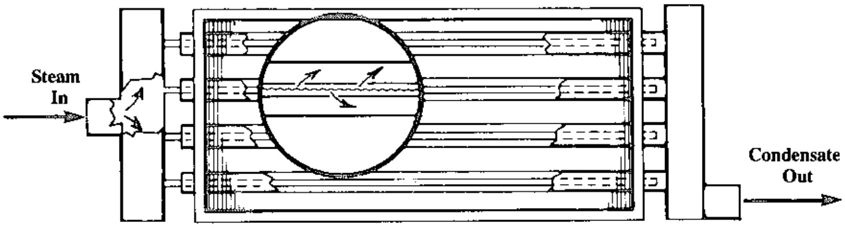
A *hot water* coil may have various tube circuiting arrangements for optimum heat exchange with the available temperature difference and the particular fluid being used. Large water coils are usually operated with continuous flow through the tubes and the output modulated by varying the fluid temperature. The output of a small coil may be modulated by varying the fluid flow. In below freezing applications an antifreeze solution may be used as the heating fluid.



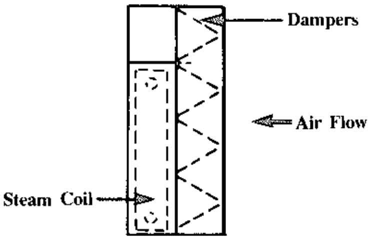
Multiple Tube Air Heating Coil
Figure 43



Serpentine Air Heating Coil
Figure 44



Steam Distributing Coil
Figure 45

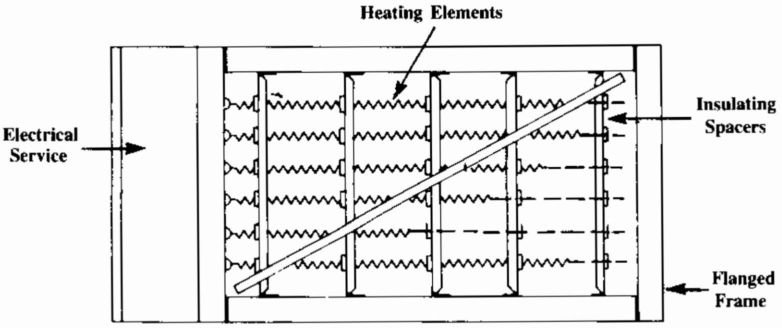


Steam Coil With Face And By-pass Dampers
Figure 46

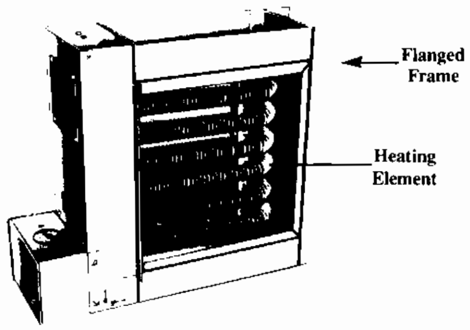
Electric air heating coils are available with bare resistance elements (Figure 47) or finned elements (Figure 48). To prevent overheating of the elements, electric coils must be protected by a high limit thermostat, and be electrically interlocked to prevent their operation when air flow is stopped. Output control may be achieved by sequenced switching of multiple elements, or by electronic switching of the power to a single element.

Refrigerant hot gas air heating coils may be selected for condensing or noncondensing of the refrigerant gas. Condensing coils are normally used in parallel with the refrigeration system condenser, and are limited to the system condensing temperature; usually 30 to 40°C.

Noncondensing coils are normally connected in series with the refrigeration system condenser, and reduce the temperature of the refrigerant gas without reaching its condensing temperature. The hot gas is usually available at 60 to 90°C.



Bare Element Electric Coil
Figure 47



Finned Element Electric Coil
Figure 48

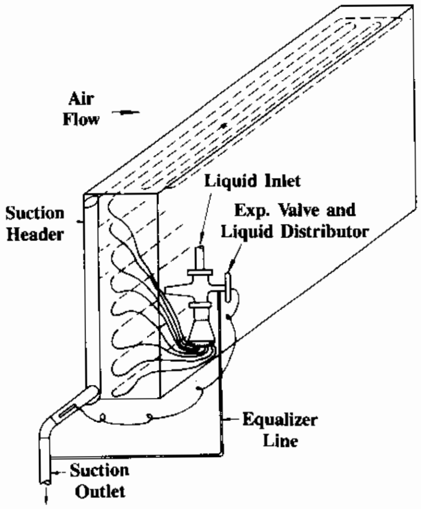
Air Cooling Coils

Air cooling coils are available for use with chilled water, and for direct refrigerant evaporation (direct expansion or "DX" coil).

Chilled water coils are similar to water heating coils except more tube rows are used to allow operation with a lower temperature difference between the water and the air. Chilled water coils are usually 4 to 8 tube rows deep, and are selected to operate with entering chilled water temperatures of 5 to 10°C and leaving air temperatures of 10 to 15°C.

A *direct expansion* coil (Figure 49) uses direct evaporation of a refrigerant to cool the air stream. The cooling effect may be modulated by the injection of refrigerant hot gas at the expansion valve.

Air cooling coils must be equipped with a means of draining water condensed from the air stream. At air velocities of over 2 m/s, mist eliminator baffles are placed downstream of the cooling coil to prevent water droplet carryover into fans or other downstream equipment.

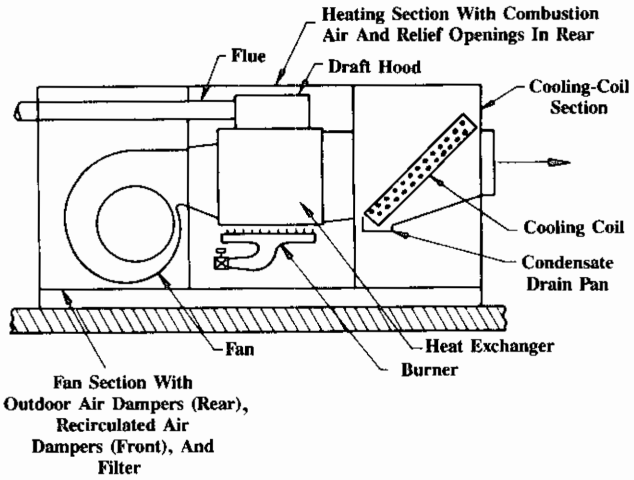


Direct Expansion Cooling Coil
Figure 49

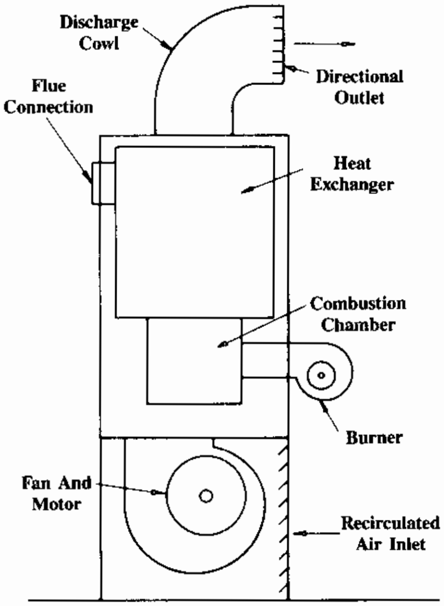
Courtesy of ASHRAE

Indirect Fired Heaters

Indirect fired heat exchangers using natural gas (Figure 50) are used on some HVAC systems. Oil fired heaters (Figure 51) are also available. Output is regulated by sequenced operation of multiple burner elements or by modulation of the fuel feed rate. Details of indirect fired heaters are provided in Process Furnaces, Dryers and Kilns, Module 7.



Gas Indirect Fired Heater
Figure 50



Oil Indirect Fired Heater
Figure 51

Courtesy of ASHRAE

Filters

Filters have the primary purpose of removing dust particles from the air stream. The common unit of measurement for dust particle size is the micrometre or "Micron".

$$1 \text{ Micron} = \frac{1}{1\ 000\ 000} \text{ m}$$

There are three basic methods of rating the efficiency of air filters.

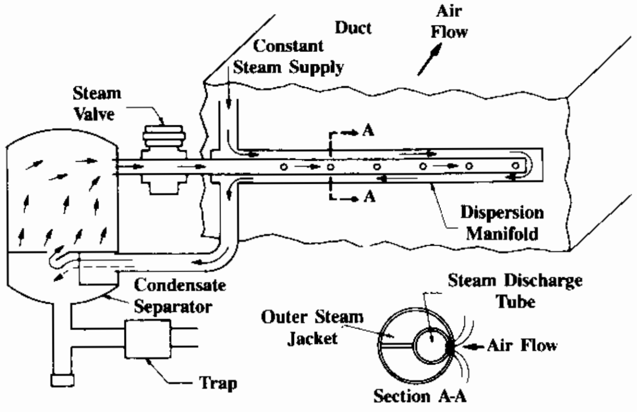
1. *Arrestance* is the weight fraction removed when a standardized synthetic dust, consisting of various particle sizes, is passed through the filter. This method is described under ASHRAE Test Standard 52-76 as "synthetic dust weight arrestance."
2. *Dust spot efficiency* is the discoloration effect of the outlet air compared to that of the incoming air when a standardized atmospheric dust is passed through the filter.
3. *Fractional efficiency* is the percentage removal of particles of a particular size.

The arrestance method is normally used for medium to low efficiency filters found in building air handling systems.

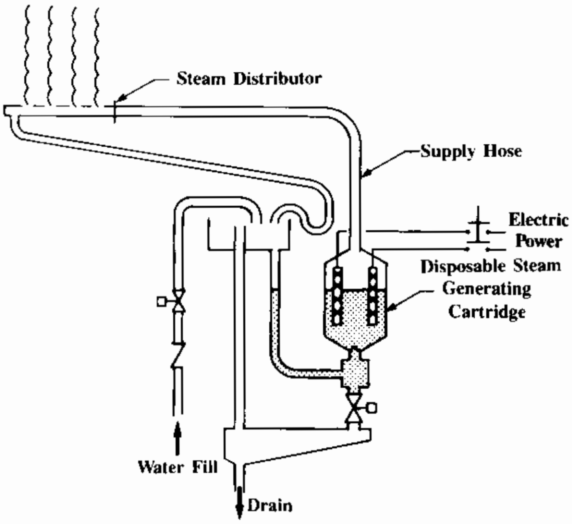
The dust spot efficiency method is used for high efficiency filters where removal of particles below 1 Micron is required.

The fractional efficiency method is used for applications involving the removal of a particular contaminant of known particle size. A special version of this method, used to test very high efficiency filters, is called the Thermal DOP method. In this method a smoke cloud of uniform 0.3 Micron droplets of an oily liquid, Di-Octyl Phthalate, is fed to the filter in a special test duct and the penetration to the outlet side of the filter is measured using photometers.

Table 6 lists common filter types and their characteristics. With the exception of electrostatic filters, the higher efficiency filters involve higher air pressure drop. A further description of certain types of filters will be found under Exhaust Air Treatment Devices.



Steam Injection Humidifier
Figure 52



Electric Steam Humidifier
Figure 53

Courtesy of ASHRAE

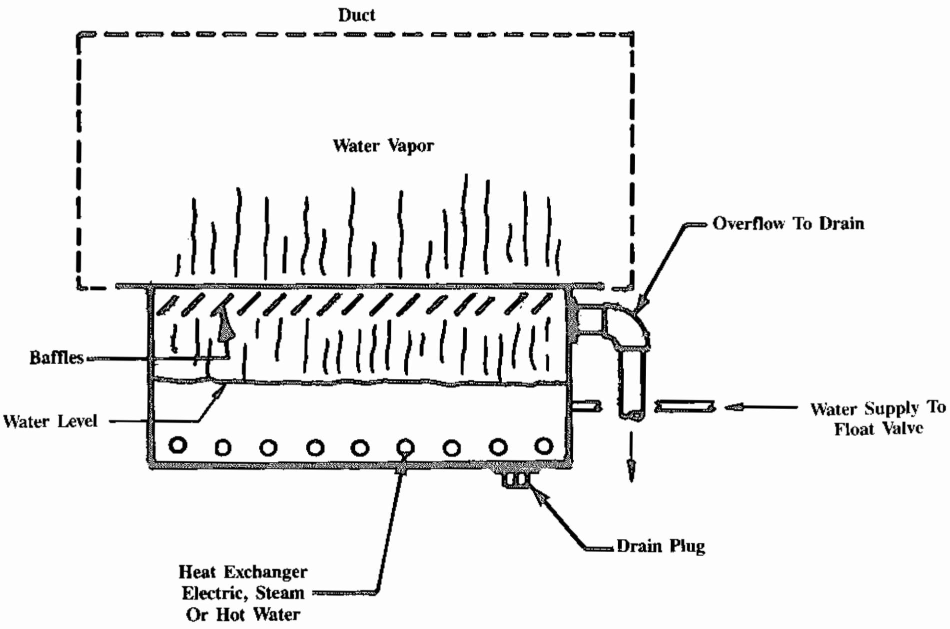
Humidifiers

Humidifiers inject water vapor into the supply air stream to increase the relative humidity of the conditioned space.

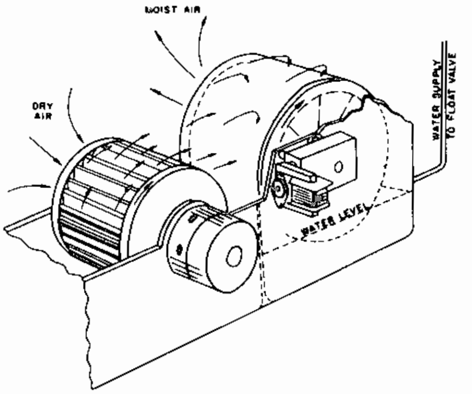
A *direct steam injection* humidifier (Figure 52) requires the least maintenance, but may inject more vapor than the air stream can absorb, resulting in condensation within the ducts. A self-contained electric steam humidifier (Figure 53) is available which uses a small electrode boiler with a disposable steam generating cartridge.

Heated pan humidifiers (Figure 54) may be heated by water, steam or electricity. Unless supplied with distilled water, the pan requires frequent cleaning to remove buildup of impurities from the water.

Wetted element humidifiers (Figure 55) are used in small systems. The wetted element material is usually designed to be easily replaced when it becomes coated with impurities from the water. The latent heat of evaporation must be supplied by the air resulting in a cooling effect on the air stream.



Heated Pan Humidifier
Figure 54

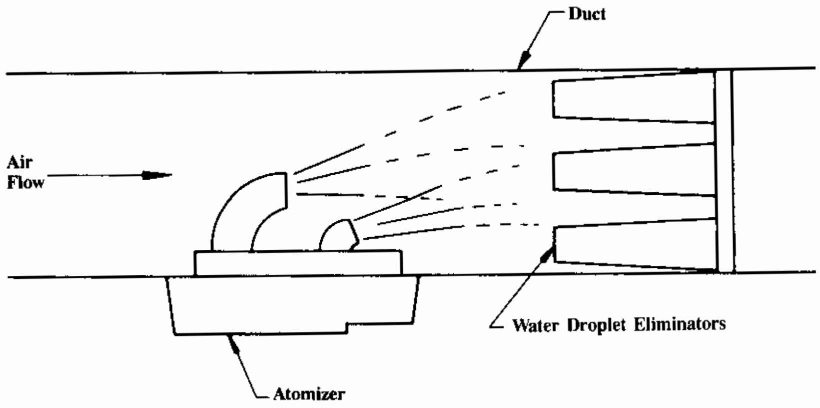


Wetted Drum Humidifier
Figure 55

Courtesy of ASHRAE

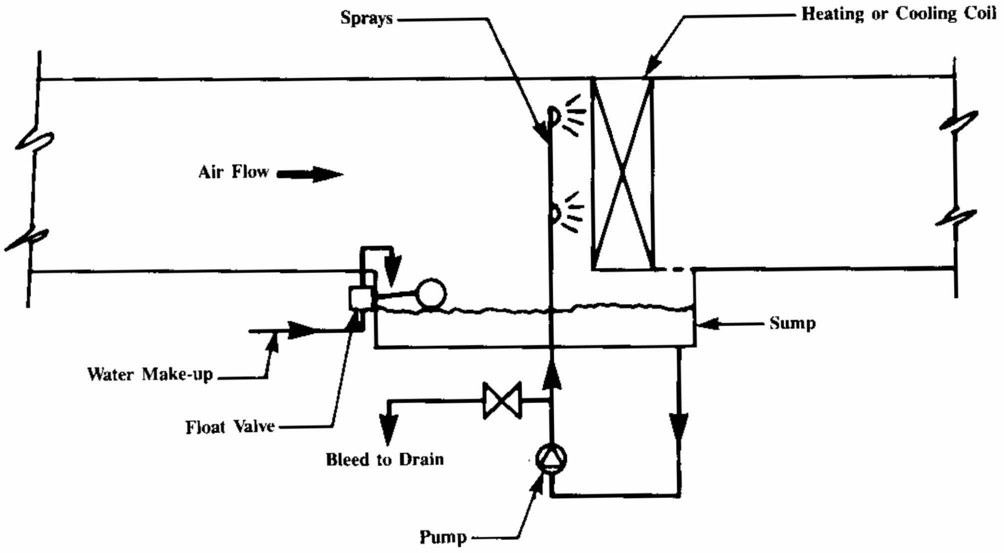
Direct water injection humidifiers are used both in very small systems (mechanical atomizers, Figure 56) and very large systems (sprayed coil systems, Figure 57). Mechanical atomizers inject water with impurities that may be present directly into the air stream, and may produce scale deposits in the ducts. Sprayed coil systems normally recirculate a portion of the water spray and control buildup of water impurities by bleeding a portion of the flow to drain. The latent heat of evaporation is supplied by the air stream. Under low summer humidity conditions, the sprayed coil type may be used for evaporative cooling to minimize the refrigerated cooling load.

Frequent cleaning and monitoring of the water condition is important for all humidifiers to ensure efficient operation and to avoid damage to other HVAC system components. Where public water supplies deliver water that contains high quantities of impurities, such as suspended solids and dissolved metals or minerals, it is often desirable to treat the humidifier water supply by softening or filtering.



Atomizing Humidifier
Figure 56

Courtesy of ASHRAE



Sprayed Coil Humidifier
Figure 57

Fans

Fans impart movement to the air stream by using the *inertia* of the air. The resulting air motion creates an inertia of motion, or *velocity pressure*, and a reaction, or *static pressure*, in the confining ducts. The duct arrangements at the inlet and outlet of a fan have a dramatic effect on the fan performance by the efficiency with which the velocity pressure is converted to useful static pressure.

Measurements of fan performance must take into account both static and velocity pressure, or *total pressure*, across the fan.

The air flow produced by a fan can be modulated by varying the fan speed, varying the angle of approach of the air stream with the fan blades, or throttling the air flow with a damper.

The fan speed may be varied by a variable speed motor or a variable speed drive between the motor and the fan. This method normally provides the most efficient use of fan energy.

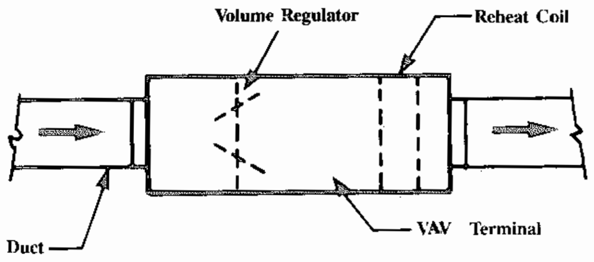
In an axial fan, the angle of approach of the air stream with the fan blades may be varied by rotating the blades. A similar effect may be achieved in a centrifugal fan by radial inlet dampers which impart a spin to the air stream.

Flow throttling dampers create an artificial back pressure to restrict the air flow. Throttling dampers on some types of centrifugal fans can be as efficient as radial inlet vanes for regulating fan power input.

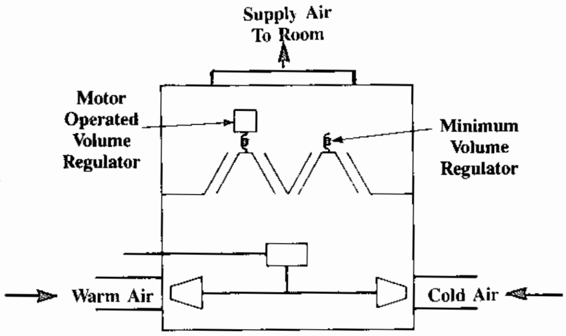
Fan types and their characteristics are described in Fans and Pumps, Module 13.

Variable Air Volume Devices

Variable air volume (VAV) devices are used to vary the air flow to individual building zones in accordance with the zone requirements. A VAV terminal reheat system uses a single inlet volume regulator (Figure 58), which operates in sequence with the reheat coil to reduce the zone air flow when reheating is required. A dual duct VAV terminal is illustrated in Figure 59. Typical control sequences are shown in Figure 60.

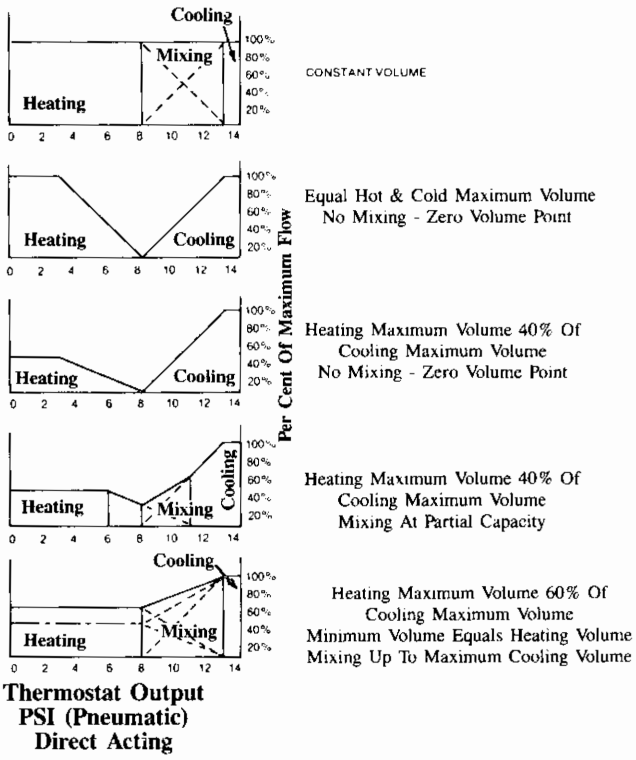


Single Duct VAV Terminal
Figure 58



Dual Duct VAV Terminal
Figure 59

Courtesy of ASHRAE



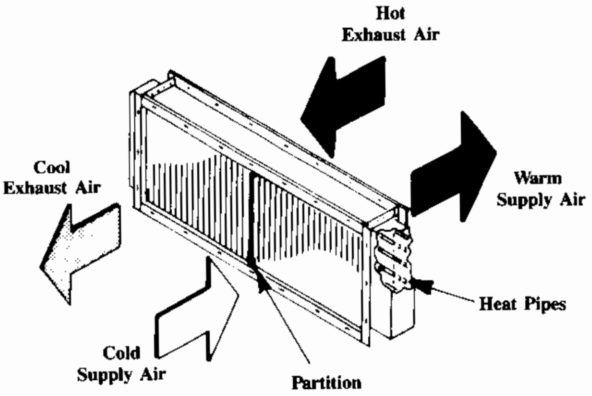
Typical VAV Control Sequences
Figure 60

Courtesy of ASHRAE

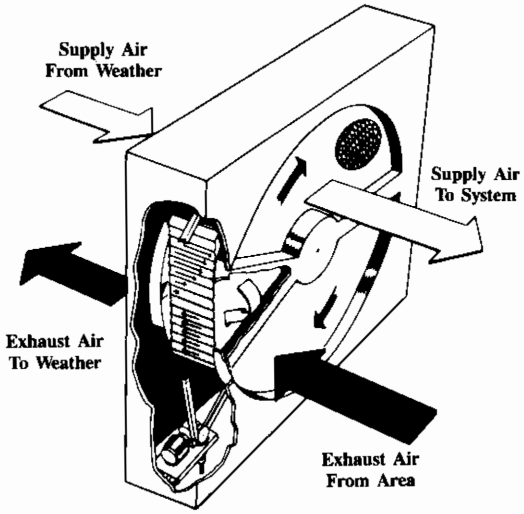
Air-to-Air Heat Exchangers

Air-to-air heat exchangers include heat pipes, rotating thermal element types (heat wheels), plate types and fluid run-around loops.

A *heat pipe* (Figure 61) utilizes the vapor phase of a working fluid to transfer heat from the warmer end of the pipe to the colder end. Heat pipe exchangers require little maintenance and can operate over a wide range of temperatures by the use of appropriate working fluids. The transfer rate is regulated by by-passing the cooler air stream. At subfreezing cold end temperatures, portions of the heat pipe in the warm air stream may drop to freezing temperature. Under this condition a portion of the cold air stream must be by-passed to prevent ice formation on the warm end.



Heat Pipe Exchanger
Figure 61



Rotating Thermal Element Heat Exchanger
Figure 62

Courtesy of ASHRAE

Rotating thermal element heat exchangers (Figure 62) absorb heat while passing through the warm air stream and give up the heat to the colder air stream. Some thermal wheel units contain absorbent materials which are capable of transferring both sensible heat and humidity (latent heat) between the air streams. Capacity is regulated by varying the speed of rotation. Where cross-flow from a contaminated to a clean air stream must be totally eliminated a purge section is provided.

Rotating element exchangers containing absorbents provide a high heat transfer efficiency, but consume a large amount of space and have a number of moving parts requiring maintenance. The air streams must be filtered to prevent fouling of the heat exchanger element.

A *plate type* heat exchanger (Figure 63) is the simplest and lowest cost type of air-to-air heat exchanger. Units with washdown provision are available for use in dirty air streams. For subfreezing applications a by-pass duct or preheat coil in the cold air stream may be required to prevent ice formation on the warm air side.

A *fluid loop* heat exchanger (Figure 64) uses an antifreeze solution circulated between water coils located in the air streams. A by-pass valve is usually provided for capacity control and protection against ice formation on the warm air coil. This system can be used on widely separately air streams, and may also be used to recover heat from multiple air streams.

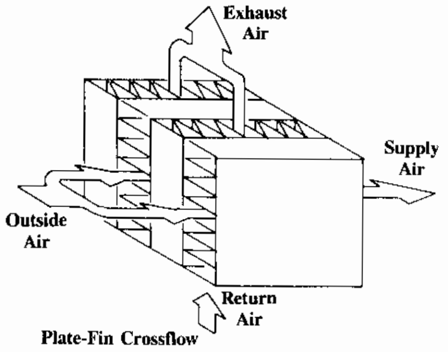
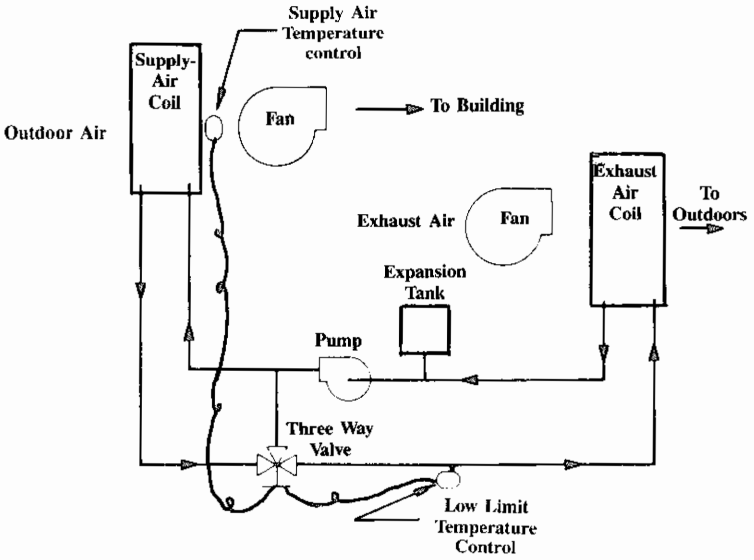


Plate Type Heat Exchange System
Figure 63



Fluid Loop Heat Exchanger
Figure 64

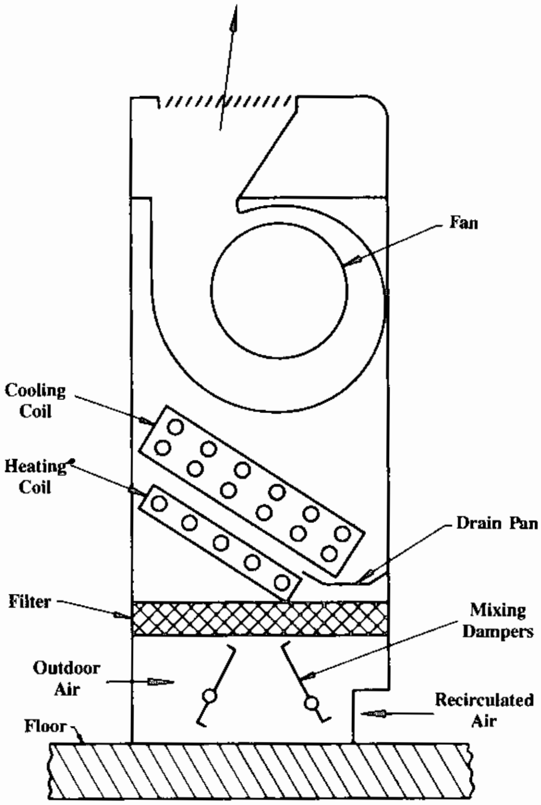
Courtesy of ASHRAE

Unitary Equipment

Fan-Coil Heating and Cooling Units

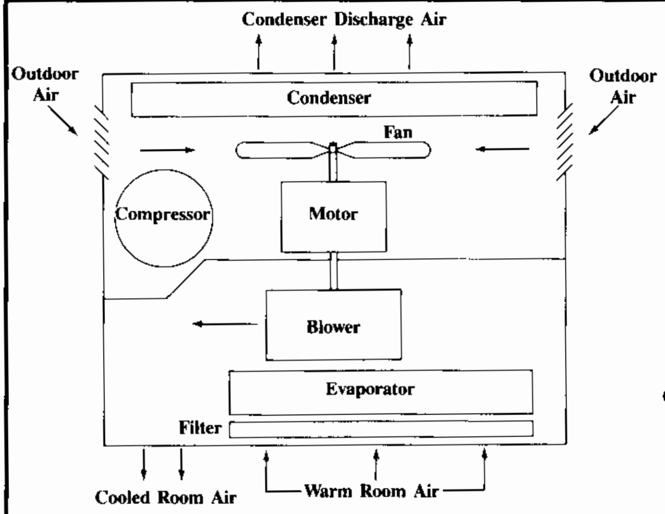
Fan-coil heating and cooling units (Figure 65), often called "unit ventilators," are available in various configurations. Separate coils may be provided for heating and cooling, and some units may have an outdoor air intake.

Fan-coil units provide an efficient system of heating and cooling, but require relatively frequent maintenance for reliable operation.

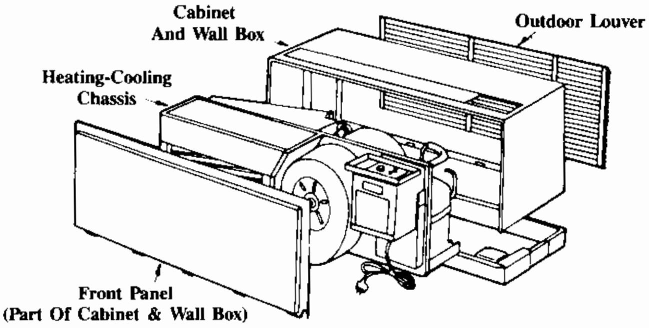


Fan-coil Heating And Cooling Unit
Figure 65

Courtesy of ASHRAE



Through-wall Air Conditioner
Figure 66



Through-wall Air Conditioner With Heating
Figure 67

Courtesy of ASHRAE

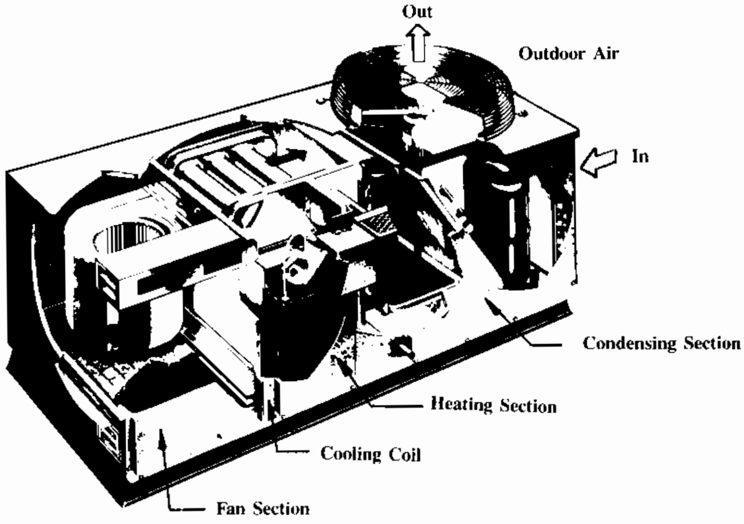
Unitary Air Conditioners

Unitary packaged air conditioners are available in various configurations.

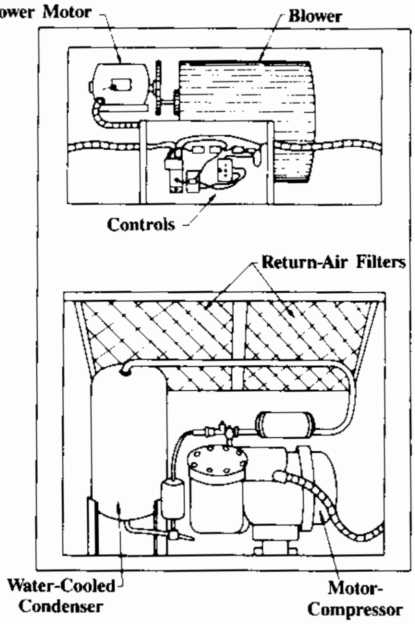
Through-wall units are available for cooling only (Figure 66), and for both heating and cooling (Figure 67). Units may have provision for an outdoor air intake. Through-wall air-conditioners are used in motels and hotels. They require frequent maintenance to ensure that outdoor air dampers and controls are properly adjusted for efficient operation.

Roof mounted packaged air-conditioning units (Figure 68) are available with a full control package including *economizer controls* for use of outdoor air for free cooling. They provide an efficient source of air-conditioning for a single zone such as a retail store. In winter, maintenance of roof mounted units may be difficult. Poor casing insulation and light weight dampers can be sources of heat loss.

Package indoor air-conditioning units (Figure 69) are used on single zone air systems for office buildings, and in local high heat gain areas such as computer rooms. Such units may use an integral water cooled condenser or a remote air cooled condenser.



Roof Mounted Packaged Air Conditioner
Figure 68



Indoor Packaged Air Conditioner
Figure 69

Induction Systems

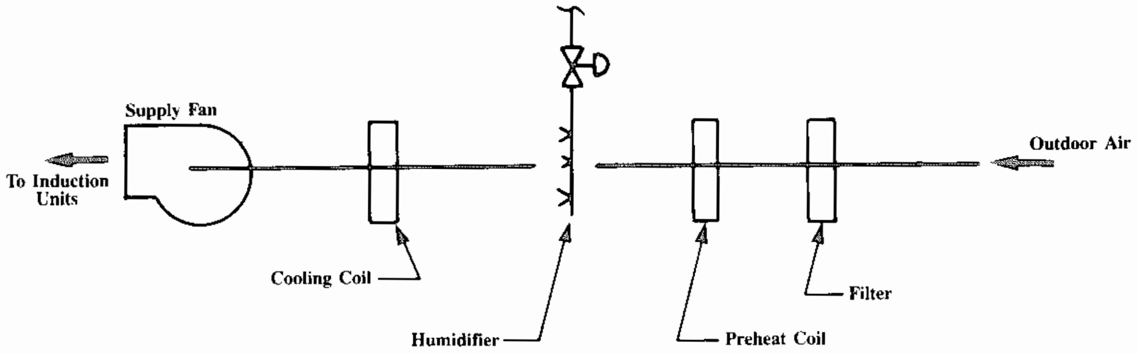
Induction Terminals

Induction terminals use a primary air supply flowing through a nozzle or group of nozzles to induce room air flow through the terminal heating and cooling coils. Single duct units (Figure 70) require both heating and cooling water for the unit coil. Dual duct units (Figure 71) may require only cooling by the unit coil with all heating supplied by the hot primary supply air.

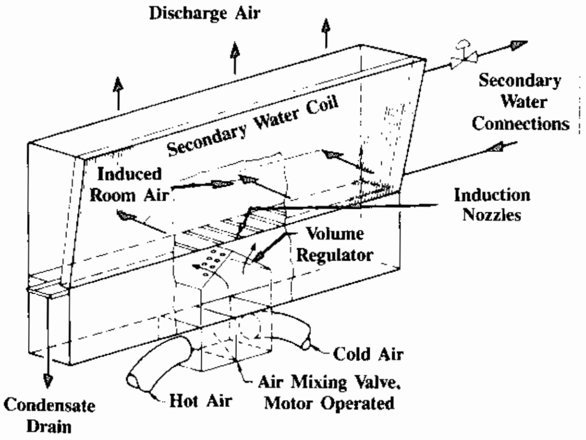
Induction units provide zone control similar to that of fan-coil units, but have less moving parts to maintain.

Induction System Supply Air Units

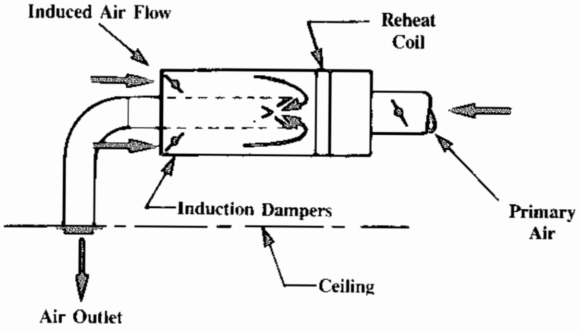
Supply air units for induction systems (Figure 72) often use 100 per cent outdoor air. The primary supply air may be 20 per cent of the air flow required for a normal central HVAC system serving the same space.



Single Duct Induction System, Central Plant
Figure 70



Dual Duct Induction Terminal
Figure 71



Induction System Supply Air Units
Figure 72

Courtesy of ASHRAE

Air Diffusion Devices

Grilles and Diffusers

Grilles and diffusers are used in several types and configurations.

Bar grilles (Figure 73) may have adjustable or fixed discharge pattern, and are used for window sill or wall mounting.

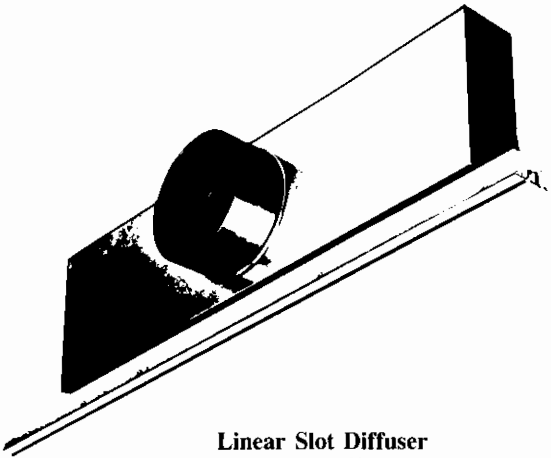
Linear slot diffusers (Figure 74) may have adjustable or fixed discharge pattern, and are usually ceiling mounted.

Multidirectional ceiling diffusers (Figure 75) are available in a wide variety of configurations for various applications. Their performance is affected by supply air temperature. When used in all-air heating and cooling systems, diffuser patterns may require seasonal adjustment for optimum comfort.

Laminar flow diffusers (Figure 76) are used for special applications such as hospital operating rooms and laboratory clean rooms where a uniform vertical flow of clean conditioned air is required.

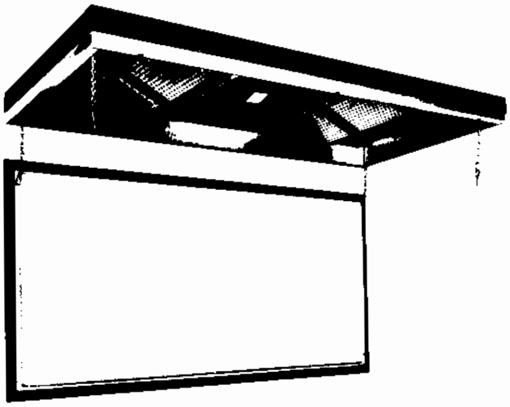
Jet-type diffusers (Figure 77) are used in high ceiling areas such as sports arenas where the conditioned air must be projected a long distance from the outlet.

Bar Grille
Figure 73

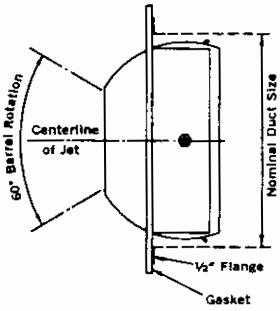


Linear Slot Diffuser
Figure 74

Multidirectional Ceiling Diffusers
Figure 75



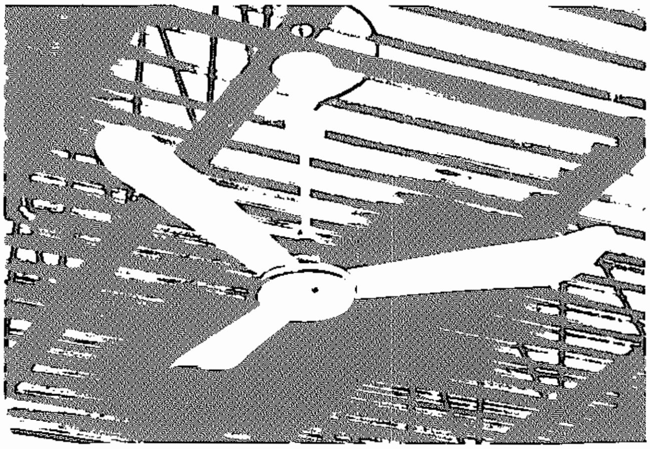
Laminar Flow Diffuser
Figure 76



Jet-type Diffuser
Figure 77

Destratification Devices

In a space with a high infiltration rate or high internal heat gain, stratification of warm air can occur near the ceiling. Down-flow fans (Figure 78) are used to induce mixing of the stratified air with air at the occupant level. Such units are available with thermostatic control to increase the fan speed in response to an increase in air temperature at the ceiling.



Down-flow Fan
Figure 78

Exhaust Systems

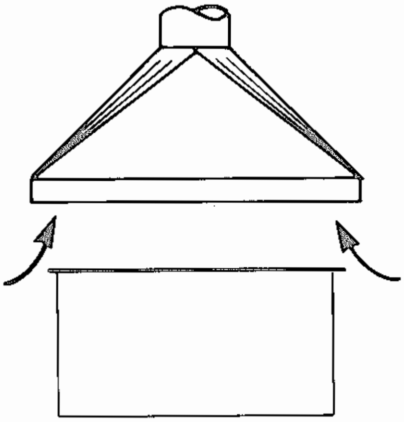
Containment Devices

Several techniques are used to contain and remove hot or contaminated air. The techniques all use controlled air movement to capture the offending air for exhaust or treatment. Table 7 lists typical air velocities required to capture various contaminants.

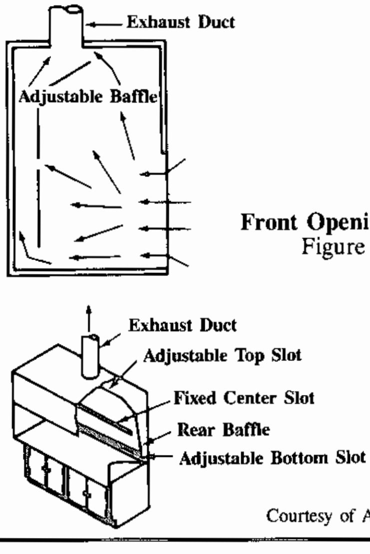
Dust or unpleasant odors may be *confined to a room which is kept at negative pressure* with respect to surrounding rooms by exhausting more air than is supplied. This approach is common for such areas as workshops and laboratories.

Canopy hoods (Figure 79) are effective in containing fumes from heat producing equipment such as cooking ranges or other hot process equipment. The natural upward draft from such equipment assists in capturing heat and fumes within the hood for removal by the exhaust system. The exhaust rate from a canopy hood is normally selected to produce a capture velocity of 0.40 to 0.75 m/s across the face area of the hood. Room air motion below the hood can seriously affect the capture efficiency.

Front opening hoods (Figure 80) are used in laboratories for handling materials which are chemically or biologically hazardous. Bench top units are most common but full-height walk-in units are also available. Front opening hoods provide a high containment efficiency with a relatively low exhaust rate owing to the restricted area of the front opening. Normal exhaust rates provide a face velocity of 0.50 to 0.75 m/s at the maximum sash opening. Hoods for handling biologically hazardous materials are often equipped with an integral recirculating air system using absolute filters.



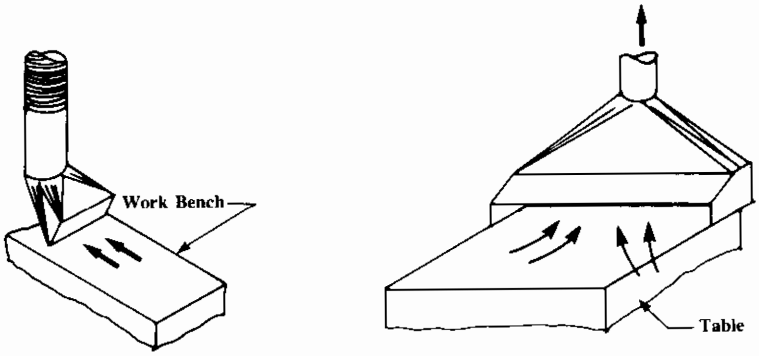
Canopy Hood
Figure 79



Front Opening Hood
Figure 80

Courtesy of ASHRAE

Proximity high velocity pickup exhaust devices (Figure 81) are used where space or function does not permit a hood enclosure. Proximity exhaust systems have a relatively high fan power consumption, and the high air velocity generates a significant noise level.



Proximity Exhaust Pickups
Figure 81

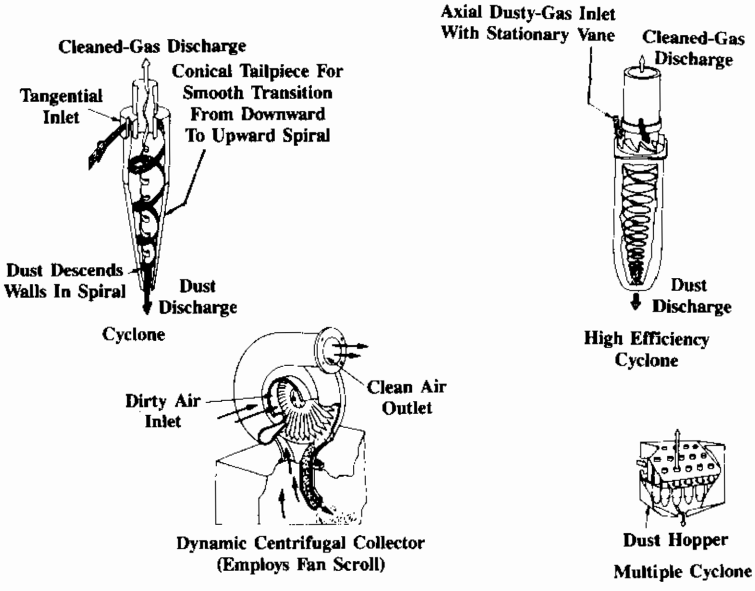
Exhaust Air Treatment Devices

Air treatment devices are used in some applications to allow recirculation and reuse of all, or a portion of a contaminated exhaust air stream. This section addresses only those treatment devices that are commonly used in such applications. Filters used in supply air systems are described in the Central Air Systems section of this module. Exhaust air treatment to meet outdoor air emission standards is beyond the scope of this module.

Most collectors and filters have the purpose of removing particulate matter from the air stream. As for supply air system filters, they are rated according to their efficiency in removing particles of various sizes.

Centrifugal collectors (Figure 82) are sometimes used in combination with a bag-type after-filter in applications such as metal-working shops and materials handling systems where heavy particles are encountered. *High efficiency cyclones* have a pressure drop of up to 2 kPa and remove over 70 per cent of particles above 5 Microns. A *dynamic centrifugal collector* is used to collect larger particles and to provide a motive force for the air stream.

Fabric filter units (Figure 83) are the commonly used particulate removal devices in industrial applications. The selection of appropriate components varies with each application, and requires expert analysis. Filter materials are available to achieve up to 99.9% removal of particles above 2 Microns. The air pressure drop through high efficiency bag filter units usually ranges from 0.5 to 1.5 kPa.



Cyclone

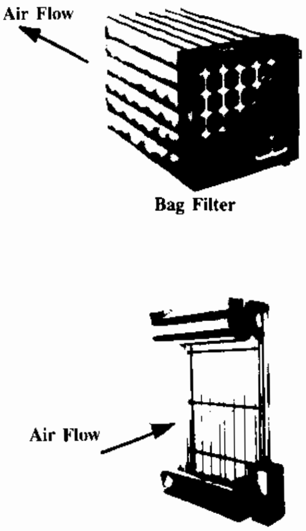
High Efficiency Cyclone

Dynamic Centrifugal Collector (Employs Fan Scroll)

Multiple Cyclone

Centrifugal Collectors
Figure 82

Courtesy of ASHRAE



Bag Filter

Roll Type Filter

Fabric Filters
Figure 83

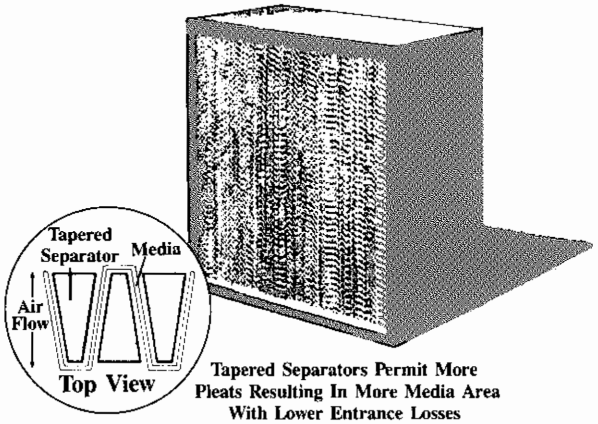
High efficiency extended surface filters (Figure 84), often referred to as "Absolute" or "High Efficiency Particulate Air" (HEPA) filters, are used in fume hood applications involving light particle loading. HEPA filters are available which provide virtually 100 per cent removal of particles over 1 Micron. Because of the cost of HEPA filters, lower efficiency prefilters may be used to extend the service life. The air flow resistance varies from 0.12 to 0.25 kPa over the service life of the filter.

Electronic air cleaners are used in packaged air handling systems to remove smoke and dust particles from the air. The ionizing plate type (Figure 85) is most commonly used. The high potential ionizing wires create a positive charge on the dust particles which causes the particles to be attracted to the grounded collector plates. The plates require periodic cleaning. Downstream panel filters are sometimes used to catch any flaking particles from overloaded collector plates, and upstream filters are sometimes used to eliminate larger particles. Electronic air cleaners alone have negligible air flow resistance. Associated panel filter resistance would be 0.035 to 0.065 kPa at normal duct velocities of 1.5 to 2.5 m/s.

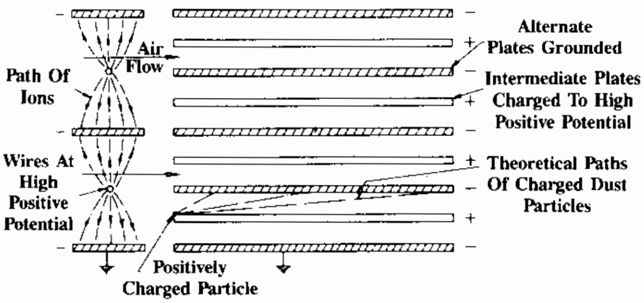
Electronic air cleaners are used to remove tobacco smoke and some types of industrial fumes to allow reduced use of outdoor air for ventilation. Typical applications include beverage rooms and welding shops. Specialized units are also capable of recovering materials for reuse such as cutting oils in machine shops.

Odor adsorbing filters may be used in the form of panel filters to allow recirculation in air systems serving rooms with objectionable odors such as gymnasiums and laboratories. Adsorbents are porous solids with large internal surface areas capable of capturing a large number of molecules. One kilogram of activated carbon, the most common adsorbent, is estimated to contain over one million square metres of internal surface area. Activated carbon will adsorb most common odor producing organic compounds. Adsorbents can normally be reactivated by heating in a furnace. In most cases, it is not economical for the user to purchase the necessary equipment, so the adsorbent media is replaced periodically and the used adsorbent returned to the supplier for reactivation. Prefilters are used to extend the service life of adsorbent filters.

Adsorbents other than activated carbon are used for special applications. Pellets of activated alumina impregnated with potassium permanganate are used to adsorb acrolein and formaldehyde. Carbon may be impregnated with chemicals to enhance its adsorption of specific gases.



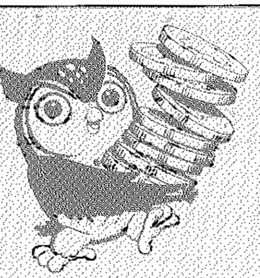
HEPA Filter
Figure 84



Electronic Air Cleaner
Figure 85

Courtesy of ASHRAE

ENERGY MANAGEMENT OPPORTUNITIES



Energy Management Opportunities is a term that represents the ways that energy can be used wisely to save money. A number of typical Energy Management Opportunities subdivided into Housekeeping, Low Cost, and Retrofit categories are outlined in this section with worked examples to illustrate the potential energy savings. This is not a complete listing of the opportunities available for heating, ventilating and air-conditioning systems. However, it is intended to provide ideas for management, operating and maintenance personnel to allow them to identify other opportunities that are applicable to a particular facility. Appropriate modules in this series should be considered for Energy Management Opportunities existing within other types of equipment and systems.

Calculations in the worked examples relating to outdoor conditions are approximations based on available degree day and monthly mean temperature data. The methods and equations presented may be used to develop more accurate analyses for particular locations and facilities. More detailed weather data, including mean daily and hourly temperatures, is available for many locations from regional offices of Environment Canada. With a little ingenuity the data can be manipulated to produce customized reference information for a particular building. Degree days above and below the calculated balance temperature of a building is a useful example of such customized information. Because of the complexity of accurately determining such factors as annual cooling energy, a detailed analysis on which to base retrofit decisions should be carried out by a competent professional.

The economics of each opportunity is rated by the *simple payback* in years. The equation is:

$$\text{Simple payback} = \frac{\text{Capital cost}}{\text{Annual cost saving}}$$

Housekeeping Opportunities

Implemented housekeeping opportunities are energy management actions that are done on a regular basis and never less than once a year. The following are typical Energy Management Opportunities in this category.

1. Adjust and tighten damper linkages, with particular attention to outdoor air dampers, multizone unit zone dampers and heating coil face and by-pass dampers.
2. Check and adjust motor drives on fans and pumps for belt tension and coupling alignment.
3. Replace air system filters to prevent restriction of air flow.
4. Shut off exhaust and make-up air systems to areas such as kitchens and laundries when the processes are not in operation.
5. Shut off lights and other heat producing equipment when not required.
6. Check and recalibrate control components such as room thermostats, air and water temperature controllers, and verify settings of time clocks.
7. Replace damaged or missing insulation on piping and duct systems.
8. Replace or repair crushed or leaking ducts in air systems.
9. Clean heat exchange surfaces, heating units and heating coils.
10. Consider rules on use of building space to permit reduction of outdoor air intake; for example, designated smoking and nonsmoking areas.
11. Establish minimum and maximum temperatures for heating and cooling during occupied and unoccupied periods, and readjust controls accordingly.
12. Adjust air flow rates to suit changing occupancy conditions and use of building space.
13. Implement a planned maintenance program to minimize component failures.
14. Promote an awareness of energy savings, and the means of achieving them, to all occupants from top management to the janitorial staff through seminars and active involvement.

Housekeeping Worked Examples

The worksheets can be used to estimate potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets. The examples are not actual case histories, but are considered typical of the conditions found in building HVAC systems.

1. Outdoor Air Damper Leakage

During a walk through audit of an office building in Edmonton it was observed that the outdoor air dampers on a 10 000 L/s air system would not fully close. With controls set for full recirculation, the mixed air stream temperature was measured to be 18°C, the outdoor air temperature was 0°C, and the return air temperature was 22°C. Using Worksheet 6-2, the outdoor air intake rate was determined to be 1818 L/s. The system operated continuously, but switched to full recirculation during unoccupied periods totalling 128 hours per week. Worksheet 6-1 was used to determine that the cost of heating the outdoor air leakage during unoccupied periods was $7,240 per year.

Adjusting the damper linkage to fully close the dampers reduced the outdoor air intake to 909 L/s.

Outdoor air rate reduction = 1818 - 909 = 909 L/s

$$\text{Cost Saving} = \frac{\$7{,}240 \times 909}{1818} = \$3{,}621/\text{yr}$$

2. Maintain Motor Drives

Improper alignment of belts and couplings on pump and fan drives can consume up to 10 per cent of the input power. Pump and fan drive efficiencies are discussed in Fans and Pumps, module 13.

3. Filter Replacement

The filters on a 7500 L/s variable air volume system were observed to be very dirty. A decrease in pressure drop of 0.200 kPa was measured after the installation of new filters. The air system operated 24 hours per day, 7 days per week. Worksheet 10-3 was used to estimate a fan energy cost saving of $876 per year by maintaining clean filters.

4. Exhaust System Shutdown

During a walk through audit of a university cafeteria located in Windsor, Ontario it was observed that the kitchen hood exhaust was left operating for 12 hours per day, although the cooking equipment was only operated for 8 hours per day. The kitchen hood exhausted 3800 L/s and was electrically interlocked with a 3800 L/s outdoor air make-up system. By operating the exhaust and make-up air systems only when required (8 hours per day) the reduction in operating hours was (12-8) x 7 = 28 hours per week.

Worksheet 10-1 was used to estimate an annual cost saving of $2,266.

5. Shut Off Lights

During a walk through audit of an air conditioned building in Winnipeg, Manitoba it was observed that the lights were switched on during the occupied hours in a section of the building not being used. The building was occupied for 84 hours per week.

Using Worksheet 10-6 the cooling energy cost saved by turning off the lights was estimated to be $6 3 per year.

6. Recalibrate Control Components

During a walk through audit of an office building in Fredericton, N.B. a terminal reheat system with economizer control was observed to be operating with the supply air temperature 3°C less than the design value.

It was recognized that, during the heating season, the system was taking in additional outdoor air to achieve the lower supply air temperature and creating added reheat load.

The total system air flow rate was 7500 L/s and the system operated 24 hours per day, 7 days per week.

Using Worksheet 10-9 it was estimated that an annual energy cost saving of $9,840 could be achieved by resetting the supply air temperature to the design value.

7. Pipe and Duct Insulation

Damaged or missing pipe and duct insulation will increase the heat loss or gain by the air or water distribution system and decrease the system efficiency. Insulation is discussed in Process Insulation, Module 1 and Steam and Condensate Systems, Module 8.

8. Duct Leakage

Duct leakage can be caused by physical damage or by poor quality installation and sealing. If a significant quantity of air leaks from the duct system, the design conditions of the space may not be met and the fan air flow may have to be increased. This requires additional fan energy due to the increased air flow and the resulting greater friction loss in the ducts. For example, an air handling system with 10 per cent leakage requires an increase in fan power of approximately 20 per cent to meet design air flow to the conditioned space.

For a duct system improperly installed and sealed, the leakage rate may be greater than 15 per cent.

9. Clean Heat Exchange Surfaces

Clean heat exchange surfaces will keep the heat transfer rate at or near the design condition. When heat exchanger surfaces become dirty or corroded the heat transfer rate is lowered, and a higher temperature is required in the heating medium to produce the desired output. A higher heating medium temperature increases the distribution system heat loss, and may reduce the efficiency of the heat source owing to the required higher generation temperature.

10. Implementation of Designated Smoking Areas

During a walk through audit of an office building in Sudbury, Ontario it was observed that smoking was allowed throughout the building. To maintain aceptable indoor environment the air system was operated with a minimum outdoor air intake of 1500 L/s during occupied periods. The building was occupied 70 hours per week. By designating smoking areas from which all return air was exhausted, it was determined that the minimum outdoor air intake could be reduced by 375 L/s. The indoor design temperature was 22°C in winter and 24°C in summer.

Worksheet No. 10-1 was used to estimate an annual cost saving of $1,372 per year for heating. Worksheet 10-5 is used to estimate an annual cost saving of $20 for cooling.

11. Thermostat Settings

The control of building space conditions to a constant temperature for all seasons is usually neither necessary nor desirable. For normal office occupancy, temperatures of 20°C in winter and 25°C in summer are considered acceptable.

An office building in Fredericton, N. B. had been operated all year at an interior space temperature of 22°C. The annual heating system fuel consumption was 70 000 litres of number 2 fuel oil.

It was decided to reset the space thermostats to 20°C during the heating season. Worksheet 10-12 was used to estimate an annual fuel cost saving of $3,599, based on a unit fuel cost of $0.40 per litre and 24 hours per day operation.

Increasing the thermostat settings to 25°C during the summer will also significantly reduce the cooling energy cost. The higher summer temperature is considered acceptable for most office activity and provides a lower physiological shock to persons entering and leaving the building.

12. Air Flow Rates

Acceptable air flow rates vary with occupancy, activity and contaminant levels in the space. If, for example, a building area previously designated for offices is changed to storage or unoccupied space, the air flow could be reduced. Operating and maintenance staff should be alert to every opportunity to save energy dollars by reducing air system flow rates.

13. Planned Maintenance Program

An effective planned maintenance program, involving a full inventory of equipment and systems, logged information on component service life and scheduled servicing of equipment, will achieve energy cost savings through improved system efficiency.

14. Energy Conservation Seminars

Energy conservation seminars should be conducted for all employees from top management to the janitorial staff. Knowledge of the energy consumption of certain equipment or processes will provide an awareness of possible savings which can be achieved by limiting their use. For example, if a kitchen exhaust fan is operated for 24 hours a day but the kitchen is only occupied for 8 hours a day, a knowledge of the energy being consumed would provide incentive for turning the fan off.

Goals of energy conservation savings should be set which motivate all employees to participate in the objectives. It has been demonstrated that energy cost savings of 10 per cent to 20 per cent can be achieved by employee actions.

Low Cost Opportunities

Implemented low cost opportunities are energy management actions that are done once and for which the cost is not considered great. The following are typical Energy Management Opportunities in this category.

1. Install time clocks to shut down air systems or switch to 100 per cent recirculation when the space served is unoccupied.
2. Install control interlocks to shut down heating or cooling system pumps when no output is required.
3. Install "economizer" controls on central air handling systems to use outdoor air to replace refrigerated cooling when appropriate.
4. Install programmable thermostats to automatically reduce the controlled space temperature during unoccupied periods of the heating season.
5. Install zone thermostat controls on perimeter heaters which use constant or scheduled water temperature.
6. Install edge seals to reduce leakage at outdoor air and relief air dampers.
7. Add automatic control valves at unit heaters and fan-coil heaters to shut off the flow of water or steam when the fans are not running.
8. Interconnect the controls for spaces with separate heating and cooling systems to prevent simultaneous heating and cooling.
9. Install load analyzers in the controls of multizone and dual duct systems to optimize hot and cold deck temperatures.
10. Install load analyzers in the controls of terminal reheat systems to optimize the supply air temperature and minimize the reheat load.
11. Reduce outdoor air intake rate to the minimum required by local code, or the minimum required for make-up of the building exhaust, whichever is greater.
12. Install barriers or walls around heat producing equipment to reduce the direct heat gain to the occupied space.
13. Install destratification fans in high ceiling areas.
14. Reduce pressure drops, where possible, in air and water circulation systems by using lower pressure drop filters and strainers, and removing high pressure loss fittings or obstructions. Adjust fan speeds and trim pump impellers to achieve the resulting savings in pumping energy.

Low Cost Worked Examples

The worksheets can be used to estimate potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets. The examples are not actual case histories, but are considered typical of the conditions found in building HVAC systems.

1. Air System Shut-Down

An air handling unit delivers 20 000 L/s of air at 1.2 kPa total pressure. The building is occupied 70 hours per week. During the remaining unoccupied periods of 98 hours per week, the air system is operated on full recirculation. It was determined that the air unit could be safely shut down during unoccupied periods. The fan energy saving was determined with the aid of Worksheet 10-3, Fan Energy Cost.

The estimated cost saving is $8,154 per year.

The estimated cost of installing a time clock to automatically shut down the unit during unoccupied periods is $400.

$$\text{Simple payback} = \frac{\$400}{\$8{,}154} = 0.05 \text{ years (18 days)}$$

2. Pump Shutdown

A heating water circulating pump serving a heating coil in the air handling unit of Example 1 delivers 20 L/s at a total pump pressure of 150 kPa. The cost saving if the pump is shut down with the air handling unit was determined with the aid of Worksheet 10-4, Pump Energy Cost.

The estimated cost saving is $1,529 per year.

The estimated capital cost to connect to the air handling unit time clock is $100.

$$\text{Simple payback} = \frac{\$100}{\$1,529} = 0.07 \text{ years (24 days)}$$

3. Economizer Controls

In Fredericton, N.B. an interior room containing a computer facility was served by a 100 per cent recirculating air conditioning system. The room was continuously occupied by 3 persons and contained 30 light fixtures of 90 watts each. The average power consumption of the computer equipment was estimated at 55 000 watts based on the computer manufacturer's data.

It was recognized that the installation of outdoor air intake and relief ducts, and an *economizer control* system would allow outdoor air to be used for *free cooling* during a portion of the year.

From Environment Canada weather data for Fredericton it was identified that the mean monthly temperature is above 18°C for 2 months per year. Using Worksheet 10-6 it was estimated that a cooling energy cost saving of $6,113 per year would be achieved by cooling with outdoor air for the remaining 10 months per year.

The estimated cost of ductwork and economizer controls is $7,500.

$$\text{Simple payback} = \frac{\$7,500}{\$6,113} = 1.2 \text{ years}$$

4. Night Setback

An electrically heated building in Fredericton, N. B. was maintained at a constant temperature of 22°C during the heating season.

It was recognized that programmable thermostats could be installed to set the space temperature back to 15°C during an average 12 hours per day. Using Worksheet 10-12 the reduction in heating energy cost was estimated to be $4,265 per year.

The estimated capital expenditure for installation of the thermostats is $7,000.

$$\text{Simple payback} = \frac{\$7,000}{\$4,265} = 1.6 \text{ years}$$

5. Perimeter Heater Zone Controls

An elementary school located in Montreal was heated to a constant temperature of 22°C by perimeter heaters. Several classrooms not being used were kept at the same temperature as the rest of the building because the classrooms did not have separate thermostat controls. Using Worksheet 10-7 the initial heating fuel consumption for the classrooms was estimated to be 5641 litres per year, based on double insulated glass and a masonry cavity wall with 50 mm polystyrene foam insulation. With the addition of zone control valves and individual thermostats the room temperatures of the unused classrooms could be reduced to 13°C. Using Worksheet 10-12 the reduced annual heating cost was estimated to be $797 per year.

The estimated cost of the zone thermostat controls is $1,400.

$$\text{Simple payback} = \frac{\$1,400}{\$797} = 1.8 \text{ years}$$

6. Damper Seals

Example 1 of Housekeeping Opportunities illustrated reduction of unwanted outdoor air intake by adjustment of damper linkages. This action resulted in a remaining damper air leakage of 909 L/s. It was determined that a further reduction could be achieved by the installation of edge seals on the damper. The seal manufacturer claimed the air leakage could be reduced to 3 percent of the design maximum outdoor air flow at the existing system pressures. The design maximum air flow with the dampers open was 10,000 L/s.

$$\text{Air leakage with edge seals} = 0.03 \times 10\ 000$$

$$= 300 \text{ L/s}$$

$$\text{Outdoor air rate reduction} = 909 - 300 = 609 \text{ L/s}$$

Using Worksheet 10-1 the cost saving was estimated to be $2,425 per year.
The estimated cost of installing the edge seals is $800.

$$\text{Simple payback} = \frac{\$800}{\$2{,}458} = 0.33 \text{ years (4 months)}$$

7. Shut Off Heating Media Flows

A wall mounted hot water fan-coil unit in an entryway of a Toronto office building was controlled by a thermostat which started the fan when the temperature dropped. When the fan was off, hot water continued to flow through the coil resulting in overheating of the entrance area during the summer. It was recognized that the condition could be corrected by installing a control valve on the hot water supply line to stop the water flow when the fan was off.

Using Worksheet 10-11 the annual fuel cost saving was estimated to be $272.
The estimated cost of installing a control valve and wiring to the thermostat is $250.

$$\text{Simple payback} = \frac{\$250}{\$272} = 0.9 \text{ years (11 months)}$$

8. Interlock Heating and Cooling Controls

A retail establishment in Edmonton, Alberta, with an exposed wall and large windows was heated by a perimeter hot water convector controlled by an outdoor air sensing thermostatic controller. The hot water was provided by a gas fired boiler. The circulating water temperature was varied inversely with the outdoor temperature and circulation was stopped when the outdoor temperature exceeded 18°C. The radiation system was sized to maintain comfort conditions with the windows fully exposed and the building unoccupied. The tenant was not billed for heat but owned and paid for the operation of an air conditioning unit which was controlled by a cooling thermostat. The settings on the hot water controls indicated that the minimum water temperature of 42°C occurred when the outdoor temperature was above 13°C. During the occupied period it was observed that the air conditioner started when the outdoor temperature rose above 13°C.

Since the perimeter heating did not shut off until 18°C outdoor temperature, the heat output was creating additional air conditioner load at outdoor temperatures between 13 and 18°C. From manufacturers data the convector heat output at 42°C water temperature was estimated to be 15,000 kJ/h.

From Environment Canada temperature frequency data the temperature is between 13 and 18°C for 2200 hours per year. The retail establishment was occupied for 72 hours per week. The occupied hours per year during which the outdoor temperature is betwen 13 and 18°C can therefore be calculated.

$$\text{Hours} = \frac{2200 \times 72}{24 \times 7} = 943$$

It was identified that the condition could be corrected by the installation of a control valve on the convector water supply and a heating-cooling room thermostat to control the heating convector and air conditioner in sequence.

$$\text{Annual heating reduction} = \frac{943 \times 15000}{1000} = 14\ 145 \text{ MJ}$$

For natural gas at $0.21/m^3, 37.2 MJ/m^3 (Appendix C) and 75 per cent boiler efficiency:

$$\text{Heating cost saving} = \frac{14\ 145 \times 0.21}{37.2 \times 0.75} = \$106/\text{yr}$$

For an air conditioning unit requiring 80 kWh/GJ of cooling and an electric energy cost of $0.05/Kwh:

$$\text{Cooling energy cost saving} = \frac{14\ 145 \times 80 \times 0.05}{1000} = \$57/\text{yr}$$

Total energy cost saving = $106 + $57 = $163/yr

The estimated cost of the control valve and thermostat, to be shared between the landlord and the tenant, is $480.

$$\text{Simple payback} = \frac{\$480}{\$163} = 2.9 \text{ years}$$

9. Terminal Reheat System Load Analyzers

An office building in Vancouver, B.C. has a 20 000 L/s terminal reheat supply air system. The air system operated for 12 hours per day, 5 days per week. The supply air temperature was set at a constant 13°C with the reheat coils supplying the additional heat for zone comfort. It was observed on several occasions that approximately 90 per cent of the reheat coils were in operation. This indicated that the supply air temperature was too low. By installing load analyzers and a reset controller the supply air temperature would automatically be reset to temperatures required by zone space conditions. An average supply air reset temperature of 16°C was determined by trial and error at various outside temperatures.

Using Worksheet 10-10 the annual cost saving was estimated to be $5,551.

The estimated cost of installing load analyzers in the controls of the terminal reheat system is $7,500.

$$\text{Simple payback} = \frac{\$7,500}{\$5,551} = 1.4 \text{ years}$$

10. Dual Duct System Load Analyzers

The effect of load analyzers on multizone and dual duct systems is more complex, but for approximation purposes the same procedure can be used as for the terminal reheat system of Low Cost Worked Example 9. The hot deck function is similar to that of a reheat coil on a terminal reheat system.

11. Reduction of Outdoor Air Requirements

The air handling system in a building in Saskatoon was operating with a minimum intake of 10 000 L/s of outdoor air for ventilation and make-up for building exhaust systems. It was determined from a review of local codes and the building occupancy that the total air required for ventilation is 7000 L/s. The building exhaust rate was 6000 L/s. A total outdoor air requirement of 7000 L/s would therefore be adequate, representing a reduction of 3000 L/s.

The annual heating energy cost saved was determined with the aid of Worksheet 10-1 to be $15,786.

The estimated cost to reset and balance the air system to accomplish the reduced outdoor rate is $1,500.

$$\text{Simple payback} = \frac{\$1,500}{\$15,786} = 0.1 \text{ years (35 days)}$$

12. Reduction of Internal Heat Gains

An interior retail space within an airport facility has a continuous cooling load. The space is occupied 24 hours per day, 365 days per year and is maintained at 20°C. It was determined that various pieces of equipment represented significant heat gains to the space. The continuous connected power load of the equipment is 8000 watts.

Sensible heat gain from equipment = 8000 x 3.6 = 28 800 kJ/h

It was decided that cooling energy savings could be achieved if the equipment was enclosed within an insulated wall to separate it from the occupied space. A thermostatically controlled fan was installed with ducts to bring outdoor air directly into the confined equipment space to maintain a temperature of 32°C at 26°C outdoor air temperature.

The required air flow can be calculated from the equation for sensible heat exchange with air:

$$Qs = fa \times (T1 - T2) \times 4.345$$

Converting this equation to determine the air flow:

$$fa = \frac{Qs}{(T1 - T2) \times 4.345} = \frac{28{,}800}{(32 - 26) \times 4.345} = 1105 \text{ L/s}$$

The remaining heat gain from the equipment space to the occupied area was estimated with the aid of Worksheet 10-7 to be 6480 kJ/h.

Reduction of heat gain to the occupied space = 28 800 - 6 480 = 22 320 kJ/h

Worksheet 10-6 was used to estimate an annual cooling cost saving of $782.

From air pressure drop through the ducts the fan total static pressure was estimated to be 0.1 kPa (gauge). Worksheet 10-3 was used to estimate an annual fan energy cost of $66.

The net annual saving = $782 - $66 = $716

The estimated capital cost to construct the enclosure around the equipment and install the fan is $3,000.

$$\text{Simple payback} = \frac{\$3{,}000}{\$716} = 4.2 \text{ years}$$

13. Destratification Fans

During a walk through audit of a large open warehouse in Fredericton, it was noted that the air temperature near the underside of the roof was 35°C although the temperature near the floor was 20°C.

To reduce the heat loss through the roof and improve the heating system performance, the addition of destratification fans was proposed. Propeller fans would be installed near the underside of the roof structure to transfer the warm air to the lower working areas and create a uniform 20°C space temperature.

Using Worksheet 10-8, the annual energy cost saving was estimated to be $48,918.

The estimated cost to install the fans is $30,000.

$$\text{Simple payback} = \frac{\$30{,}000}{\$48{,}918} = 0.6 \text{ years (7 months)}$$

14. Reduce Pressure Drops for Savings

An audit of the devices which contributed to pressure drop in an 11 980 L/s air circulation system identified opportunities to reduce the total pressure drop through the system by several actions.

Action	**Reduction**
Increase roll filter cycles to expose cleaner filter media	17 Pa
Modify exhaust louvres to airfoil type with powered operator	9
Place baffles in fan cabinets to reduce inertia loss	7
Add turning vanes to reduce elbow loss	14
Replace exhaust louvre fly screen with a more open bird screen	2
	49 pa

Using Worksheet 10-3 the fan energy cost saving was estimated to be \$343 per year.

The estimated implementation cost is \$114 for materials, and the estimated increased cost of roll filters is \$80 per year.

Net cost saving = \$343 - \$80 = \$263/yr

$$\text{Simple payback} = \frac{\$114}{\$263} = 0.4 \text{ years (5 months)}$$

Retrofit Opportunities

Implemented retrofit opportunities are energy management actions which are done once and for which the cost is significant. Many of the opportunities in this category will require detailed analysis by specialists, and therefore cannot be covered effectively in this module. Worked examples are provided for some of the listed Energy Management Opportunities, while in other cases there is only commentary. The following are typical Energy Management Opportunities in the retrofit category.

1. Install heat recovery systems to extract heat from exhaust air and preheat outdoor make-up air. Examples of such systems include glycol loops, air-to-air heat exchangers, or heat pipe energy transfer units.
2. Install local recirculating air treatment units, such as electronic air cleaners or activated charcoal odour adsorbing filters, to allow a reduction in the amount of outdoor air required for ventilation.
3. Install air treatment equipment on exhaust air streams to allow all, or a portion of, the air stream to be recirculated. High efficiency filters and activated charcoal filters can be used on kitchen hood exhausts to allow recirculation of up to 75 per cent of the air stream. Bag filters and centrifugal dust arrestors can be used on certain workshop and plant exhaust systems to allow up to 100 per cent recirculation of the air stream.
4. Reduce building air flow rates by moving conditioned air from spaces requiring high quality environment through spaces where a lower quality environment is acceptable. Examples include flowing return air from offices through storage spaces, and drawing exhaust make-up air for toilets through shower and locker areas.
5. Install a separate air system where one area in a building has a unique requirement affecting the operation of a large central system. Examples might be areas with a different occupancy schedule or high internal heat gain, such as a computer room or an assembly theatre.
6. Add variable air volume controls to a constant volume terminal reheat system.
7. Install additional insulation on piping systems.
8. Install additional insulation in ducts located outside the space being served by the air system.
9. Install dual condenser heat recovery chillers and use the condenser heat in the hot deck of multizone and dual duct systems, in perimeter heating systems, or for zone reheat in terminal reheat and induction systems.
10. Extend the utilization of heat recovery chillers to 12 month operation by incorporating heat recovery cooling loads from building ventilation and process heat sources. The heat sources might include (a) cold storage or small process refrigeration condenser heat, (b) cooling coils in exhaust air streams, (c) equipment room cooling and (d) operation of an air handling system at minimum outdoor air intake to provide maximum recirculation of warm return air through the cooling coil.
11. Install carbon monoxide detector to automatically control parking garage ventilation systems.
12. Install a microprocessor energy management system to monitor and integrate the control functions of the building energy systems. The management functions could include (a) scheduling and optimization of system start-stop times, (b) enthalpy-based control of economizer cycle, (c) automatic reset of air and water supply temperatures to suit the heating and cooling load requirements, (d) monitoring of energy consumption by various building systems to identify increasing consumption trends and allow corrective action, (e) monitoring of electrical energy consumption and control of interruptible loads to limit peak demand, and (f) monitoring of the operating and maintenance status of HVAC systems for quick response to equipment malfunctions.
13. Install water sprays on large roof areas to reduce summer cooling loads.

Retrofit Worked Examples

The worksheets can be used to estimate the potential cost savings. The following examples, numbered to correspond with the previously listed opportunities, illustrate the use of the worksheets. The examples are not actual case histories, but are considered typical of the conditions found in building HVAC systems.

1. Heat Recovery From Exhaust air

A hospital operating suite in Sudbury, Ontario is served by an air system circulating 12 000 L/s of 100 per cent outside air. The system operates continuously and all return air is exhausted by a separate fan. The supply air is preheated to 13°C by steam from a boiler plant using number 6 fuel oil. The operating rooms are maintained at 22°C and 50 per cent relative humidity.

A glycol runaround loop system is proposed to recover heat from the exhausted air to preheat the supply air. From a review of coil manufacturers' data it is determined that a system can be designed to provide a 60 per cent average sensible heat recovery efficiency.

Using Worksheet 10-1 the cost to heat the air flow to 22°C and provide 50 per cent relative humidity was estimated to be $65,290 per year.

Worksheet 10-9 was used to estimate $17,156 as the portion of the preceding cost which is caused by reheat above 13°C.

The estimated net cost of preheating and humidifying the outdoor air is therefore $65,290 - $17,134 = $48,134 per year.

The heat recovery saving would then be $48,134 x 0.60 = $28,880 per year.

The selected glycol coils would create an additional air pressure drop of 0.25 kPa in the exhaust air stream and 0.15 kPa in the supply air stream. Using Worksheet 10-3, the additional fan energy cost was estimated to be $2,803 per year.

The selected coils also would require a glycol circulation rate of 32 L/s at a total pump pressure of 3.8 kPa(gauge). Worksheet 10-4 was used to estimate the annual pump energy cost to be $96.

Net heat recovery cost saving = $28,880 - $2,803 - $96 = $25,981/yr

The estimated capital cost to install the glycol loop and adjust the fan drives is $40,000.

$$\text{Simple payback} = \frac{\$40,000}{\$25,981} = 1.5 \text{ years}$$

2. Air Treatment to Reduce Outdoor Air Intake

A beverage room in Halifax used an exhaust fan with a flow rate of 950 L/s to clear smoke during peak occupancy periods totalling 6 hours per day, 7 days per week. Make-up air was allowed to enter through doorways and open windows, and often created cold drafts. It was recognized that operation of the fan could be reduced and the cold drafts could be eliminated by the installation of an electronic air cleaner with full recirculating air flow. The room was heated by hot water from a boiler fired with Number 2 oil.

Using Worksheet 10-1, the value of the heating energy saved is estimated to be $1,443 per year. The electrical energy consumed by the electronic air cleaner was determined to be no greater than previously consumed by the exhaust fan.

The capital cost to install the electronic air cleaner is estimated to be $1,500.

$$\text{Simple payback} = \frac{\$1,500}{\$1,443} = 1.0 \text{ years}$$

3. Exhaust Air Treatment for Recirculation

Treatment in the form of filtration and odor removal can often be used to allow recirculation of previously exhausted air, with a resulting saving of energy cost. Successful air treatment systems are available for kitchen hood exhaust, commercial laundry dryer exhaust, and a variety of process plant and workshop exhaust situations. In a laundry dryer exhaust, the treatment and recirculation results in substantial process energy cost saving, in addition to saving outdoor air preheating cost.

The outdoor air preheating cost saving for such applications can be estimated using Worksheet 10-1 and the same procedure as Retrofit Worked Example 2.

4. Reduced Air Handling

The total air circulation rate in a building can often be reduced if cross flow between rooms is used to advantage to minimize the number of supply air outlets. Local code requirements may restrict the use of this technique.

A 4500 m^2 office building in Montreal was heated and air conditioned by a 24 000 L/s terminal reheat air handling system. The air system switched to full recirculation during the unoccupied periods of 118 hours per week.

During the occupied periods of 50 hours per week the supply air temperature was controlled at a constant 13°C. The total fan pressure was measured at 1.5 kPa(gauge).

The building was heated with steam from a boiler plant burning Number 6 fuel oil. By relocating return air inlets in corridors and storage areas and eliminating supply air outlets to those areas, it was determined that the total air circulation rate could be reduced by 2400 L/s. From observation of the zone controls, it was determined that the average reheat temperature rise was 5°C for the affected zones.

Using Worksheet 10-9, reheat energy cost saving was estimated to be $835 per year.

Using Worksheet 10-3, the fan energy cost saving was estimated to be $2,102 per year.

The capital cost for duct revisions and rebalancing of the air system was estimated to be $9,500.

$$\text{Simple payback} = \frac{\$9{,}500}{\$835 + \$2{,}102} = 3.2 \text{ years}$$

5. Add Local Air System

A lecture theatre in a university academic building was served by a 15 000 L/s central air sytem which also served the remainder of the building. The building was occupied 10 hours per day, 6 days per week. The lecture theatre was occupied, on average, 8 hours per week when the building was occupied and 6 hours per week when the rest of the building was unoccupied. The central system was operated continuously 18 hours per day, 6 days per week to make ventilation available to the lecture theatre.

It was proposed that a local 2000 L/s air system be installed to serve the lecture theatre. This would allow reduction of the central system air flow to 13 000 L/s and shut down of the central system during unoccupied periods.

The total fan pressure of the central system was measured at 1.12 kPa(gauge). The total fan pressure required for the local system was estimated to be 0.56 kPa(gauge).

The air system operating conditions are summarized.

- Present central air system operates at 15 000 L/s for 18 x 6 = 108 hours per week.
- Proposed central air system would operate at 13 000 L/s for 10 x 6 = 60 hours per week.
- Proposed local air system would operate at 2000 L/s for 8 + 6 = 14 hours per week.

Worksheet 10-3 was used to estimate the annual fan energy costs.

- Existing central system, $6,290.
- Proposed central system, $3,028.
- Proposed local system, $54.

The net fan energy cost saving is $6,290 - $3,028 - $54 = $3,208 per year.

The estimated capital cost of installing the local air system is $14,000.

$$\text{Simple payback} = \frac{\$14{,}000}{\$3{,}208} = 4.4 \text{ years}$$

6. Add Variable Air Volume to Terminal Reheat System

The office building of Retrofit Worked Example 4 is considered for installation of variable air volume (VAV) control devices and a variable speed drive on the supply fan to achieve further reheat energy savings.

The air flow rate after the previous retrofit was 21 600 L/s, and the total fan pressure was 1.5 kPa(gauge).

The supply air temperature was controlled at a constant 13°C, and the space temperature was 22°C. The minimum outdoor air intake for ventilation was determined from the occupancy and local codes to be 3400 L/s.

The proposed VAV devices would automatically reduce the air flow to each zone to match the cooling load. Reheat coil operation would be allowed only after the air flow reached 50 per cent of the maximum flow. To meet the space temperature requirement of a zone, the reheat coil would then require less energy input to heat the air temperature to the *balance temperature* of the zone. The effect would be a saving of at least 50 per cent of interior zone reheat load plus 50 per cent of that portion of perimeter zone reheat load relating to the balance temperature of each zone.

An accurate assessment of VAV savings requires an hour-by-hour analysis of each zone. However, the savings can be approximated by using the worksheets to analyze the actual heating and cooling energy consumption for the building.

The recorded heating fuel consumption for the 9 heating months was 105 000 litres. At $0.26 per litre the cost was $27,300.

Using Worksheet 10-1, the cost of fuel for heating the ventilation air was estimated to be $4,412. Using Worksheet 10-7, the cost of fuel to account for the building enclosure losses was estimated to be $9,523.

The net fuel cost to reheat to zone balance temperatures was $27,300 - $4,412 - $9,523 = $13,365.

Since this portion of the reheat load is directly proportional to air flow, a 50 per cent reduction in air flow during the heating months would save 0.5 x $13,365 = $6,683.

The recorded heating fuel consumption for the 3 cooling months was 7300 litres. The cost at $0.26 per litre was $1,898. The saving in cooling season reheat would be 0.5 x $1,898 = $949.

The reheat reductions during the cooling months would also be reflected in a comparable reduction in the cooling energy consumption.

The $949 reduction in fuel cost represents $\frac{\$949 \times 42.3 \times 0.75}{0.26}$ = 115 796 MJ of heating.

The comparable reduction in cooling energy at 80 kWh/GH is $\frac{115\ 796 \times 80}{1000}$ = 9264 kWH.

The cost saving at $0.05 per kWh is 9264 x 0.05 = $463.

The fan energy is affected by an increase in static pressure required by the VAV terminal devices, and the average reduction in air flow and corresonding system friction losses. Referring to the fan laws in Fans and Pumps, Module 13, the system friction losses are proportional to the square of the air flow. For a typical VAV system average air flow of 75 per cent of the constant volume system flow the new system pressure loss would be 1.5 x (0.75)2 = 0.84 kPa(gauge). Allowing 0.18 kPa for VAV device pressure loss, the new system fan pressure would be 0.84 + 0.18 = 1.02 kPa(gauge).

Using Worksheet 10-3 the fan energy cost would be $18,922 for the constant volume system and $9,724 for the VAV system. The energy cost saving is $18,922 - $9,724 = $9,198 per year.

The estimated energy cost savings from conversion to a VAV system are:

• Heating season reheat energy savings	= $ 6,683
• Cooling season reheat energy savings	= $ 949
• Cooling season cooling energy savings	= $ 463
• Fan energy cost saving	= $ 9,198
• Total energy savings	= $17,293

The estimated capital cost of installing the variable air volume devices and controls is $40,500.

Simple payback = $\frac{\$40{,}500}{\$17{,}293}$ = 2.3 years

7. Add Pipe Insulation

The methods for evaluating pipe insulation in Steam and Condensate Systems, Module 8 can also be used to evaluate the addition of heating system pipe insulation. Because of escalating fuel costs, pipe insulation levels should be reviewed, at least, every 5 years.

8. Install Duct Insulation

Ducts running outside the conditioned space, such as in attics or exterior duct shafts, should be insulated to a coefficient of transmission at least equivalent to that of the building enclosure. Heat loss and gain in such ducts result in higher energy input rates at the air handling unit to meet the space requirements, and reductions in the system efficiency.

9. Install Heat Recovery Chiller

The installation of a chiller which is capable of rejecting its condenser heat at the temperature of the building heating water can provide the energy for reheat over a significant period of the year. The heat rejected by a chiller includes both the cooling load and the energy input to operate the chiller. For a chiller requiring 80 kWh input per gigajoule (1000 MJ) of cooling the heat energy rejected will be (80 x 3.6) + 1000 = 1288 MJ per gigajoule of cooling.

$$\text{Heat energy output per kWh of input energy} = \frac{1288}{80}$$

$$= 16.1 \text{ MJ/kWh}$$

$$\text{At \$0.05 per kWh the cost of heat energy} = \frac{\$0.05}{16.1}$$

$$= \$0.00310/\text{MJ}$$

$$= \$3.10/\text{GJ}$$

The comparative cost of heat energy from a #6 oil-fired boiler plant using 1 per cent sulphur oil with a heating value of 40.5 MJ/L at $0.26 per litre and 75 per cent efficiency would be:

$$\text{Fuel oil cost} = \frac{\$0.26}{40.5 \times 0.75} = \$0.0086/\text{MJ}$$

$$= \$8.60/\text{GJ}$$

During the cooling season the heat energy recovered is essentially free, since it would otherwise be rejected to outside air. During intermediate seasons the lower cost energy from the chiller creates an opportunity for cost saving by recirculating return air through the air system cooling coil instead of using outdoor air. The resulting chiller load provides low cost energy for reheat and perimeter heating systems, and saves boiler plant fuel.

Evaluation of the energy cost saving requires an hour-by-hour analysis of the building coincident heating and cooling loads, and cannot reasonably be accomplished by manual calculation methods.

10. Extend Utilization of Heat Recovery Chiller

The utilization of the heat recovery chiller of Retrofit Worked Example 9 can be extended into the winter season by adding optional cooling loads from process heat sources and exhaust air systems. Where process loads offer a suitable minimum chiller load the operation can be extended to 12 months per year.

11. Control Garage Ventilation by Carbon Monoxide Level

A parking garage in an apartment building in Fredericton was continuously ventilated with 5500 L/s of outdoor air. The air was heated to maintain the garage space temperature at 5°C. Heat was provided from a boiler plant fired with Number 2 fuel oil. The ventilation system total fan pressure was measured at 0.35 kPa(gauge). From observation of the traffic to and from the garage it was estimated that operation of the ventilation system could be reduced to 4 hours per day through the installation of a carbon monoxide sensor to automatically start and stop the fans.

Using Worksheet 10-1, the heating energy cost saving was estimated to be $5,905 per year.

Using Worksheet 10-3, the fan energy cost saving was estimated to be $935 per year.

The estimated capital cost to install the carbon monoxide monitor and control system is $10,000.

$$\text{Simple payback} = \frac{\$10{,}000}{\$5{,}905 + \$935} = 1.5 \text{ years}$$

12. Install Building Energy Management System

A computerized building energy management system can accomplish energy cost savings in addition to the savings from individual actions by monitoring and integrating the various control functions. Microprocessors for small office buildings, and powerful microcomputers and minicomputers for larger building comlexes are available.

The analysis and selection of such equiment should be based on a professional review of the requirements for the particular facility.

13. Install Roof Water Sprays

For buildings with large roof areas such as factories, the installation of roof sprays can significantly reduce the summer cooling energy. The evaporative cooling effect of the water spray effectively reduces the summer heat gain through the roof, and may eliminate the need for refrigerated cooling in buildings with low internal heat gains.

Outside Air Intake Rate

Worksheet 10-2

(Page 1 of 1)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Return air temperature (T1) 22 °C

Outdoor air temperature (T2) 0 °C

Mixed air stream temperature (T3) 18 °C

Mixed air stream flow rate (fm) 10 000 L/s

Outdoor air intake rate, fo $= \frac{fm \times (T1-T3)}{T1 - T2}$

$$= \frac{10000 \times (22-18)}{22-0}$$

$= 1818$ L/s

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #1

Date: 85/09/10

Location: HOUSEKEEPING

By: MBE

Degree Days Below 18°C (DDh) 5990 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) -34 °C (NBC)

Outdoor Air Flow (fa) 1818 L/s

Maximum Temperature Difference (T1-T2) 56 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

= 1818 x (56) x 4.345

= 442356 kJ/h

Operating hours per week 128 h (1)

Average operating hours per day = $\frac{(1)}{7}$ = 18.29 h (2)

Annual Sensible Heat, AHs = $\frac{\text{Qs x DDh x (2)}}{\text{(T1 - T2) x 1000}}$

= $\frac{442356 \times 5990 \times 18.29}{56 \times 1000}$

= 865415 MJ (3)

Indoor humidity 35 %

Indoor humidity factor (H1) 6 g/kg (Figure 4)

Outdoor humidity 60 % (TYPICAL FOR WINTER)

Outdoor humidity factor (H2) 1 g/kg (Figure 4)

(BY EXTRAPOLATION)

Maximum humidification heat, QL = fa x (H1 - H2) x 10.84

= 1818 x (6-1) x 10.84

= 98536 kJ/h

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE # 1　　Date: 85/09/10

Location: HOUSEKEEPING　　By: MBE

$$\text{Annual humidification heat, AHL} = \frac{\text{QL} \times \text{DDh} \times \frac{(2)}{2}}{(\text{T1} - \text{T2}) \times 1000}$$

$$= \frac{98536 \times 5990 \times (18.29/2)}{56 \times 1000}$$

$$= 96387 \text{ MJ} \qquad (4)$$

Total annual heat = (3) + (4)

= 961802 MJ (5)

Fuel type NATURAL GAS

Fuel cost/unit $ 0.21 /m^3 (6)

Fuel heat value/unit 37.20 MJ (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

$$\text{Annual cost} = \frac{(5) \times (6)}{(7) \times (8)}$$

$$= \frac{961802 \times 0.21}{37.20 \times 0.75}$$

= $ 7240 (9)

Reduction in operating hours/week 128 h (10)

Initial operating hours/week 128 h (11)

$$\text{Annual cost saving} = \frac{(9) \times (10)}{(11)}$$

$$= \frac{7240 \times 128}{128}$$

= $ 7240 /yr

Fan Energy Cost

Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #3 Date: 85 /09/10

Location: HOUSEKEEPING By: MBE

Air flow rate (fa) 7500 L/s

Total fan pressure (P) 0.200 (REDUCTION) kPa

Fan power, Wf = $\frac{fa \times P}{750}$

$$= \frac{7500 \times 0.200}{750}$$

= 2.0 kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 2.0 × 8760 × 0.05

= $ 876 /yr

Heating Of Outside Air

Worksheet 10-1

(Page 1 of 2)

Company: WORKED EXAMPLE #4 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Degree Days Below 18°C (DDh) 3622 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) -18 °C (NBC)

Outdoor Air Flow (fa) 3800 L/s

Maximum Temperature Difference (T1-T2) 40 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

= 3800 x (40) x 4.345

= 660440 kJ/h

Operating hours per week 84 h (1)

Average operating hours per day $= \frac{(1)}{7} =$ 12 h (2)

Annual Sensible Heat AHs $= \frac{\text{Qs x DDh x (2)}}{\text{(T1-T2) x 1000}}$

$= \frac{660440 \times 3622 \times 12}{40 \times 1000}$

= 717 634 MJ (3)

Indoor humidity 35 %

Indoor humidity factor (H1) 6 g/kg (Figure 4)

Outdoor humidity 60 % (TYPICAL FOR WINTER)

Outdoor humidity factor (H2) 1 g/kg (Figure 4)

(BY EXTRAPOLATION)

Maximum humidification heat, QL = fa x (H1-H2) x 10.84

= 3800 x (6 - 1) x 10.84

= 205960 kJ/h

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE #4 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1\text{-}T2) \times 1000}$

$= \dfrac{205960 \times 3622 \times (12/2)}{40 \times 1000}$

= 111 898 MJ (4)

Total annual heat = (3) + (4)

= 829 532 MJ (5)

Fuel type #6 FUEL OIL

Fuel cost/unit $ 0.26 /L (6)

Fuel heat value/unit 42.3 MJ/L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

$= \dfrac{829532 \times 0.26}{42.3 \times 0.75}$

= $ 6798 (9)

Reduction in operating hours/week 28 h (10)

Initial operating hours/week 84 h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

$= \dfrac{6798 \times 28}{84}$

= $ 2266 /yr

Cooling Of Internal Heat Gains

Worksheet 10-6

(Page 1 of 2)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

*No. of months with monthly mean temperature above 18°C 3 (Environment Canada) (1)

Number of lights 50 (2)

Power input per light 100 W (3)

Heat from lights = (2) x (3) x 3.6

= 50 x 100 x 3.6

= 18 000 kJ/h (4)

Number of people N/A (5)

Sensible heat gain per person N/A kJ/h (Table 5) (6)

Latent heat gain per person N/A kJ/h (Table 5) (7)

Total heat gain from people = (5) x [(6) + (7)]

= ______

= N/A kJ/h (8)

Process heat gain N/A kJ/h (9)

Factors of duration:

fd (lights) 0.8 , fd (people) N/A , fd (proc) N/A

Factors of utilization:

fu (lights) 60 MIN./h , fu (people) N/A , fu (proc) N/A

Total correction factors $\frac{\text{fd x fu}}{60}$:

0.8 (10) N/A (11) N/A (12)

Cooling of Internal Heat Gains

Worksheet 10-6
(Page 2 of 2)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Total cooling load (Qt) = [(4) x (10)] + [(8) x(11)] + [(9) x (12)]

= 18000 x 0.8

= 14400 kJ/h (13)

Hours of operation per month 84 x 52/12 = 364 (h) (14)

No. of months in operation 3 (15)

(For systems using outdoor air frèe cooling use value from (1), otherwise use actual operating months)

Annual heat gain cooling, ACi $= \frac{(13) \text{ x } (14) \text{ x } (15)}{1000}$

$= \frac{14400 \times 364 \times 3}{1000}$

= 15725 MJ/yr (16)

Unit energy cost $ 0.05 /kWh (17)

Energy consumption/GJ cooling 80 (18)
(if unknown use 80 kWh/GJ)

Annual cost $= \frac{(16) \text{ x } (17) \text{ x } (18)}{1000}$

$= \frac{15725 \times 0.05 \times 80}{1000}$

= $ 63 (19)

Reduction in operating period 84 h/WEEK (20)

Initial operating period 84 h/WEEK (21)

Cost saving $= \frac{(19) \text{ x } (20)}{(21)}$

$= \frac{63 \times 84}{84}$

= $ 63 /yr

*Can be determined from Environment Canada weather data for specific locations.

Heating Of Outside Air
Average Temperature Method

Worksheet 10-9

(Page 1 of 1)

Company: WORKED EXAMPLE #6 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

*No. of months with monthly mean temperature below 18°C 10 (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= 10 x 730 = 7300 h (2)

Average temperature rise 3 °C (3)

Hours of operation/week 168 h (4)

Hours of operation during heating season = (4) x (1) x 4.345

= 7300 h (5)

Outdoor air flow (fa) 7500 L/s

Average heating, Q = fa x (3) x 4.345

= 7500 x 3 x 4.345

= 97763 kJ/h (6)

Annual energy, AH = $\frac{(6) \times (5)}{1000}$

= $\frac{97763 \times 7300}{1000}$

= 713 670 MJ (7)

Fuel type #2 FUEL OIL

Fuel cost/unit $0.40/L (8)

Fuel heat value/unit 38.68 MJ/L (APPENDIX C) (9)

Heating system efficiency 0.75 (10)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost = $\frac{(7) \times (8)}{(9) \times (10)}$

= $\frac{713670 \times 0.40}{38.68 \times 0.75}$

= $ 9840

*Can be determined from Environment Canada weather data for specific locations.

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #10 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Degree Days Below 18°C (DDh) 5451 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) -30 °C (NBC)

Outdoor Air Flow (fa) 375 L/s

Maximum Temperature Difference (T1-T2) 52 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

$= 375 \times (52) \times 4.345$

$= 84728$ kJ/h

Operating hours per week 70 h (1)

Average operating hours per day $= \frac{(1)}{7} =$ 10 h (2)

Annual Sensible Heat, AHs $= \frac{Qs \times DDh \times (2)}{(T1 - T2) \times 1000}$

$= \frac{84728 \times 5451 \times 10}{52 \times 1000}$

$= 88818$ MJ (3)

Indoor humidity 35 %

Indoor humidity factor (H1) 6 g/kg (Figure 4)

Outdoor humidity 60 % (TYPICAL FOR WINTER)

Outdoor humidity factor (H2) 1 g/kg (Figure 4) (BY EXTRAPOLATION)

Maximum humidification heat, QL = fa x (H1 - H2) x 10.84

$= 375 \times (6-1) \times 10.84$

$= 20325$ kJ/h

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE #10

Date: 85/09/10

Location: HOUSEKEEPING

By: MBE

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1 - T2) \times 1000}$

$= \dfrac{20325 \times 5451 \times (10/2)}{52 \times 1000}$

$=$ 10 653 MJ (4)

Total annual heat = (3) + (4)

= 99 471 MJ (5)

Fuel type #2 FUEL OIL

Fuel cost/unit $0.40/L (6)

Fuel heat value/unit 38.68 MJ/L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

$= \dfrac{99471 \times 0.40}{38.68 \times 0.75}$

= $ 1372 (9)

Reduction in operating hours/week 70 h (10)

Initial operating hours/week 70 h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

$= \dfrac{1372 \times 70}{70}$

= $ 1372 /yr

Cooling Of Outdoor Air — Degree Day Method

Worksheet 10-5
(Page 1 of 2)

Company: WORKED EXAMPLE #10 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Degree days above 18°C (DDc) 123 (Table 2) (1)

Design outdoor conditions

Dry bulb temperature (T1) 29 °C (NBC)

Wet bulb temperature 21 °C (NBC)

Humidity factor (H1) 12.5 g/kg (Figure 4)

Indoor design conditions

Dry bulb temperature (T2) 24 °C

Relative humidity 50 %

Humidity factor (H2) 9.5 g/kg (Figure 4)

Outdoor air flow rate (fa) 375 L/s

Sensible cooling load, Qs = fa x (T1-T2) x 4.345

= 375 × (29 - 24) × 4.345

= 8147 kJ/h (2)

Latent cooling load, QL = fa x (H1-H2) x 10.84

= 375 × (12.5 - 9.5) × 10.84

= 12195 kJ/h (3)

Total cooling load = (2) + (3) = 20342 kJ/h (4)

Cooling Of Outdoor Air — Degree Day Method

Worksheet 10-5
(Page 2 of 2)

Company: WORKED EXAMPLE #10 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Operating hours per week 70 h (5)

Average operating hours per day $= \frac{(5)}{7}$

$=$ 10 h (6)

Annual outdoor air cooling, AC $= \frac{(4) \times (1) \times (6)}{(T1\text{-}T2) \times 1000}$

$= \frac{20342 \times 123 \times 10}{(29-24) \times 1000}$

$=$ 5004 MJ (7)

Unit energy cost $ 0.05 /kWh (8)

Energy consumption/GJ cooling 80 kWh (9)
(if unknown use 80 kWh/GJ)

Annual cost $= \frac{(7) \times (8) \times (9)}{1000}$

$= \frac{5004 \times 0.05 \times 80}{1000}$

= $ 20 (10)

Reduction in operating hours/week 70 h (11)

Initial operating hours/week 70 h (12)

Cost saving $= \frac{(10) \times (11)}{(12)}$

$= \frac{20 \times 70}{70}$

= $ 20 /yr

Building Heat Loss Reduction By Lowering Space Temperature

Worksheet 10-12

(Page 1 of 1)

Company: WORKED EXAMPLE #11 Date: 85/09/10

Location: HOUSEKEEPING By: MBE

Degree days below 18°C (DDh) 4740 (Table 2) (1)

No. of months with avg. temp. below 18°C 10 (Environment Canada) (2)

No. of days in heating season = (2) x 30.4

= 10 x 30.4

= 304 (3)

Type of fuel used #2 FUEL OIL

Unit cost of fuel $0.40/L (4)

Recorded annual fuel consumption 70000L (5)

Annual fuel consumption per degree day $= \frac{(5)}{(1)}$

$= \frac{70000}{4740}$

= 14.8L (6)

Initial space temperature (T1) 22 °C

Reduced space temperature (T2) 20 °C

Hours per day at reduced temperature 24 h (7)

Reduction in heating degree days $= \frac{(3) \times (T1 - T2) \times (7)}{24}$

$= \frac{304 \times (22-20) \times 24}{24}$

= 608 degree days (8)

Annual fuel cost saving = (6) x (8) x (4)

= 14.8 x 608 x 0.40

= $ 3599

Fan Energy Cost

Worksheet 10-3

(Page 1 of 1)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: LOW COST By: MBE

Air flow rate (fa) 20 000 L/s

Total fan pressure (P) 1.2 kPa

$$\text{Fan power, Wf} = \frac{\text{fa x P}}{750}$$

$$= \frac{20\,000 \times 1.2}{750}$$

= 32 kW (1)

Reduced operating time 98 x 52 = 5096 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 32 x 5096 x 0.05

= $ 8 154 /yr

Pump Energy Cost

Worksheet 10-4

(Page 1 of 1)

Company: WORKED EXAMPLE #2 Date: 85/09/10

Location: LOW COST By: MBE

Water flow rate (fw) 20 L/s

Total pump pressure (P) 150 kPa

Pump Power, Wp $= \frac{fw \times P}{500}$

$= \frac{20 \times 150}{500}$

= 6 kW (1)

Reduced operation time 98 x 52 = 5096 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost Saving, = (1) x (2) x (3)

= 6 x 5096 x 0.05

= $ 1529 /yr

Cooling Of Internal Heat Gains

Worksheet 10-6
(Page 1 of 2)

Company: WORKED EXAMPLE #3 Date: 85/09/10

Location: LOW COST By: MBE

*No. of months with monthly mean temperature above 18°C 2 (Environment Canada) (1)

Number of lights 30 (2)

Power input per light 90 W (3)

Heat from lights = (2) x (3) x 3.6

= 30 x 90 x 3.6

= 9720 kJ/h (4)

Number of people 3 (5)

Sensible heat gain per person 270 kJ/h (Table 5) (6)

Latent heat gain per person 270 kJ/h (Table 5) (7)

Total heat gain from people = (5) x [(6) + (7)]

= 3 x (270 + 270)

= 1620 kJ/h (8)

Process heat gain 198000 kJ/h (9)

Factors of duration:

fd (lights) 1 , fd (people) 1 , fd (proc) 1

Factors of utilization:

fu (lights) 60 , fu (people) 60 , fu (proc) 60

Total correction factors $\frac{\text{fd x fu}}{60}$:

1 (10) 1 (11) 1 (12)

Cooling of Internal Heat Gains

Worksheet 10-6
(Page 2 of 2)

Company: WORKED EXAMPLE #3 Date: 85/09/10

Location: LOW COST By: MBE

Total cooling load (Qt) = [(4) x (10)] + [(8) x(11)] + [(9) x (12)]

= 9720 + 1620 + 198000

= 209340 kJ/h (13)

Hours of operation per month 730 (h) (14)

No. of months in operation 12 (15)

(For systems using outdoor air free`cooling use value from (1), otherwise use actual operating months)

Annual heat gain cooling, ACi = $\frac{(13) \times (14) \times (15)}{1000}$

= $\frac{209340 \times 730 \times 12}{1000}$

= 1833818 MJ/yr (16)

Unit energy cost $ 0.05 /kWh (17)

Energy consumption/GJ cooling 80 (18)
(if unknown use 80 kWh/GJ)

Annual cost = $\frac{(16) \times (17) \times (18)}{1000}$

= $\frac{1833818 \times 0.05 \times 80}{1000}$

= $ 7335 (19)

Reduction in operating period 10 MONTHS (20)

Initial operating period 12 MONTHS (21)

Cost saving = $\frac{(19) \times (20)}{(21)}$

= $\frac{7335 \times 10}{12}$

= $ 6113 /yr

*Can be determined from Environment Canada weather data for specific locations.

Building Heat Loss Reduction
By Lowering Space Temperature
Worksheet 10-12
(Page 1 of 1)

Company: WORKED EXAMPLE #4 Date: 85/09/10

Location: LOW COST By: MBE

Degree days below 18°C (DDh) 4740 (Table 2) (1)

No. of months with avg. temp. below 18°C 10 (Environment Canada) (2)

No. of days in heating season = (2) x 30.4

= 10 x 30.4

= 304 (3)

Type of fuel used ELECTRICITY

Unit cost of fuel $ 0.05 /KWh (4)

Recorded annual fuel consumption 3 80 000 KWh (5)

Annual fuel consumption per degree day $= \frac{(5)}{(1)}$

$= \frac{380\,000}{4740}$

= 80.17 KWh (6)

Initial space temperature (T1) 22 °C

Reduced space temperature (T2) 15 °C

Hours per day at reduced temperature 12 h (7)

Reduction in heating degree days $= \frac{(3) \times (T1 - T2) \times (7)}{24}$

$= \frac{304 \times (22 - 15) \times 12}{24}$

= 1064 degree days (8)

Annual fuel cost saving = (6) x (8) x (4)

= 80.17 x 1064 x 0.05

= $ 4265

Annual Heat Transfer Through A Building Enclosure
Degree Day Method
Worksheet 10-7
(Page 1 of 2)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: LOW COST By: MBE

Degree Days Below 18°C (DD) 4538 (Table 2)

Warm Side Temperature (T1) 22 °C

Cold Side Temperature (T2) -26 °C

Wall coef. of transmission (Uw) 0.5 W/(m^2.°C)

Window coef. of transmission (Ug) 3.3 W/(m^2.°C)

Roof coef. of transmission (Ur) N/A W/(m^2.°C)

Gross wall area 275 m^2 (1)

Window area (Ag) 100 m^2

Net wall area, Aw = (1) − Ag

= 175 m^2

Roof area N/A m^2

Enclosure heat loss = [(Aw x Uw) + (Ag x Ug) + (Ar x Ur)] x (T1-T2) x 3.6

= (175×0.5)+(100×3.3)+(0)×[(22-(-26)]×3.6

= (87.5+330)×48×3.6

= 72144 kJ/h (2)

Infiltration rate (CH) 0.50 air changes/h (Table 4)

Room volume (V) 1140 m^3

Annual Heat Transfer Through A building Enclosure
Degree Day Method
Worksheet 10-7
(Page 2 of 2)

Company: WORKED EXAMPLE #5   Date: 85/09/10

Location: LOW COST   By: MBE

Infiltration heat loss $= \frac{V \times CH}{3.6} \times (T1\text{-}T2) \times 4.345$

$= \frac{(1140 \times 0.5)}{3.6} \times 48 \times 4.345$

$=$ 33022 kJ/h (3)

Total heat loss, Q = (2) + (3)

$=$ 105160 kJ/h

Annual Heat Flow Through Component, AH $= \frac{Q \times DD \times 18}{(T1\ \text{-}T2) \times 1000}$

$= \frac{105160 \times 4538 \times 18}{48 \times 1000}$

$=$ 178956 kJ (4)

Fuel Type #6 FUEL OIL

Fuel Cost/Unit $0.26/L (5)

Fuel Heat Value/Unit 42.3 MJ/L (APPENDIX C) (6)

Heating System Efficiency 0.75 (7)
(If unknown use 0.75 for oil or gas, 1.0 for electricity).

Annual fuel consumption $= \frac{(4)}{(6) \times (7)}$

$= \frac{178956}{42.3 \times 0.75}$

$=$ 5641 L (8)

Annual Costs = (8) x (5)

$=$ 5641 x 0.26

$= \$$ 1467

Building Heat Loss Reduction
By Lowering Space Temperature
Worksheet 10-12
(Page 1 of 1)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: LOW COST By: MBE

Degree days below 18°C (DDh) 4538 (Table 2) (1)

No. of months with avg. temp. below 18°C 9 (Environment Canada) (2)

No. of days in heating season = (2) x 30.4

= 9 x 30.4

= 274 (3)

Type of fuel used #6 FUEL OIL

Unit cost of fuel $ 0.26/L (4)

ESTIMATED ~~Recorded~~ annual fuel consumption 5641 L (WORKSHEET 6-7) (5)

Annual fuel consumption per degree day $= \frac{(5)}{(1)}$

$= \frac{5641}{4538}$

= 1.243 L (6)

Initial space temperature (T1) 22 °C

Reduced space temperature (T2) 13 °C

Hours per day at reduced temperature 24 h (7)

Reduction in heating degree days $= \frac{(3) \times (T1-T2) \times (7)}{24}$

$= \frac{274 \times (22-13) \times 24}{24}$

= 2466 degree days (8)

Annual fuel cost saving = (6) x (8) x (4)

= 1.243 x 2466 x 0.26

= $ 797

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #6 Date: 85/09/10

Location: LOW COST By: MBE

Degree Days Below 18°C (DDh) 5991 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) -34 °C (NBC)

Outdoor Air Flow (fa) 609 L/s

Maximum Temperature Difference (T1-T2) 56 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

= 609 x 56 x 4.345

= 148182 kJ/h

Operating hours per week 128 h (1)

Average operating hours per day = $\frac{(1)}{7}$ = 18.29 h (2)

Annual Sensible Heat, AHs = $\frac{\text{Qs x DDh x (2)}}{\text{(T1 - T2) x 1000}}$

= $\frac{148182 \times 5991 \times 18.29}{56 \times 1000}$

= 289948 MJ (3)

Indoor humidity 35 %

Indoor humidity factor (H1) 6 g/kg (Figure 4)

Outdoor humidity 60 % (TYPICAL FOR WINTER)

Outdoor humidity factor (H2) 1 g/kg (Figure 4)

(BY EXTRAPOLATION)

Maximum humidification heat, QL = fa x (H1 - H2) x 10.84

= 609 x (6-1) x 10.84

= 33008 kJ/h

Heating Of Outside Air
Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE #6 Date: 85/09/10

Location: LOW COST By: MBE

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1 - T2) \times 1000}$

$= \dfrac{33008 \times 5991 \times (18.29/2)}{56 \times 1000}$

$=$ 32293 MJ (4)

Total annual heat = (3) + (4)

= 322241 MJ (5)

Fuel type NATURAL GAS

Fuel cost/unit $0.21/M^3 (6)

Fuel heat value/unit 37.20 MJ/M^3 (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

$= \dfrac{322241 \times 0.21}{37.20 \times 0.75}$

= $ 2425 (9)

Reduction in operating hours/week 128 h (10)

Initial operating hours/week 128 h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

$= \dfrac{2425 \times 128}{128}$

= $ 2425 /yr

Excess Heat Output By A Terminal Heating Unit

Worksheet 10-11

Page 1 of 2

Company: WORKED EXAMPLE #7 Date: 85/09/10

Location: LOW COST By: MBE

Measured heating water supply temperature 90 °C (1)

Measured heating water leaving temperature with fan operating (T2) 80 °C (2)

Average heating water temperature, $Ta = \frac{(T1 + T2)}{2}$

$= \frac{90 + 80}{2}$

= 85 °C

Catalogue output rating at Ta water temperature (Qr) 45 000 kJ/h

Calculate heating water flow, Qr = fw x (T1 - T2) x 15 000

Therefore, $fw = \frac{Qr}{(T1 - T2) \times 15\ 000}$

$= \frac{45\ 000}{(90 - 80) \times 15\ 000}$

= 0.3 L/s (3)

Measured heating water leaving temperature with fan off 88 °C (4)

Calculate output with fan off, Q = (3) x [(1) – (4)] x 15 000

= 0.3 x (90 - 88) x 15000

= 9 000 kJ/h (5)

*No. of months with avg. temp. above 18°C 3 (Environment Canada) (6)

Excess Heat Output By A Terminal Heating Unit

Worksheet 10-11

(Page 2 of 2)

Company: WORKED EXAMPLE #7 Date: 85/09/10

Location: LOW COST By: MBE

No. of hours with excess heat output = (6) x 730

= 3 x 730

= 2190 h (7)

Type of fuel used #2 FUEL OIL

Unit cost of fuel $0.40/L (8)

Unit heat value of fuel 38.68 MJ/L (APPENDIX C) (9)

Heating system efficiency 0.75 (10)

(If unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual fuel consumption = $\frac{(5) \times (7)}{(9) \times (10) \times 1000}$

= $\frac{9000 \times 2190}{38.68 \times 0.75 \times 1000}$

= 680 L (11)

Annual fuel cost saving = (11) x (8)

= 680 x 0.40

= $ 272

*Can be determined from Environment Canada weather data for specific locations.

Supply Air Reheat
Average Temperature Method
Worksheet 10-10
(Page 1 of 1)

Company: WORKED EXAMPLE #9   Date: 85/09/10

Location: LOW COST   By: MBE

*No. of months with monthly mean temperature below 18°C 10 (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= 7300 h (2)

Original Supply Air Temperature (T1) 13 °C

Average increased supply air temperature (T2) 16 °C

Hours of Operation/Week 60 (3)

Hours of operation during heating season = (3) x (1) x 4.33

= 2598 h (4)

Supply air flow (fa) 20000 L/s

Average reduction in reheat, Q = fa x (T2-T1) x 4.345

= 20000 x (16-13) x 4.345

= 260700 kJ/h (5)

Annual reheat savings $= \frac{(4) \times (5)}{1000}$

$= \frac{2598 \times 260700}{1000}$

= 677299 MJ (6)

Fuel type #6 FUEL OIL

Fuel cost/unit $0.26/L (7)

Fuel heat value/unit 42.3 MJ/L (APPENDIX C) (8)

Heating system efficiency 0.75 (9)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual fuel cost savings $= \frac{(6) \times (7)}{(8) \times (9)}$

$= \frac{677299 \times 0.26}{42.3 \times 0.75}$

= $ 5551

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #11 Date: 85/09/10

Location: LOW COST By: MBE

Degree Days Below 18°C (DDh) 6063 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) -37 °C (NBC)

Outdoor Air Flow (fa) 3000 L/s

Maximum Temperature Difference (T1 - T2) 59 °C

Maximum Sensible Heating, Qs = fa x (T1 - T2) x 4.345

= 3000 x 59 x 4.345

= 769065 kJ/h

Operating hours per week 168 h (1)

Average operating hours per day = $\frac{(1)}{7}$ = 24 h (2)

Annual Sensible Heat, AHs = $\frac{Qs \times DDh \times (2)}{(T1 - T2) \times 1000}$

= $\frac{769065 \times 6063 \times 24}{59 \times 1000}$

= 1896749 MJ (3)

Indoor humidity 35 %

Indoor humidity factor (H1) 6 g/kg (Figure 4)

Outdoor humidity 60 %

Outdoor humidity factor (H2) 1 g/kg (Figure 4)

Maximum humidification heat, QL = fa x (H1 - H2) x 10.84

= 3000 x (6 - 1) x 10.84

= 162600 kJ/h

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE # 11 Date: 85/09/10

Location: LOW COST By: MBE

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1 - T2) \times 1000}$

$= \dfrac{162600 \times 6063 \times (24/2)}{59 \times 1000}$

= 200511 MJ (4)

Total annual heat = (3) + (4)

= 2097260 MJ (5)

Fuel type NATURAL GAS

Fuel cost/unit \$ 0.21 / m^3 (6)

Fuel heat value/unit 37.2 MJ/m^3 (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

$= \dfrac{2097260 \times 0.21}{37.2 \times 0.75}$

= \$ 15786 (9)

Reduction in operating hours/week N/A h (10)

Initial operating hours/week N/A h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

= __________

= \$ 15786 /yr

Annual Heat Transfer Through A Building Enclosure
Degree Day Method
Worksheet 10-7
(Page 1 of 2)

Company: WORKED EXAMPLE #12 Date: 85/09/10

Location: LOW COST By: MBE

Degree Days Below 18°C (DD) N/A (Table 2)

Warm Side Temperature (T1) 32 °C

Cold Side Temperature (T2) 20 °C

Wall coef. of transmission (Uw) 5.0 W/(m²·°C)

Window coef. of transmission (Ug) N/A W/(m²·°C)

Roof coef. of transmission (Ur) N/A W/(m²·°C)

Gross wall area 30 m² (1)

Window area (Ag) N/A m²

Net wall area, Aw = (1) − Ag

= 30 m²

Roof area N/A m²

Enclosure heat loss = [(Aw x Uw) + (Ag x Ug) + (Ar x Ur)] x (T1 - T2) x 3.6

= 30 x 5.0 x (32 - 20) x 3.6

=

= 6480 kJ/h (2)

Infiltration rate (CH) N/A air changes/h (Table 4)

Room volume (V) N/A m³

Annual Heat Transfer Through A Building Enclosure
Degree Day Method
Worksheet 10-7
(Page 2 of 2)

Company: WORKED EXAMPLE #12 Date: 85/09/10

Location: LOW COST By: MBE

Infiltration heat loss $= \frac{V \times CH}{3.6} \times (T1 - T2) \times 4.345$

= ______

= N/A kJ/h (3)

Total heat loss, Q = (2) + (3)

= 6480 kJ/h (GAIN TO OCCUPIED SPACE)

Annual heat flow through component, AH $= \frac{Q \times DD \times 24}{(T1 - T2) \times 1000}$

= ______

= N/A kJ (4)

Fuel Type N/A

Fuel Cost/Unit N/A (5)

Fuel Heat Value/Unit N/A (APPENDIX C) (6)

Heating System Efficiency N/A (7)
(If unknown use 0.75 for oil or gas, 1.0 for electricity).

Annual fuel consumption $= \frac{(4)}{(6) \times (7)}$

= ______

= N/A (8)

Annual Costs = (8) x (5)

= ______

= $ N/A

Cooling Of Internal Heat Gains

Worksheet 10-6

(Page 1 of 2)

Company: WORKED EXAMPLE #12 Date: 85/09/10

Location: LOW COST By: MBE

*No. of months with monthly mean temperature above 18°C N/A (Environment Canada) (1)

Number of lights N/A (2)

Power input per light N/A W (3)

Heat from lights = (2) x (3) x 3.6

= ________

= N/A kJ/h (4)

Number of people N/A (5)

Sensible heat gain per person N/A kJ/h (Table 5) (6)

Latent heat gain per person N/A kJ/h (Table 5) (7)

Total heat gain from people = (5) x [(6) + (7)]

= ________

= N/A kJ/h (8)

Process heat gain 22320 kJ/h (9)

Factors of duration:

fd (lights) N/A , fd (people) N/A , fd (proc) 1

Factors of utilization:

fu (lights) N/A , fu (people) N/A , fu (proc) 60

Total correction factors $\frac{\text{fd x fu}}{60}$:

N/A (10) N/A (11) 1 (12)

Cooling of Internal Heat Gains

Worksheet 10-6

(Page 2 of 2)

Company: WORKED EXAMPLE #12 Date: 85/09/10

Location: LOW COST By: MBE

Total cooling load (Qt) = [(4) x (10)] + [(8) x(11)] + [(9) x (12)]

= 22320 x 1

= 22320 kJ/h (13)

Hours of operation per month 730 (h) (14)

No. of months in operation 12 (15)

(For systems using outdoor air free cooling use value from (1), otherwise use actual operating months)

Annual heat gain cooling, ACi = $\frac{(13) \times (14) \times (15)}{1000}$

= $\frac{22320 \times 730 \times 12}{1000}$

= 195523 MJ/yr (16)

Unit energy cost $ 0.05 /kWh (17)

Energy consumption/GJ cooling 80 (18)
(if unknown use 80 kWh/GJ)

Annual cost = $\frac{(16) \times (17) \times (18)}{1000}$

= $\frac{195523 \times 0.05 \times 80}{1000}$

= $ 782 (19)

Reduction in operating period N/A h (20)

Initial operating period N/A h (21)

Cost saving = $\frac{(19) \times (20)}{(21)}$

=

= $ 782 /yr

*Can be determined from Environment Canada weather data for specific locations.

Fan Energy Cost
Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #12 Date: 85/09/10

Location: LOW COST By: MBE

Air flow rate (fa) 1105 L/s

Total fan pressure (P) 0.1 kPa

Fan power, Wf $= \frac{fa \times P}{750}$

$= \frac{1105 \times 0.1}{750}$

$= 0.15$ kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 0.15 x 8760 x 0.05

= $ 66 /yr

Annual Heat Transfer Through A Building Enclosure Component — Average Temperature Method

Worksheet 10-8
(Page 1 of 1)

Company: WORKED EXAMPLE #13 Date: 85/09/10

Location: LOW COST By: MBE

*No. of months with monthly mean temperature below 18°C 10 (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= 7300 h (2)

Average temperature difference across component 35-20=15 °C (3)

Coefficient of transmission (U) 0.6 $W/(m^2 \cdot °C)$

Area of enclosure component (A) 15 000 m^2

Heat flow rate through component, Q = A x U x (3) x 3.6

= 15000 x 0.6 x 15 x 3.6

= 486 000 kJ/h (4)

Annual heat flow through component, AH = $\frac{(2) \times (4)}{1000}$

$= \frac{7300 \times 486000}{1000}$

= 3547800 MJ (5)

Fuel type #2 FUEL OIL

Fuel cost/unit $ 0.40 / L (6)

Fuel heat value/unit 38.68 MJ / L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity

Annual Costs = $\frac{(5) \times (6)}{(7) \times (8)}$

$= \frac{3547800 \times 0.40}{38.68 \times 0.75}$

= $ 48918

*Can be determined from Environment Canada weather data for specific locations.

Fan Energy Cost

Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #14

Date: 85/09/10

Location: LOW COST

By: MBE

Air flow rate (fa) 11980 L/s

Total fan pressure (P) 0.049 kPa (REDUCTION)

Fan power, Wf = $\frac{\text{fa x P}}{750}$

$= \frac{11980 \times 0.049}{750}$

= 0.783 kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 0.783 × 8760 × 0.05

= $ 343 /yr

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: RETROFIT By: MBE

Degree Days Below 18°C (DDh) 5451 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) - 30 °C (NBC)

Outdoor Air Flow (fa) 12000 L/s

Maximum Temperature Difference (T1-T2) 52 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

$$= 12000 \times (52) \times 4.345$$

$$= 2711280 \text{ kJ/h}$$

Operating hours per week 168 h (1)

Average operating hours per day $= \frac{(1)}{7} =$ 24 h (2)

$$\text{Annual Sensible Heat AHs} = \frac{Qs \times DDh \times (2)}{(T1-T2) \times 1000}$$

$$= \frac{2711280 \times 5451 \times 24}{52 \times 1000}$$

$$= 6821163 \text{ MJ} \qquad (3)$$

Indoor humidity 50 %

Indoor humidity factor (H1) 8 g/kg (Figure 4)

Outdoor humidity 60 % (TYPICAL FOR WINTER)

Outdoor humidity factor (H2) 1 g/kg (Figure 4) (BY EXTRAPOLATION)

Maximum humidification heat, QL = fa x (H1-H2) x 10.84

$$= 12000 \times (8-1) \times 10.84$$

$$= 910560 \text{ kJ/h}$$

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: RETROFIT By: MBE

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1\text{-}T2) \times 1000}$

$= \dfrac{910560 \times 5451 \times (24/2)}{52 \times 1000}$

= 1145414 MJ (4)

Total annual heat = (3) + (4)

= 7966577 MJ (5)

Fuel type #6 FUEL OIL

Fuel cost/unit $0.26/L (6)

Fuel heat value/unit 42.3 MJ/L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

$= \dfrac{7966577 \times 0.26}{42.3 \times 0.75}$

= $ 65290 (9)

Reduction in operating hours/week N/A h (10)

Initial operating hours/week N/A h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

=

= $ N/A /yr

Heating Of Outside Air
Average Temperature Method
Worksheet 10-9
(Page 1 of 1)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: RETROFIT By: MBE

*No. of months with monthly mean temperature below 18°C 11 (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= 11x730 = 8030 h (2)

Average temperature rise 5 °C (3)

Hours of operation/week 168 h (4)

Hours of operation during heating season = (4) x (1) x 4.345

= 8030 h (5)

Outdoor air flow (fa) 12000 L/s

Average heating, Q = fa x (3) x 4.345

= 12000 x 5 x 4.345

= 260700 kJ/h (6)

Annual energy, AH = $\frac{(6) \times (5)}{1000}$

= $\frac{260700 \times 8030}{1000}$

= 2093421 MJ (7)

Fuel type #6 FUEL OIL

Fuel cost/unit $0.26/L (8)

Fuel heat value/unit 42.3 MJ/L (APPENDIX C) (9)

Heating system efficiency 0.75 (10)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost = $\frac{(7) \times (8)}{(9) \times (10)}$

= $\frac{2093421 \times 0.26}{42.3 \times 0.75}$

= $ 17156

*Can be determined from Environment Canada weather data for specific locations.

Fan Energy Cost

Worksheet 10-3

(Page 1 of 1)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 12 000 L/s

Total fan pressure (P) 0.15 + 0.25 = 0.40 kPa

Fan power, Wf = $\frac{fa \times P}{750}$

= $\frac{12000 \times 0.40}{750}$

= 6.4 kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 6.4 × 8760 × 0.05

= $ 2803 /yr

Pump Energy Cost
Worksheet 10-4
(Page 1 of 1)

Company: WORKED EXAMPLE #1 Date: 85/09/10

Location: RETROFIT By: MBE

Water flow rate (fw) 32 L/s

Total pump pressure (P) 3.8 kPa

Pump Power, Wp = $\frac{fw \times P}{500}$

= $\frac{32 \times 3.8}{500}$

= 0.24 kW (1)

operation time 11 x 730 = 8030 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost = (1) x (2) x (3)

= 0.24 x 8030 x 0.05

= $ 96 /yr

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #2 Date: 85/09/10

Location: RETROFIT By: MBE

Degree Days Below 18°C (DDh) 4225 (Table 2)

Indoor Temperature (T1) 20 °C

Minimum Outside Temperature (T2) -18 °C (NBC)

Outdoor Air Flow (fa) 950 L/s

Maximum Temperature Difference (T1-T2) 38 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

= 950 × 38 × 4.345

= 156855 kJ/h

Operating hours per week 168 h (1)

Average operating hours per day $= \frac{(1)}{7} =$ 24 h (2)

Annual Sensible Heat, AHs $= \frac{\text{Qs x DDh x (2)}}{\text{(T1 - T2) x 1000}}$

$= \frac{156855 \times 4225 \times 24}{38 \times 1000}$

= 418555 MJ (3)

Indoor humidity N/A %

Indoor humidity factor (H1) N/A g/kg (Figure 4)

Outdoor humidity N/A %

Outdoor humidity factor (H2) N/A g/kg (Figure 4)

Maximum humidification heat, QL = fa x (H1 - H2) x 10.84

= ____________________

= ____________________ kJ/h

Heating Of Outside Air
Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE #2 Date: 85/09/10

Location: RETROFIT By: MBE

$$\text{Annual humidification heat, AHL} = \frac{QL \times DDh \times \frac{(2)}{2}}{(T1 - T2) \times 1000}$$

= ______________

= N/A MJ (4)

Total annual heat = (3) + (4)

= 418 555 MJ (5)

Fuel type #2 FUEL OIL

Fuel cost/unit $0.40/L (6)

Fuel heat value/unit 38.68 MJ/L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

$$\text{Annual cost} = \frac{(5) \times (6)}{(7) \times (8)}$$

$$= \frac{418\,555 \times 0.40}{38.68 \times 0.75}$$

= $ 5771 (9)

Reduction in operating hours/week 6 x 7 = 42 h (6 days at 7h/day) (10)

Initial operating hours/week 168 h (11)

$$\text{Annual cost saving} = \frac{(9) \times (10)}{(11)}$$

$$= \frac{5771 \times 42}{168}$$

= $ 1443 /yr

Heating Of Outside Air
Average Temperature Method

Worksheet 10-9
(Page 1 of 1)

Company: WORKED EXAMPLE #4 Date: 85/09/10

Location: RETROFIT By: MBE

*No. of months with monthly mean temperature below 18°C 9 (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= 9 x 730 = 6570 h (2)

Average temperature rise 5 °C (3)

Hours of operation/week 50 h (4)

Hours of operation during heating season = (4) x (1) x 4.345

= 1955 h (5)

Outdoor air flow (fa) 2400 L/s

Average heating, Q = fa x (3) x 4.345

= 2400 x 5 x 4.345

= 52140 kJ/h (6)

Annual energy, AH = $\frac{(6) \times (5)}{1000}$

= $\frac{52140 \times 1955}{1000}$

= 101934 MJ (7)

Fuel type #6 FUEL OIL

Fuel cost/unit $0.26/L (8)

Fuel heat value/unit 42.3 MJ/L (APPENDIX C) (9)

Heating system efficiency 0.75 (10)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost = $\frac{(7) \times (8)}{(9) \times (10)}$

= $\frac{101934 \times 0.26}{42.3 \times 0.75}$

= $ 835

*Can be determined from Environment Canada weather data for specific locations.

Fan Energy Cost
Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #4 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 2400 L/s

Total fan pressure (P) 1.5 kPa

Fan power, $Wf = \frac{fa \times P}{750}$

$= \frac{2400 \times 1.5}{750}$

$= 4.8$ kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost \$ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

$= 4.8 \times 8760 \times 0.05$

= \$ 2120 /yr

Fan Energy Cost

Worksheet 10-3

(Page 1 of 1)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 15 000 L/s (EXIST. CENT. SYST.)

Total fan pressure (P) 1.12 kPa

Fan power, Wf = $\frac{fa \times P}{750}$

= $\frac{15000 \times 1.12}{750}$

= 22.4 kW (1)

Reduced operating time 108 × 52 = 5616 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 22.4 × 5616 × 0.05

= $ 6290 /yr

Fan Energy Cost
Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 13 000 L/s (PROPOSED CENT. SYSTEM)

Total fan pressure (P) 1.12 kPa

Fan power, Wf = $\frac{fa \times P}{750}$

$= \frac{13000 \times 1.12}{750}$

= 19.41 kW (1)

Reduced operating time 60 × 52 = 3120 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 19.41 × 3120 × 0.05

= $ 3 028 /yr

Fan Energy Cost

Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #5 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 2000 L/s (PROPOSED LOCAL SYST.)

Total fan pressure (P) 0.56 kPa

$$\text{Fan power, Wf} = \frac{\text{fa} \times \text{P}}{750}$$

$$= \frac{2000 \times 0.56}{750}$$

= 1.49 kW (1)

operating time 14 × 52 = 728 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost = (1) x (2) x (3)

= 1.49 × 728 × 0.05

= $ 54 /yr

Heating Of Outside Air
Worksheet 10-1
(Page 1 of 2)

Company: WORKED EXAMPLE #6 Date: 85/09/10

Location: RETROFIT By: MBE

Degree Days Below 18°C (DDh) 4538 (Table 2)

Indoor Temperature (T1) 22 °C

Minimum Outside Temperature (T2) -26 °C (NBC)

Outdoor Air Flow (fa) 3 400 L/s

Maximum Temperature Difference (T1-T2) 48 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

$= 3400 \times 48 \times 4.345$

$= 709104$ kJ/h

Operating hours per week 50 h (1)

Average operating hours per day $= \frac{(1)}{7} =$ 7.14 h (2)

Annual Sensible Heat AHs $= \frac{Qs \times DDh \times (2)}{(T1\text{-}T2) \times 1000}$

$= \frac{709104 \times 4538 \times 7.14}{48 \times 1000}$

$= 478\,665$ MJ (3)

Indoor humidity 35 %

Indoor humidity factor (H1) 5.8 g/kg (Figure 4)

Outdoor humidity 60 % (TYPICAL FOR WINTER)

Outdoor humidity factor (H2) 1 g/kg (Figure 4)
(BY EXTRAPOLATION)

Maximum humidification heat, QL = fa x (H1-H2) x 10.84

$= 3400 \times (5.8 - 1) \times 10.84$

$= 176\,909$ kJ/h

Heating Of Outside Air

Worksheet 10-1

(Page 2 of 2)

Company: WORKED EXAMPLE #6

Date: 85/09/10

Location: RETROFIT

By: MBE

$$\text{Annual humidification heat, AHL} = \frac{\text{QL x DDh x } \frac{(2)}{2}}{\text{(T1-T2) x 1000}}$$

$$= \frac{176909 \times 4538 \times (7.14/2)}{48 \times 1000}$$

= 59709 MJ (4)

Total annual heat = (3) + (4)

= 538374 MJ (5)

Fuel type # 6 FUEL OIL

Fuel cost/unit $0.26/L (6)

Fuel heat value/unit 42.3 MJ/L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

$$\text{Annual cost} = \frac{(5) \times (6)}{(7) \times (8)}$$

$$= \frac{538374 \times 0.26}{42.3 \times 0.75}$$

= $ 4412 (9)

Reduction in operating hours/week N/A h (10)

Initial operating hours/week N/A h (11)

$$\text{Annual cost saving} = \frac{(9) \times (10)}{(11)}$$

= ______

= $ N/A /yr

Annual Heat Transfer Through A Building Enclosure
Degree Day Method

Worksheet 10-7

(Page 1 of 2)

Company: WORKED EXAMPLE #6 Date: 85/09/10

Location: RETROFIT By: MBE

Degree Days Below 18°C (DD) 4538 (Table 2)

Warm Side Temperature (T1) 22 °C

Cold Side Temperature (T2) −26 °C

Wall coef. of transmission (Uw) 0.5 $W/(m^2 \cdot °C)$

Window coef. of transmission (Ug) 3.3 $W/(m^2 \cdot °C)$

Roof coef. of transmission (Ur) 0.6 $W/(m^2 \cdot °C)$

Gross wall area 2500 m^2 (1)

Window area (Ag) 600 m^2

Net wall area, Aw = (1) − Ag

= 1900 m^2

Roof area (Ar) 850 m^2

Enclosure heat loss = [(Aw x Uw) + (Ag x Ug) + (Ar x Ur)] x (T1-T2) x 3.6

$= [(1900 \times 0.5) + (600 \times 3.3) + (850 \times 0.6)] \times 48 \times 3.6$

$= (950 + 1980 + 510) \times 48 \times 3.6$

= 594 432 kJ/h (2)

(Av'G)
Infiltration rate (CH) 0.25 air changes/h (Table 4)

Room volume (V) 6100 m^3

Annual Heat Transfer Through A building Enclosure
Degree Day Method
Worksheet 10-7
(Page 2 of 2)

Company: WORKED EXAMPLE #6 Date:

Location: RETROFIT By: MBE

Infiltration heat loss $= \frac{V \times CH}{3.6} \times (T1\text{-}T2) \times 4.345$

$= \frac{(6100 \times 0.25)}{3.6} \times 48 \times 4.345$

= 88348 kJ/h (3)

Total heat loss, Q = (2) + (3)

= 594432 + 88348 = 682780 kJ/h

Annual Heat Flow Through Component, AH $= \frac{Q \times DD \times 18}{(T1 - T2) \times 1000}$

$= \frac{682780 \times 4538 \times 18}{48 \times 1000}$

= 1161921 kJ (4)

Fuel Type #6 FUEL OIL

Fuel Cost/Unit $0.26/L (5)

Fuel Heat Value/Unit 42.3 MJ/L (APPENDIX C) (6)

Heating System Efficiency 0.75 (7)
(If unknown use 0.75 for oil or gas, 1.0 for electricity).

Annual fuel consumption $= \frac{(4)}{(6) \times (7)}$

$= \frac{1161921}{42.3 \times 0.75}$

= 36625 (8)

Annual Costs = (8) x (5)

= 36625 x 0.26

= $ 9523

Fan Energy Cost
Worksheet 10-3
(Page 1 of 1)

Company: WORKED EXAMPLE #6

Date: 85/09/10

Location: RETROFIT

By: MBE

Air flow rate (fa) 21600 L/s

Total fan pressure (P) 1.5 kPa

} EXIST. SYSTEM

Fan power, $Wf = \frac{fa \times P}{750}$

$= \frac{21600 \times 1.5}{750}$

= 43.2 kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 43.2 × 8760 × 0.05

= $ 18922 /yr

Fan Energy Cost

Worksheet 10-3

(Page 1 of 1)

Company: WORKED EXAMPLE # 6 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 16200 L/s

Total fan pressure (P) 1.03 kPa

PROPOSED SYSTEM

Fan power, Wf = $\frac{fa \times P}{750}$

= $\frac{16200 \times 1.03}{750}$

= 22.2 kW (1)

Reduced operating time 8760 h/yr (2)

Unit energy cost \$ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 22.2 × 8760 × 0.05

= \$ 9724 /yr

Heating Of Outside Air

Worksheet 10-1

(Page 1 of 2)

Company: WORKED EXAMPLE #11 Date: 85/09/10

Location: RETROFIT By: MBE

Degree Days Below ~~10°C~~ 0°C (DDh) 896 (Table 2)

Indoor Temperature (T1) 5 °C

Minimum Outside Temperature (T2) -27 °C (NBC)

Outdoor Air Flow (fa) 5500 L/s

Maximum Temperature Difference (T1-T2) 32 °C

Maximum Sensible Heating, Qs = fa x (T1-T2) x 4.345

= 5500 x 32 x 4.345

= 764720 kJ/h

Operating hours per week 168 h (1)

Average operating hours per day $= \frac{(1)}{7} =$ 24 h (2)

Annual Sensible Heat AHs $= \frac{Qs \times DDh \times (2)}{(T1\text{-}T2) \times 1000}$

$= \frac{764720 \times 896 \times 24}{32 \times 1000}$

= 513892 MJ (3)

Indoor humidity N/A %

Indoor humidity factor (H1) N/A g/kg (Figure 4)

Outdoor humidity N/A %

Outdoor humidity factor (H2) N/A g/kg (Figure 4)

Maximum humidification heat, QL = Fa x (H1-H2) x 10.84

= __________

= N/A kJ/h

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: WORKED EXAMPLE #11 Date: 85/09/10

Location: RETROFIT By: MBE

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1\text{-}T2) \times 1000}$

= ______

= N/A MJ (4)

Total annual heat = (3) + (4)

= 513 892 MJ (5)

Fuel type #2 FUEL OIL

Fuel cost/unit $0.40/L (6)

Fuel heat value/unit 38.68 MJ/L (APPENDIX C) (7)

Heating system efficiency 0.75 (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

$= \dfrac{513.892 \times 0.40}{38.68 \times 0.75}$

= $ 7086 (9)

Reduction in operating hours/week 140 h (10)

Initial operating hours/week 168 h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

$= \dfrac{7086 \times 140}{168}$

= $ 5905 /yr

Fan Energy Cost

Worksheet 10-3

(Page 1 of 1)

Company: WORKED EXAMPLE #11 Date: 85/09/10

Location: RETROFIT By: MBE

Air flow rate (fa) 5500 L/s

Total fan pressure (P) 0.35 kPa

Fan power, Wf = $\frac{\text{fa x P}}{750}$

$$= \frac{5500 \times 0.35}{750}$$

= 2.57 kW (1)

Reduced operating time 7280 h/yr (2)

Unit energy cost $ 0.05 /kWh (3)

Cost saving, = (1) x (2) x (3)

= 2.57 × 7280 × 0.05

= $ 935 /yr

APPENDICES

Glossary

Activated Carbon — a form of carbon which has been processed to make it capable of adsorbing odors.

Air Changes — a method of expressing the amount of air flow into or out of a building or room in terms of the number of building volumes or room volumes of air exchanged per hour.

Air Cleaner — a device used to remove airborne impurities.

Air-Conditioning — the process of treating air to simultaneously control its temperature, humidity, cleanliness, and distribution to meet the comfort requirements of the occupants of a conditioned space.

Air-Conditioning Unit – an assembly of equipment for the treatment of air to simultaneously control its temperature, humidity, cleanliness, and distribution to meet the requirements of a conditioned space.

Aspirator — a device which uses a high velocity jet of liquid or air through a shaped nozzle to induce a secondary flow of liquid or air through the device.

Balance Temperature — the outside temperature at which the heat losses from a building space are equal to the internal heat gains.

Blowdown — a term used for the controlled release of pressurized water.

Boiler Blowdown — the controlled release of water from a boiler to limit the build-up of solids.

Chilled Liquid — a cooled liquid, usually water, circulated between a refrigeration system and terminal cooling devices.

Condensate — liquid formed by the condensation of a vapor.

Condensation — process of changing a vapor into liquid by extracting heat.

Convection — heat transfer by the movement of air or a liquid.

Cooling Tower — a device for removing heat energy from a liquid system to outdoor air.

Daily Mean Temperature — the mean temperature for a day is reached by adding the minimum and the maximum together and dividing that number by two.

Damper — a device for regulating the flow of air or other fluid.

Degree Day — a totalization of the difference between the mean daily outdoor dry bulb temperature and a reference temperature. For example, ten heating degree days below 18°C is equivalent to one day at 8°C or ten days at 17°C. Degree days may be referred to as either "heating" or "cooling".

Direct Expansion — the evaporation of a refrigerant liquid in an air cooling coil.

Dry Bulb Temperature — the temperature of air measured with a dry sensing device in such a way as to avoid the effects of radiation and evaporative cooling.

Economizer Control — a control system which compares outside air temperature with the space temperature and automatically provides the optimum use of outdoor air for free cooling.

Evaporative Cooling — the cooling effect caused by water absorbing its latent heat of vaporization during evaporation.

Exfiltration — air flow outward through a building enclosure.

Filter — a device to remove solid material from a fluid.

Flash Steam — steam produced by water which is above the saturation temperature for a given pressure.

Free Cooling — cooling with outdoor air which is at or below the supply air temperature required for cooling a building space.

Heat Exchanger — a device to transfer heat between two physically separated fluids.

Heat of Fusion — the latent heat involved in changing between the solid and liquid state.

Heat of Vaporization — the latent heat involved in changing between the liquid and gaseous state.

Humidify — to add water vapor to air.

Humidity Factor — the weight of water vapor per unit weight of dry air.

Inertia — the property of matter by virtue of which a mass persists in its state of rest or uniform motion until acted upon by some external force.

Infiltration — air flowing inward as through a building enclosure.

Interior Zone — a building area that has no exterior walls and is completely surrounded by conditioned space.

Latent Heat — the transfer of heat energy required to produce a change of state in a substance from a solid to a liquid or from a liquid to a gas (vapor).

Latent Heat of Vaporization — the amount of heat required to change one kilogram of boiling water to steam at a given pressure.

Load Analyzer — a device which monitors the output signals from a number of zone thermostats and transmits the signal from the zone requiring the most heating and the zone requiring the most cooling.

Monthly Mean Temperature — the monthly average of daily mean temperatures.

Perimeter Zone — a building area adjacent to an exterior wall.

Psychrometric Chart — a chart which illustrates the relationship of air-water-vapor mixtures with regard to dry and wet bulb temperature, relative humidity, humidity factor, sensible heat, latent heat, total heat and other properties.

Radiation — the transfer of energy in the form of low frequency light rays. The energy is transferred directly from the emitting surface to the receiving surfaces and can be transmitted through a vacuum.

Radiant Heat — heat transferred by radiation.

Saturated Air — air containing the maximum amount of water vapor that can exist in gaseous form at the particular temperature and pressure.

Saturated Steam — steam at the *saturation temperature* for the existing pressure.

Saturation Temperature (of a Gas) — the temperature at which further loss of heat energy will cause condensation of a gas to a liquid at the existing pressure.

Saturation Temperature (of a Liquid) — the temperature at which further gain of heat energy will cause evaporation of a liquid to a gas at the existing temperature.

Sensible Heat — the heat required to produce a change in temperature in a substance without creating a change of state.

Stack Effect — the buoyant effect of air at a higher temperature and lower density than the air it is displacing.

Static Pressure — the pressure exerted at a 90 degree angle to the direction of flow in a moving air stream.

Stratification — the separation of air or liquid into layers, or strata, of different temperatures and densities.

Thermal Conductivity — the rate of heat flow through a unit area and unit thickness of a homogeneous material under constant flow conditions.

Thermodynamics — the science of heat energy and its transformation.

Thermostat —an automatic control device actuated by temperature and producing a signal used to control heating or cooling devices.

Total Pressure — the sum of velocity and static pressure.

Velocity Pressure — the impact pressure of a moving air stream exerted in the direction of flow.

Ventilation — the process of supplying and removing air by natural or mechanical means to and from a building space.

Water Hammer — rapid pressure changes in a pipe or pressure vessel caused by sudden changes in velocity of a liquid. It may be caused by such conditions as sudden changes in velocity of flow, collapse of vapor bubbles in a liquid, or pockets of liquid in a high velocity steam flow arriving at a restriction such as a control valve.

Wet Bulb Temperature — the temperature measured by a thermometer whose sensing bulb is covered with a wet cloth and exposed to a moving air stream. The value is affected by the moisture content of the air stream. It is used in conjunction with a psychrometric chart to determine the air stream relative humidity.

TYPICAL COEFFICIENTS OF HEAT TRANSFER (U)

TABLE 1

Component	U [W/(m²·°C)]
Wood frame wall, uninsulated	1.1
Wood frame wall with 90 mm batt insulation	0.5
Wood frame wall with 150 mm batt insulation	0.3
Masonry cavity wall with 50 mm polystyrene foam insulation	0.5
Masonry cavity wall with 100 mm polystyrene foam insulation	0.3
Built-up roof with 50 mm rigid glass fibre insulation on concrete deck	0.6
Single glass	6.2
Double insulating glass	3.3
Triple insulating glass	2.2

DEGREE DAYS FOR SELECTED LOCATIONS

TABLE 2

(Averages 1951 to 1980 Compiled by Environment Canada
Based on Mean Daily Temperatures)

Location	Annual Degree Days Below 18°C	Annual Degree Days Below 0°C	Annual Degree Days Above 18°C
Vancouver, B.C	3031	41	36
Whitehourse, Yukon	6988	2171	6
Yellowknife, N.W.T.	8530	3698	28
Edmonton, Alta.	5990	1693	31
Saskatoon, Sask.	6063	1971	111
Winnipeg, Man.	5923	1940	178
Sudbury, Ont.	5451	1418	123
Toronto, Ont.	4144	634	347
Windsor, Ont.	3622	445	391
Montreal, Que.	4538	957	251
Quebec, Que.	5165	1198	123
Bagotville, Que.	5805	1603	93
Fredericton, N.B.	4740	896	143
Halifax, N.S.	4425	604	88
Charlottetown, P.E.I.	4689	738	89
St. John's, Nfld.	4824	473	29

TYPICAL MAXIMUM SUMMER HEAT GAIN RATES

(kJ/h·m^2)

TABLE 3

(Compiled using ASHRAE data)

Building Component	**City (Latitude)** **Windsor Ont. (42°)**	 **Saint John N.B. (45°)**	 **Vancouver B.C. (49°)**	 **Edmonton Alta. (54°)**
Flat Roof (U=0.6): (18.00h)	38	26	25	29
Masonry Cavity Walls (U=0.5):				
• South (18.00h)	21	14	16	20
• West (20.00h)	19	10	10	13
• North (20.00h)	9	1	1	2
• East (13.00h)	12	3	3	6
Untinted Double Glass (U=3.3):				
• South (13.00h)	351	420	490	534
• West (16.00h)	1517	1585	1572	1653
• North (18.00h)	175	180	173	181
• East (10.00h)	190	303	301	316

TYPICAL INFILTRATION RATES

TABLE 4

Room Exposure	Infiltration In air Changes Per Hour
One exterior wall, no windows or sealed, double glazed windows	0.25
One exterior wall, with openable, weatherstripped windows	0.5
One exterior wall with openable, non-weatherstripped windows or exterior doors	1.0
Two exterior walls with sealed, double glazed windows	0.5
Two exterior walls with openable, weatherstripped windows	0.7
Two exterior walls with openable, non-weatherstripped windows or exterior doors	1.5
Entrance halls	2.0 (+)

NOTE: Room volumes to which the above rates are applied should be based on a room depth of no more than 5 metres from the exterior wall. For rooms having a greater depth, calculate the infiltration for the first 5 metres of depth only.

RATES OF HEAT GAIN FROM OCCUPANTS OF CONDITIONED SPACES

TABLE 5

Degree of Activity	Typical Application	Sensible kJ/h	Latent kJ/h
Seated at rest	Theater, movie	220	150
Seated, very light work, writing	Offices, hotels, apts	240	200
Seated, eating	Restaurant	270	340
Seated, light work, typing	Offices, hotels, apts	270	270
Standing, light work or walking slowly	Retail store, bank	330	340
Light bench work	Factory	360	460
Walking, light machine work	Factory	360	730
Bowling	Bowling alley	360	650
Moderate dancing	Dance hall	430	920
Heavy work, machine work, lifting	Factory	600	1090
Heavy work, athletics	Gymnasium	670	1230

CHARACTERISTICS OF COMMON FILTERS

TABLE 6

Filter Type	ASHRAE Weight Arrestance %	ASHRAE Dust Spot Efficiency %	MIL-STD 282 DOP Efficiency %	Face Velocity Drop m/s	Average Pressure Pa
Viscous impingement panel filters 25-45 mm thick	50-75	5-15	N.A.	1-4	25-125
Medium efficiency bag filters	70-95	15-90	0-55	1-3.8	25-125
High efficiency bag filters	N.A.	90-98	75-90	1-3.8	125-250
Very high efficiency extended surface (HEPA) filters	N.A.	N.A.	95-99.999	1.3	125-250
Electronic air	N.A.	90	N.A.	1.5-2.5	35-65

RANGE OF CAPTURE VELOCITIES
TABLE 7

Condition of Contaminant Dispersion	Examples	Capture (Control) Velocity fpm	m/s
Released with essentially no velocity into still air	Evaporation from tanks, degreasing, plating	50 to 100	0.25 to 0.5
Released at low velocity into moderately still air	Container filling, low speed conveyor transfers, welding	100 to 200	0.5 to 10
Active generation into zone of rapid air motion	Barrel filling, chute loading of conveyors, crushing, cool shakeout	200 to 500	1.0 to 2.5
Released at high velocity into moderately still air	Grinding, abrasive blasting, tumbling, hot shakeout	500 to 2000	2.5 to 10.0

Lower End of Range	Upper End of Range
1. Room air currents or favorable to capture	1. Disturbing room air currents
2. Contaminants of low toxicity or of nuisance value only	2. Contaminants of high toxicity.
3. Intermittent, low production.	3. High production, heavy use.
4. Large hood-large air mass in motion	4. Small hood-local control only.

COMMON CONVERSIONS

1 barrel (35 Imp gal) (42 US gal) = 159.1 litres
1 gallon (Imp) = 1.20094 gallon (US)
1 horsepower (boiler) = 9809.6 watts
1 horsepower = 2545 Btu/hour
1 horsepower = 0.746 kilowatts
1 joule = 1 N-m
Kelvin = (°C + 273.15)

1 kilowatt·hour = 3600 kilojoules
1 Newton = 1 kg-m/s^2
1 therm = 10^5 Btu
1 ton (refrigerant) = 12002.84 Btu/hour
1 ton (refrigerant) = 3516.8 watts
1 watt = 1 joule/second
Rankine = (°F + 459.67)

Cubes

$1\ yd^3 = 27\ ft^3$
$1\ ft^3 = 1728\ in^3$
$1\ cm^3 = 1000\ mm^3$
$1\ m^3 = 10^6\ cm^3$
$1\ m^3 = 1000\ L$

Squares

$1\ yd^2 = 9\ ft^2$
$1\ ft^2 = 144\ in^2$
$1\ cm^2 = 100\ mm^2$
$1\ m^2 = 10000\ cm^2$

SI PREFIXES

Prefix	Symbol	Magnitude	Factor
tera	T	1 000 000 000 000	10^{12}
giga	G	1 000 000 000	10^9
mega	M	1 000 000	10^6
kilo	k	1 000	10^3
hecto	h	100	10^2
deca	da	10	10^1
deci	d	0.1	10^{-1}
centi	c	0.01	10^{-2}
milli	m	0.001	10^{-3}
micro	u	0.000 001	10^{-6}
nano	n	0.000 000 001	10^{-9}
pica	p	0.000 000 000 001	10^{-12}

UNIT CONVERSION TABLES
METRIC TO IMPERIAL

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
amperes/square centimetre	A/cm^2	amperes/square inch	A/in^2	6.452
Celsius	°C	Fahrenheit	°F	(°C × 9/5) + 32
centimetres	cm	inches	in	0.3937
cubic centimetres	cm^3	cubic inches	in^3	0.06102
cubic metres	m^3	cubic foot	ft^3	35.314
grams	g	ounces	oz	0.03527
grams	g	pounds	lb	0.0022
grams/litre	g/L	pounds/cubic foot	lb/ft^3	0.06243
joules	J	Btu	Btu	9.480×10^{-4}
joules	J	foot-pounds	ft-lb	0.7376
joules	J	horsepower-hours	hp-h	3.73×10^{-7}
joules/metre, (Newtons)	J/m, N	pounds	lb	0.2248
kilograms	kg	pounds	lb	2.205
kilograms	kg	tons (long)	ton	9.842×10^{-4}
kilograms	kg	tons (short)	tn	1.102×10^{-3}
kilometres	km	miles (statute)	mi	0.6214
kilopascals	kPa	atmospheres	atm	9.87×10^{-3}
kilopascals	kPa	inches of mercury (@ 32°F)	in Hg	0.2953
kilopascals	kPa	inches of water (@ 4°C)	in H_2O	4.0147
kilopascals	kPa	pounds/square inch	psi	0.1450
kilowatts	kW	foot-pounds/second	ft-lb/s	737.6
kilowatts	kW	horsepower	hp	1.341
kilowatt-hours	kWh	Btu	Btu	3413
litres	L	cubic foot	ft^3	0.03531
litres	L	gallons (Imp)	gal (Imp)	0.21998
litres	L	gallons (US)	gal (US)	0.2642
litres/second	L/s	cubic foot/minute	cfm	2.1186
lumen/square metre	lm/m^2	lumen/square foot	lm/ft^2	0.09290
lux, lumen/square metre	lx, lm/m^2	footcandles	fc	0.09290
metres	m	foot	ft	3.281
metres	m	yard	yd	1.09361
parts per million	ppm	grains/gallon (Imp)	gr/gal (Imp)	0.07
parts per million	ppm	grains/gallon (US)	gr/gal (US)	0.05842
permeance (metric)	PERM	permeance (Imp)	perm	0.01748
square centimetres	cm^2	square inches	in^2	0.1550
square metres	m^2	square foot	ft^2	10.764
square metres	m^2	square yards	yd^2	1.196
tonne (metric)	t	pounds	lb	2204.6
watt	W	Btu/hour	Btu/h	3.413
watt	W	lumen	lm	668.45

UNIT CONVERSION TABLES
IMPERIAL TO METRIC

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
ampere/in^2	A/in^2	ampere/cm^2	A/cm^2	0.1550
atmospheres	atm	kilopascals	kPa	101.325
British Thermal Unit	Btu	joules	J	1054.8
Btu	Btu	kilogram-metre	kg-m	107.56
Btu	Btu	kilowatt-hour	kWh	2.928×10^{-4}
Btu/hour	Btu/h	watt	W	0.2931
calorie, gram	cal or g-cal	joules	J	4.186
chain	chain	metre	m	20.11684
cubic foot	ft^3	cubic metre	m^3	0.02832
cubic foot	ft^3	litre	L	28.32
cubic foot/minute	cfm	litre/second	L/s	0.47195
cycle/second	c/s	Hertz	Hz	1.00
Fahrenheit	°F	Celsius	°C	(°F-32)/1.8
foot	ft	metre	m	0.3048
footcandle	fc	lux, lumen/ square metre	lx, lm/m^2	10.764
footlambert	fL	candela/square metre	cd/m^2	3.42626
foot-pounds	ft-lb	joule	J	1.356
foot-pounds	ft-lb	kilogram-metres	kg-m	0.1383
foot-pounds/second	ft-lb/s	kilowatt	kW	1.356×10^{-3}
gallons (Imp)	gal (Imp)	litres	L	4.546
gallons (US)	gal (US)	litres	L	3.785
grains/gallon (Imp)	gr/gal (Imp)	parts per million	ppm	14.286
grains/gallon (US)	gr/gal (US)	parts per million	ppm	17.118
horsepower	hp	watts	W	745.7
horsepower-hours	hp-h	joules	J	2.684×10^6
inches	in	centimetres	cm	2.540
inches of Mercury (@ 32°F)	in Hg	kilopascals	kPa	3.386
inches of water (@ 4°C)	in H_2O	kilopascals	kPa	0.2491

UNIT CONVERSION TABLES
IMPERIAL TO METRIC (cont'd)

FROM	SYMBOL	TO	SYMBOL	MULTIPLY BY
lamberts	* L	candela/square metre	cd/m^2	3.183
lumen/square foot	lm/ft^2	lumen/square metre	lm/m^2	10.76
lumen	lm	watt	W	0.001496
miles (statute)	mi	kilometres	km	1.6093
ounces	oz	grams	g	28.35
perm (at 0°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.721×10^{-11}
perm (at 23°C)	perm	kilogram per pascal-second-square metre	$kg/Pa\text{-}s\text{-}m^2$ (PERM)	5.745×10^{-11}
perm-inch (at 0°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4532×10^{-12}
perm-inch (at 23°C)	perm. in.	kilogram per pascal-second-metre	kg/Pa-s-m	1.4593×10^{-12}
pint (Imp)	pt	litre	L	0.56826
pounds	lb	grams	g	453.5924
pounds	lb	joules/metre, (Newtons)	J/m, N	4.448
pounds	lb	kilograms	kg	0.4536
pounds	lb	tonne (metric)	t	4.536×10^{-4}
pounds/cubic foot	lb/ft^3	grams/litre	g/L	16.02
pounds/square inch	psi	kilopascals	kPa	6.89476
quarts	qt	litres	L	1.1365
slug	slug	kilograms	kg	14.5939
square foot	ft^2	square metre	m^2	0.09290
square inches	in^2	square centimetres	cm^2	6.452
square yards	yd^2	square metres	m^2	0.83613
tons (long)	ton	kilograms	kg	1016
tons (short)	tn	kilograms	kg	907.185
yards	yd	metres	m	0.9144

* "L" as used in Lighting

4138T/T33(-2)

The following typical values for conversion factors may be used when actual data are unavailable. The MJ and Btu equivalencies are heats of combustion. Hydrocarbons are shown at the higher heating value, wet basis. Some items listed are typically feedstocks, but are included for completeness and as a reference source. The conversion factors for coal are approximate since the heating value of a specific coal is dependent on the particular mine from which it is obtained.

ENERGY TYPE	METRIC	IMPERIAL
COAL		
— metallurgical	29,000 megajoules/tonne	25.0×10^6 Btu/ton
— anthracite	30,000 megajoules/tonne	25.8×10^6 Btu/ton
— bituminous	32,100 megajoules/tonne	27.6×10^6 Btu/ton
— sub-bituminous	22,100 megajoules/tonne	19.0×10^6 Btu/ton
— lignite	16,700 megajoules/tonne	14.4×10^6 Btu/ton
COKE		
— metallurgical	30,200 megajoules/tonne	26.0×10^6 Btu/ton
— petroleum		
— raw	23,300 megajoules/tonne	20.0×10^6 Btu/ton
— calcined	32,600 megajoules/tonne	28.0×10^6 Btu/ton
PITCH	37,200 megajoules/tonne	32.0×10^6 Btu/ton
CRUDE OIL	38,5 megajoules/litre	5.8×10^6 Btu/bbl
No. 2 OIL	38.68 megajoules/litre	5.88×10^6 Btu/bbl $.168 \times 10^6$ Btu/IG
No. 4 OIL	40.1 megajoules/litre	6.04×10^6 Btu/bbl $.173 \times 10^6$ Btu/IG
No. 6 OIL (RESID. BUNKER C)		
@ 2.5% sulphur	42.3 megajoules/litre	6.38×10^6 Btu/bbl $.182 \times 10^6$ Btu/IG
@ 1.0% sulphur	40.5 megajoules/litre	6.11×10^6 Btu/bbl $.174 \times 10^6$ Btu/IG
@ .5% sulphur	40.2 megajoules/litre	6.05×10^6 Btu/bbl $.173 \times 10^6$ Btu/IG
KEROSENE	37.68 megajoules/litre	$.167 \times 10^6$ Btu/IG
DIESEL FUEL	38.68 megajoules/litre	$.172 \times 10^6$ Btu/IG
GASOLINE	36.2 megajoules/litre	$.156 \times 10^6$ Btu/IG
NATURAL GAS	37.2 megajoules/m^3	1.00×10^6 Btu/MCF
PROPANE	50.3 megajoules/kg 26.6 megajoules/litre	$.02165 \times 10^6$ Btu/lb $.1145 \times 10^6$ Btu/IG
ELECTRICITY	3.6 megajoules/kWh	$.003413 \times 10^6$ Btu/kWh

Heating Of Outside Air

Worksheet 10-1
(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Degree Days Below 18°C (DDh) ______________________ (Table 2)

Indoor Temperature (T1) ______________________ °C

Minimum Outside Temperature (T2) ______________________ °C (NBC)

Outdoor Air Flow (fa) ______________________ L/s

Maximum Temperature Difference (T1 - T2) ______________________ °C

Maximum Sensible Heating, Qs = fa x (T1 - T2) x 4.345

= ______________________

= ______________________ kJ/h

Operating hours per week ______________________ h (1)

Average operating hours per day = $\frac{(1)}{7}$ = ______________________ h (2)

Annual Sensible Heat, AHs = $\frac{\text{Qs x DDh x (2)}}{\text{(T1 - T2) x 1000}}$

= ______________________

= ______________________ MJ (3)

Indoor humidity ______________________ %

Indoor humidity factor (H1) ______________________ g/kg (Figure 4)

Outdoor humidity ______________________ %

Outdoor humidity factor (H2) ______________________ g/kg (Figure 4)

Maximum humidification heat, QL = fa x (H1 - H2) x 10.84

= ______________________

= ______________________ kJ/h

Heating Of Outside Air

Worksheet 10-1
(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Annual humidification heat, AHL $= \dfrac{QL \times DDh \times \frac{(2)}{2}}{(T1 - T2) \times 1000}$

= ______________________

= ______________________ MJ (4)

Total annual heat = (3) + (4)

= ______________________ MJ (5)

Fuel type ______________________

Fuel cost/unit ______________________ (6)

Fuel heat value/unit ______________________ (APPENDIX C) (7)

Heating system efficiency ______________________ (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost $= \dfrac{(5) \times (6)}{(7) \times (8)}$

= ______________________

= $______________________ (9)

Reduction in operating hours/week ______________________ h (10)

Initial operating hours/week ______________________ h (11)

Annual cost saving $= \dfrac{(9) \times (10)}{(11)}$

= ______________________

= $______________________ /yr

Outside Air Intake Rate

Worksheet 10-2
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Return air temperature (T1) ______________________ °C

Outdoor air temperature (T2) ______________________ °C

Mixed air stream temperature (T3) ______________________ °C

Mixed air stream flow rate (fm) ______________________ L/s

Outdoor air intake rate, fo $= \dfrac{\text{fm x (T1-T3)}}{\text{T1 - T2}}$

= ______________________

= ______________________ L/s

Fan Energy Cost

Worksheet 10-3

(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Air flow rate (fa) ______________________ L/s

Total fan pressure (P) ______________________ kPa

Fan power, $Wf = \frac{fa \times P}{750}$

= ______________________

= ______________________ kW (1)

Reduced operating time ______________________ h/yr (2)

Unit energy cost $ ______________________ /kWh (3)

Cost saving, = (1) x (2) x (3)

= ______________________

= $ ______________________ /yr

Pump Energy Cost
Worksheet 10-4
(Page 1 of 1)

Company: ______________________________ Date: ______________________________

Location: ______________________________ By: ______________________________

Water flow rate (fw) ______________________________ L/s

Total pump pressure (P) ______________________________ kPa

Pump Power, $Wp = \frac{fw \times P}{500}$

= ______________________________

= ______________________________ kW (1)

Reduced operation time ______________________________ h/yr (2)

Unit energy cost $ ______________________________ /kWh (3)

Cost Saving, = (1) x (2) x (3)

= ______________________________

= $ ______________________________ /yr

Cooling Of Outdoor Air — Degree Day Method

Worksheet 10-5

(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Degree days above 18°C (DDc) ______________________ (Table 2) (1)

Design outdoor conditions

Dry bulb temperature (T1) ______________________ °C (NBC)

Wet bulb temperature ______________________ °C (NBC)

Humidity factor (H1) ______________________ g/kg (Figure 4)

Indoor design conditions

Dry bulb temperature (T2) ______________________ °C

Relative humidity ______________________ %

Humidity factor (H2) ______________________ g/kg (Figure 4)

Outdoor air flow rate (fa) ______________________ L/s

Sensible cooling load, Qs = fa x (T1 - T2) x 4.345

= ______________________

= ______________________ kJ/h (2)

Latent cooling load, QL = fa x (H1 - H2) x 10.84

= ______________________

= ______________________ kJ/h (3)

Total cooling load = (2) + (3) = ______________________ kJ/h (4)

Cooling Of Outdoor Air — Degree Day Method

Worksheet 10-5

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Operating hours per week ______________________ h (5)

Average operating hours per day $= \frac{(5)}{7}$

= ______________________ h (6)

Annual outdoor air cooling, AC $= \frac{(4) \times (1) \times (6)}{(T1 - T2) \times 1000}$

= ______________________

= ______________________ MJ (7)

Unit energy cost $______________________ /kWh (8)

Energy consumption/GJ cooling ______________________ (9)
(if unknown use 80 kWh/GJ)

Annual cost $= \frac{(7) \times (8) \times (9)}{1000}$

= ______________________

= $ ______________________ (10)

Reduction in operating hours/week ______________________ h (11)

Initial operating hours/week ______________________ h (12)

Cost saving $= \frac{(10) \times (11)}{(12)}$

= ______________________

= $ ______________________ /yr

Cooling Of Internal Heat Gains

Worksheet 10-6

(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

*No. of months with monthly mean temperature above 18°C ______(Environment Canada) (1)

Number of lights ______________________ (2)

Power input per light______________________W (3)

Heat from lights = (2) x (3) x 3.6

= ______________________

= ______________________ kJ/h (4)

Number of people ______________________ (5)

Sensible heat gain per person ______________________kJ/h (Table 5) (6)

Latent heat gain per person ______________________kJ/h (Table 5) (7)

Total heat gain from people = (5) x [(6) + (7)]

= ______________________

= ______________________ kJ/h (8)

Process heat gain ______________________kJ/h (9)

Factors of duration:

fd (lights) __________ , fd (people) __________ , fd (proc) __________

Factors of utilization:

fu (lights) __________ , fu (people) __________ , fu (proc) __________

Total correction factors $\frac{\text{fd x fu}}{60}$:

__________ (10) __________ (11) __________ (12)

Cooling of Internal Heat Gains

Worksheet 10-6
(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Total cooling load (Qt) = [(4) x (10)] + [(8) x(11)] + [(9) x (12)]

= ______________________

= ______________________kJ/h (13)

Hours of operation per month ______________________(h) (14)

No. of months in operation ______________________ (15)

(For systems using outdoor air free cooling use value from (1), otherwise use actual operating months)

$$\text{Annual heat gain cooling, ACi} = \frac{(13) \times (14) \times (15)}{1000}$$

= ______________________

= ______________________ MJ/yr (16)

Unit energy cost $ ______________________/kWh (17)

Energy consumption/GJ cooling ______________________ (18)
(if unknown use 80 kWh/GJ)

$$\text{Annual cost} = \frac{(16) \times (17) \times (18)}{1000}$$

= ______________________

= $______________________ (19)

Reduction in operating period ______________________h (20)

Initial operating period ______________________h (21)

$$\text{Cost saving} = \frac{(19) \times (20)}{(21)}$$

= ______________________

= $ ______________________/yr

*Can be determined from Environment Canada weather data for specific locations.

Annual Heat Transfer Through A Building Enclosure
Degree Day Method
Worksheet 10-7
(Page 1 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Degree Days Below 18°C (DD) ______________________(Table 2)

Warm Side Temperature (T1) ______________________°C

Cold Side Temperature (T2) ______________________°C

Wall coef. of transmission (Uw) ______________________W/(m²·°C)

Window coef. of transmission (Ug) ______________________W/(m²·°C)

Roof coef. of transmission (Ur) ______________________W/(m²·°C)

Gross wall area ______________________m² (1)

Window area (Ag) ______________________m²

Net wall area, Aw = (1) − Ag

= ______________________m²

Roof area ______________________m²

Enclosure heat loss = [(Aw x Uw) + (Ag x Ug) + (Ar x Ur)] x (T1 - T2) x 3.6

= ______________________

= ______________________

= ______________________ kJ/h (2)

Infiltration rate (CH) ______________________air changes/h (Table 4)

Room volume (V) ______________________m^3

Annual Heat Transfer Through A Building Enclosure
Degree Day Method

Worksheet 10-7

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Infiltration heat loss = $\frac{V \times CH}{3.6} \times (T1 - T2) \times 4.345$

= ______________________

= ______________________ kJ/h (3)

Total heat loss, Q = (2) + (3)

= ______________________ kJ/h

Annual heat flow through component, AH = $\frac{Q \times DD \times 18}{(T1 - T2) \times 1000}$

= ______________________

= ______________________ kJ (4)

Fuel Type ______________________

Fuel Cost/Unit ______________________ (5)

Fuel Heat Value/Unit ______________________ (APPENDIX C) (6)

Heating System Efficiency ______________________ (7)

(If unknown use 0.75 for oil or gas, 1.0 for electricity).

Annual fuel consumption = $\frac{(4)}{(6) \times (7)}$

= ______________________

= ______________________ (8)

Annual Costs = (8) x (5)

= ______________________

= $ ______________________

Annual Heat Transfer Through A Building Enclosure Component — Average Temperature Method

Worksheet 10-8

(Page 1 of 1)

Company: ____________________ Date: ____________________

Location: ____________________ By: ____________________

*No. of months with monthly mean temperature below 18°C ______ (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= ____________________ h (2)

Average temperature difference across component ______________ °C (3)

Coefficient of transmission (U) ____________________ W/(m^2·°C)

Area of enclosure component (A) ____________________ m^2

Heat flow rate through component, Q = A x U x (3) x 3.6

= ____________________

= ____________________ kJ/h (4)

Annual heat flow through component, AH = $\frac{(2) \times (4)}{1000}$

= ____________________

= ____________________ MJ (5)

Fuel type ____________________

Fuel cost/unit ____________________ (6)

Fuel heat value/unit ____________________ (APPENDIX C) (7)

Heating system efficiency ____________________ (8)
(if unknown use 0.75 for oil or gas, 1.0 for electricity

Annual Costs = $\frac{(5) \times (6)}{(7) \times (8)}$

= ____________________

= $ ____________________

*Can be determined from Environment Canada weather data for specific locations.

Heating Of Outside Air
Average Temperature Method
Worksheet 10-9
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

*No. of months with monthly mean temperature below 18°C _____ (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= ______________________ h (2)

Average temperature rise ______________________ °C (3)

Hours of operation/week ______________________ h (4)

Hours of operation during heating season = (4) x (1) x 4.345

= ______________________ h (5)

Outdoor air flow (fa) ______________________ L/s

Average heating, Q = fa x (3) x 4.345

= ______________________

= ______________________ kJ/h (6)

Annual energy, AH = $\frac{(6) \text{ x } (5)}{1000}$

= ______________________

= ______________________ MJ (7)

Fuel type ______________________

Fuel cost/unit ______________________ (8)

Fuel heat value/unit ______________________ (APPENDIX C) (9)

Heating system efficiency ______________________ (10)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual cost = $\frac{(7) \text{ x } (8)}{(9) \text{ x } (10)}$

= ______________________

= $ ______________________

*Can be determined from Environment Canada weather data for specific locations.

Supply Air Reheat
Average Temperature Method

Worksheet 10-10
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

*No. of months with monthly mean temperature below 18°C ______ (Environment Canada) (1)

No. of hours in heating season = (1) x 730

= ______________________ h (2)

Original Supply Air Temperature (T1) ______________________ °C

Average increased supply air temperature (T2) ______________ °C

Hours of Operation/Week ______________________ h (3)

Hours of operation during heating season = (3) x (1) x 4.33

= ______________ h (4)

Supply air flow (fa) ______________________ L/s

Average reduction in reheat, Q = fa x (T2 - T1) x 4.345

= ______________________

= ______________________ kJ/h (5)

Annual reheat savings = $\frac{(4) \times (5)}{1000}$

= ______________________

= ______________________ MJ (6)

Fuel type ______________________

Fuel cost/unit ______________________ (7)

Fuel heat value/unit ______________________ (APPENDIX C) (8)

Heating system efficiency ______________________ (9)
(if unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual fuel cost savings = $\frac{(6) \times (7)}{(8) \times (9)}$

= ______________________

= $ ______________________

Excess Heat Output By A Terminal Heating Unit

Worksheet 10-11

Page 1 of 2

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Measured heating water supply temperature ______________ °C (1)

Measured heating water leaving temperature with fan operating (T2) ______________ °C (2)

Average heating water temperature, Ta $= \frac{(T1 + T2)}{2}$

= ______________________

= ______________________ °C

Catalogue output rating at Ta water temperature (Qr) __________ kJ/h

Calculate heating water flow, Qr = fw x (T1 - T2) x 15 000

Therefore, fw $= \frac{Qr}{(T1 - T2) \times 15\ 000}$

= ______________________

= ______________________ L/s (3)

Measured heating water leaving temperature with fan off ________ °C (4)

Calculate output with fan off, Q = (3) x [(1) – (4)] x 15 000

= ______________________

= ______________________ kJ/h (5)

*No. of months with avg. temp. above 18°C______________ (Environment Canada) (6)

Excess Heat Output By A Terminal Heating Unit

Worksheet 10-11

(Page 2 of 2)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

No. of hours with excess heat output = (6) x 730

= ______________________

= ______________________ h (7)

Type of fuel used ______________________

Unit cost of fuel ______________________ (8)

Unit heat value of fuel ______________________ (APPENDIX C) (9)

Heating system efficiency ______________________ (10)
(If unknown use 0.75 for oil or gas, 1.0 for electricity)

Annual fuel consumption = $\frac{(5) \times (7)}{(9) \times (10) \times 1000}$

= ______________________

= ______________________ (11)

Annual fuel cost saving = (11) x (8)

= ______________________

= $ ______________________

*Can be determined from Environment Canada weather data for specific locations.

Building Heat Loss Reduction
By Lowering Space Temperature
Worksheet 10-12
(Page 1 of 1)

Company: ______________________ Date: ______________________

Location: ______________________ By: ______________________

Degree days below 18°C (DDh) ______________________ (Table 2) (1)

No. of months with avg. temp. below 18°C ______________ (Environment Canada) (2)

No. of days in heating season = (2) x 30.4

= ______________________

= ______________________ (3)

Type of fuel used ______________________

Unit cost of fuel ______________________ (4)

Recorded annual fuel consumption ______________________ (5)

Annual fuel consumption per degree day $= \frac{(5)}{(1)}$

= ______________

= ______________ (6)

Initial space temperature (T1) ______________________ °C

Reduced space temperature (T2) ______________________ °C

Hours per day at reduced temperature ______________________ h (7)

Reduction in heating degree days $= \frac{(3) \times (T1 - T2) \times (7)}{24}$

= ______________________

= ______________________ degree days (8)

Annual fuel cost saving = (6) x (8) x (4)

= ______________________

= $ ______________________