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# Impact of Air-Filter Condition on HVAC Equipment

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OPERATED BY
MARTIN MARIETTA ENERGY SYSTEMS, INC.
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## EXECUTIVE SUMMARY

Air filters are primarily used in heating, ventilation, and air-conditioning (HVAC) systems to remove from the airstream those particulates that can cause fouling of the heat-transfer surfaces. However, the reduction of the amount of particulates in occupied areas is becoming an increasingly important concern for all air-filtration systems. The inexpensive, common, throwaway filter (which is suitable for the majority of applications) is designed to remove large-size particles. The removal of smaller-size particles, however, requires a more-efficient or special-purpose filter.

This analysis examines the effect of air-filter fouling on HVAC systems based on the characteristics of several components, the condition of the heat-transfer surfaces, and the cooling or heating cycle (vapor-compression refrigeration for air conditioners and gas-fired for furnaces). The performance of the air conditioner is estimated at various filter conditions. Energy consumption and filter replacement costs were calculated for a single-family military housing unit and for a modern barracks, using different filter replacement schedules during a typical heating and cooling season.

For military housing, the optimum schedule was to replace the filter monthly; there was, however, only a slight difference in the cost savings from this schedule when compared with the other schedules examined. Although the energy cost savings are slight, filter replacement is an important part of any preventive maintenance program. Also, an operating condition that can lead to a premature compressor failure was identified: clogged filter combined with a high outdoor temperature, beginning at about 105°F, for units having a capillary tube as the expansion device. Regular filter replacement can contribute to a longer life for compressors and motors and can be very cost-effective if early failure is prevented. Further, the filters should be replaced before they become clogged and dirt begins to accumulate on the evaporator coils. Replacing the filter is quick and easy; cleaning the evaporator coil is time consuming and tedious.

For the type of modern barracks chosen, a quite different situation exists. One hundred and fifty small room filters and eight larger filters in the air handler were considered. To determine an optimal filter-changing schedule, the maximum possible energy-cost savings were estimated by using the difference between all the filters being clogged and being clean. Because of the large number of filters involved and the associated labor cost to replace these filters, the estimated maximum energy-cost savings is easily offset by a few complete filter changes. If the room-filters' lifetimes can be extended by more effective use of the air-handler filters that already act as prefilters, the filter replacement cost can be minimized. Experience will be necessary to determine when the room filters will need changing. However, filters should be changed before becoming clogged.

For the modern military barracks and large buildings that use an air handler for circulating air for heating or cooling, damper adjustment and proper operation are important to energy conservation. As much as 30% of the annual energy cost can be saved, and the lifetimes of the filters in the air handler can be extended.

Inexpensive filters may suffice for most general purpose uses. This unconfirmed speculation arises from several small-sample-size observations of filter and evaporator coil conditions in air conditioners. For units located in areas with moderate humidities and air-infiltration rates, evaporator coils were found to be shiny clean. The dripping condensate may be the reason; a simple monitoring program could confirm this observation. If valid, there are savings to be realized, particularly in the purchase price of filters and in energy consumption. Inexpensive filters that remove the large-size particulates may be sufficient.

The condenser coils should be checked, especially in the fall and spring, for accumulation of leaves, seed pods from trees, or pine needles. Single-layered condenser coils are not prone to get dirty if properly installed. Multilayered condenser coils, however, are more likely to clog because of debris becoming trapped between the coils. Units in a high-dust area with heavy particles, such as those near a coal-fired plant or in an arid area, should be checked periodically.

Mowers should be cautioned not to blow grass clippings into the coils. If the condenser coils become dirty or clogged with debris, the discharge temperature and pressure will rise. Unless protected by a high-pressure limit switch, the compressor can be damaged.

Motors that are not permanently lubricated should be lubricated annually. The two most common causes of motor failures are motors operating at high temperatures and with a lack of lubrication.

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ABSTRACT

An analysis was performed of changes in energy consumption caused by dirty filters for heating, ventilation, and air-conditioning systems during both heating and cooling cycles. Basic heat-transfer fundamentals and the vapor-compression refrigeration cycle for air conditioners were used. The energy cost savings between a dirty and a clean filter were small and were mainly offset by the filter cost. However, the consequences of deferring filter replacement could be very costly because of early equipment failures.

## 1. INTRODUCTION

### 1.1 Purpose

The purpose of this study is to determine (1) the impact that varying degrees of air-filter fouling have upon energy consumption in occupied buildings for various types of heating, ventilation, and air-conditioning (HVAC) equipment and (2) the energy savings that may be expected based upon filter conditions and a replacement schedule.

### 1.2 Scope

The impact of varying degrees of fouled filters on the energy consumption of HVAC equipment (air conditioners, air-handling units, heat pumps, fan coil units, and furnaces) is discussed. Different types of air filters are described. Recommendations are made concerning the installation, maintenance, and replacement of air filters on U.S. Army facilities. Although detailed coverage of maintenance for HVAC systems is beyond the scope of this report, some simple maintenance actions that can improve efficiency have been identified. The economic and energy-saving aspects of regular air-filter replacement are discussed.

To appreciate fully the importance of air filters in HVAC systems, a brief discussion about various air filters and some basic fundamentals involved in cooling and/or heating systems will be presented. This information will aid in understanding the interactions among different components of the HVAC system.

### 1.3 Background and Methodology

The U.S. Army Facilities Engineering Support Agency (USAFESA), Fort Belvoir, Virginia, issued Task Order No. 0003 on April 22, 1985, requesting that the impact of air-filter condition on HVAC equipment energy consumption be evaluated.[1] Visits to two Army installations were specified as part of this task. Fort Bragg in Fayetteville, North Carolina, was designated as one of the sites because it has initiated a filter evaluation program. The other site selected was, in fact, two sites: Fort McNair and Fort Myer in the Military District of Washington were combined because of their close proximity and relatively small sizes. The last two forts did not have a planned filter-change program other than routine, scheduled maintenance of the HVAC systems.

This task specified that a literature search be performed to determine whether any previously completed work or study would satisfy USAFESA's requirement for information on air filters and energy conservation. Thus, a computerized literature search was conducted in the DIALOG, Compendex, DOE Energy, and DTIC data bases. Because only a limited amount of material was uncovered, a manual literature search was also conducted. In addition, many independent testing organizations and product manufacturers were contacted for related information.

Only a single document[2] that had as one of its goals the measurement and calculation of the effects of clogged filters in a warm-air furnace and in a central air conditioner was found. Because the calculated values were not supported by the measured values, a careful evaluation was made. The desired effects were not simulated in the experimental procedure. The calculation seems reasonable, but in some cases, the assumption did not describe realistic situations. However, some of the computation relating to furnaces is believed valid and will be incorporated.

Lacking experimental data, this analysis will rely upon basic effects and behavioral fundamentals of components. The discussion will be limited to only what is thought necessary to understand the effects that filters and improper maintenances have on HVAC systems. Because the basic fundamentals of HVAC systems are well understood, supplemental testing is not recommended.

# 2. AIR-FILTER PURPOSE, TESTING, TYPES, AND SELECTION

## 2.1 Air-Filter Purpose

Air filters are used to remove from the upstream air particulate matters that can cause fouling of the heat-transfer surfaces, fans, motors, and duct interiors in the HVAC system. A side benefit that has become increasingly more important is the reduction of particulates — such as fly ash, pollen, carbon, oil, soot, lint, hair, bacteria, and metallic oxides — to occupied areas. The largest-size particles, in the range of 1 to 150 μm in diameter, are dusts thrown into the air by mechanical agencies, such as in grinding, crushing, drilling, and blasting. These particles can be abrasive and can damage motors in the HVAC system. Fortunately, because of their sizes and weights, these particles do not travel great distances from their sources. The medium-size particles, in the range of 0.2 to 1 μm in diameter, are fumes as a result of, for example, reactions from distillation, oxidation of metal fumes, and chemicals; the most prevalent example would be sulphuric acid mist. Particles <0.3 μm in diameter are the smallest and usually come from incomplete combustions of coal, oil, tar, and tobacco; the smoke from these reactions can travel great distances. Of the total, >99% of the particles are 1 μm or less; a few particles in excess of 30 μm are found. Also, there may be up to 10 million particles/$ft^3$ of air, and air sampling studies indicate that urban areas can have up to ten times more suspended particulate matter than rural locations. The particulate concentrations vary within the same location depending on the height above street level, day of the week, activity, and season. Because of the many particles of varying types, sizes, and concentration levels, no single air-filter type is suitable for every condition.

## 2.2 Air-Filter Testing

Some considerations for comparing air-filter performance are (1) how thoroughly a filter removes particulates from the airstream, (2) how much dust a filter can hold, (3) how much resistance to airflow the filter presents, and (4) how frequently the filter must be replaced or cleaned. To answer these questions, testing procedures have been developed. The most commonly accepted testing technique uses the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) standard 52-76, *Method of Testing Air-Cleaning Devices Used in General Ventilation for Removing Particulate Matter.*[3]

"Arrestance" and "efficiency" are expressions used to define an air cleaner's performance. Arrestance is a measure of the ability of a filter to remove large-size particles but is not an indication of the filter's ability to remove small-size particles. Because of this shortcoming, the dust-spot efficiency test was developed and is a measure of the ability of the filter to remove smaller particles that can discolor the interior surfaces of the occupied space. Both measurements are normally performed whenever a filter is tested. However, if the manufacturer states that the dust-spot efficiency for a given air filter is <20%, the filter need not be subjected to this test. Usually, ordinary filters have a high arrestance and a low efficiency. Special filters have a high arrestance with the efficiency varying from 20 to >99%. The dust-holding capacity that is measured during the arrestance test for a disposable or non-self-renewable filter is the weight of dust held by the filter when the maximum operating resistance or pressure drop across the filter, as set forth by the manufacturer, is reached. The dust-holding capacity does not indicate the life of a filter but does give a comparison guide to determine which type of filter will last longer under the same operational conditions.

## 2.3 Filter Types

The throwaway filter [Fig. 1(*a*)] is the most common. If the filter medium is contained in a fiberboard frame, the complete unit is thrown away when it becomes dirty. If a permanent holding frame is used, the filter medium — often fiberglass or some other high-porosity, coarse-woven fiber — is replaced. To improve on dust retention, the medium is coated with a viscous adhesive substance and is effective on the larger (>5 μ) dust particles. A technique to increase the dust-holding capacity

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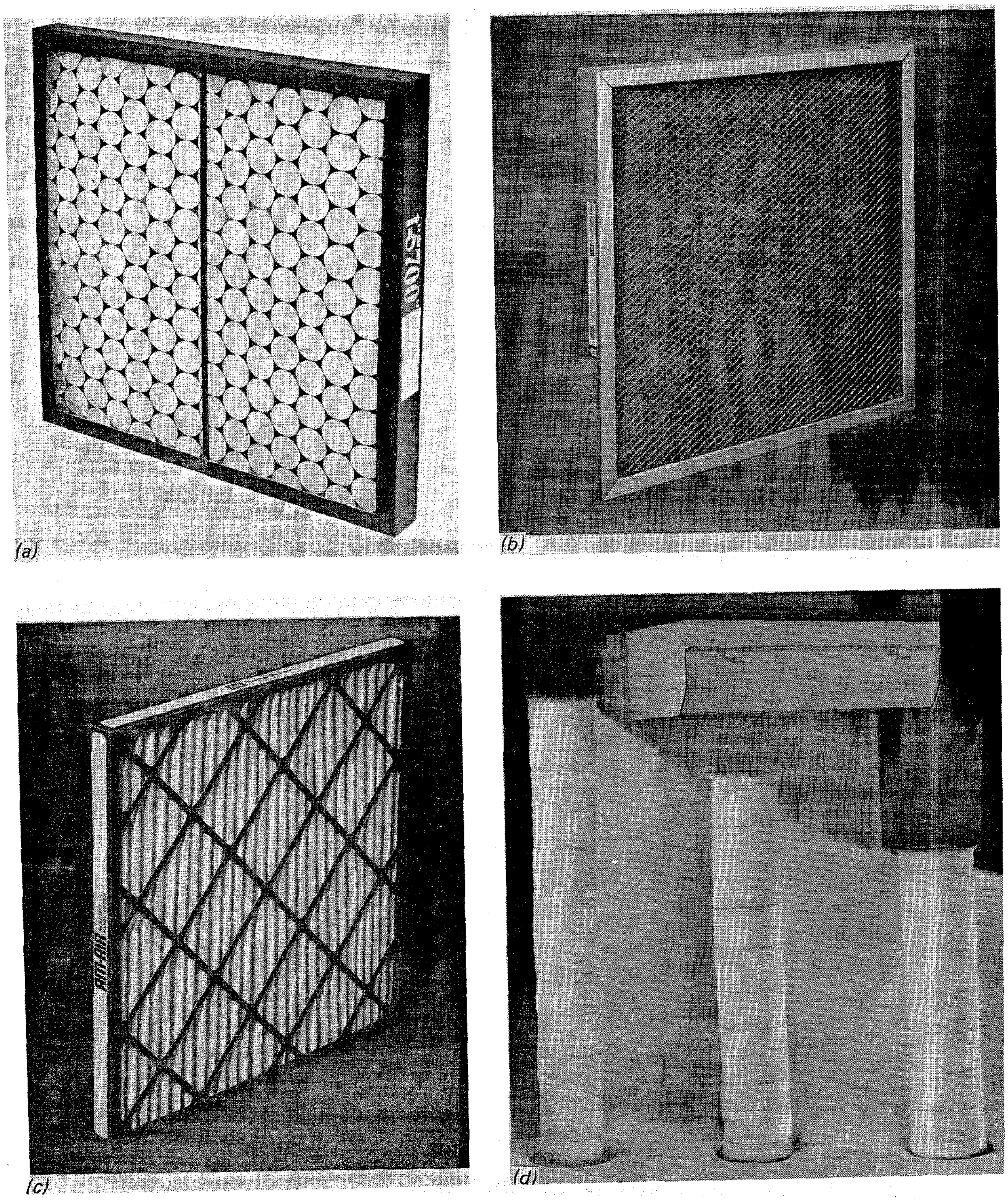


Fig. 1. Various filter types (courtesy of American Air Filter Company).

and arrestance is to construct the medium of progressive density. The open-weave fiber pattern is on the upstream side of the airflow and is normally colored to aid in correct installation. The fiber diameters become smaller and the weave progressively tighter toward the air-leaving side. If the wrong side is turned to the airflow, the efficiency is not reduced; the usable lifetime, however, is shortened.

A permanent filter [Fig. 1(*b*)] is constructed of metal and can be cleaned and reused. The filter medium, a viscous, adhesive-coated, metal mesh, is overlapped to eliminate any direct airflow through the mesh. After being washed with a warm detergent solution, the mesh must be recoated. Because a washing facility with warm water must be available and if there is any labor cost associated with cleaning these filters, replacement with a throwaway type is more cost-effective.

A semipermanent filter is the thin, polyfoam medium often found in window air conditioners or residential heating/cooling units. The medium can be permanently attached to a frame, inserted into a permanent frame, or cut from a larger piece to fit a filter opening. If the latter is the case, some support is provided to hold the medium in position. The polyfoam is washable, and a special filter spray can be used to improve the dust collection capability. Age and use will eventually cause the medium to disintegrate. The medium replacement can be the same type or the thin, viscous-coated medium.

The dry, or "straining," filter has a higher dust-spot efficiency and can be considered as another throwaway type. Cotton and synthetic fibers are the medium for filters with an average dust-spot efficiency of 15 to 40%. To obtain efficiencies up to 90 to 95%, ultrafine fiberglass is used. These higher efficiencies result in higher initial and final pressure drops across the medium. For longer life and a greater surface area, the medium is either contour pleated [Fig. 1(*c*)] or shaped into inflatable bags, which are collapsed for shipment and storage space reduction. The filter medium can be fitted into a supporting frame or basket or can come supplied with its own frame. Because of this medium's relatively higher cost, prefiltering with a low-efficiency, cheaper filter is recommended.

In the special filter category are the high-efficiency, particulate air filters (HEPA) and the ultra-low-penetration air filters (ULPA), both of interest to those concerned with clean rooms and nuclear power plant operations. Slightly less efficient than these two types are those designed to remove airborne biological contaminants in hospital critical areas and food and pharmaceutical processing plants. These special filters are beyond the scope of this report and will not be discussed. Also not covered are electronic air cleaners.

All of the filters discussed above can be classified as unit filters and come packaged individually or have a discrete replacement piece of medium. For large air volumes, rows of these filters can be installed in the air-handling portion of the HVAC system. For still larger air volumes, the filters are installed on a wall of a room devoted to handling the filters; there may be more than one of these rooms. All of the filters in the room are usually changed at the same time.

Another method for filtering large quantities of air is the use of roll filters [Fig. 1(*d*)]. The medium is a large, bulk roll of filter material that forms a curtain in a frame. Similar to the unit filters, as many frames as necessary in the filter room can be used to handle the airflow. The choice of medium is determined by the amount of filtration required. When the medium becomes dirty, fresh medium is unrolled. The used medium is rolled at the bottom if the frame is vertically mounted (horizontal mounting is sometimes employed). The medium can be intermittently fed manually or automatically by some predetermined criterion. By small incremental feeding of the medium, the dust-holding capacity is almost doubled over that obtained by feeding the medium a full sheet at a time. Some criteria used to roll the medium are visual inspection, a preset time interval, a preset pressure drop across the filter, and a measured light attenuation level through the dirty medium. The least-reliable method is the visual inspection because a clogged filter may not appear dirty and a seemingly dirty filter may not be clogged. The timer method is not much better because of the many variations in the pollutant levels and the fact that the timer does not allow full use of the first sheet of medium. The pressure drop across the medium should be the best criterion, but in actual practice there have been some problems with the pressure switches causing troublesome operation. The use of a photoelectric sensor has been more reliable and is the preferred method.

## 2.4 Air-Filter Selection

Because of the numerous types of filters for many different purposes, the selection of the most-economical filter must be based on the consideration of certain factors. One obvious factor is the availability of filters that match the frame size and thickness of the filter housing. More important, the filter must be compatible with the rated specifications of the HVAC system, namely, (1) the filter-face velocity, (2) the pressure drop across the filter, and (3) the airflow capacity. For the more-common viscous impingement filters, some typical values are presented. Depending on the thickness of the filter, which typically ranges from 1/2 to 4 in., the values are 300 to 500 ft/min for the initial filter-face velocity and 0.10 to 0.25 in., water gauge, for the initial resistance. The final pressure drop is expected to be ~0.5 in., water gauge. The final airflow capacity can be determined from the size of the opening and the other two quantities. The average arrestance is ~80%. The average dust-spot efficiency will range from 5 to 10% for the 1/2-in.-thick medium and 10 to 25% for the 4-in.-thick medium.

A determination must be made in the selection of the degree of filtration required. If protecting the heat-transfer surface is the main goal, a low-efficiency filter will be satisfactory under normal operating conditions. If cleaner air is required, a wide variety of higher-efficiency filters is available. These filters offer more resistance to the airflow and affect the energy consumption by requiring more horsepower to move the air through the filter.

Some major filter manufacturers have representatives available to advise on the proper selection of air filters and to conduct seminars; some will provide life-cycle studies. Reference 4 contains a comprehensive list of air-filter manufacturers.

# 3. FUNDAMENTALS OF HVAC SYSTEMS

## 3.1 Refrigeration Cycle

A brief description of the simple vapor-compression refrigeration cycle for an air-to-air system is presented[5] to aid in understanding some of the problems that clogged air filters or dirty heat-transfer surfaces can cause in air-conditioning systems. The basic components (Fig. 2) are the evaporator, compressor, condenser, and expansion device.

The evaporator is the coil through which indoor air is passed to provide cooling. The refrigerant enters the evaporator from an expansion device as a liquid-vapor mixture and leaves as a low-pressure, low-temperature, slightly superheated vapor (point 1). This gas then enters the compressor where it is compressed to a high-temperature, high-pressure, superheated vapor (point 2). This hot vapor enters the condenser coil where it is cooled and condensed — at constant pressure — to

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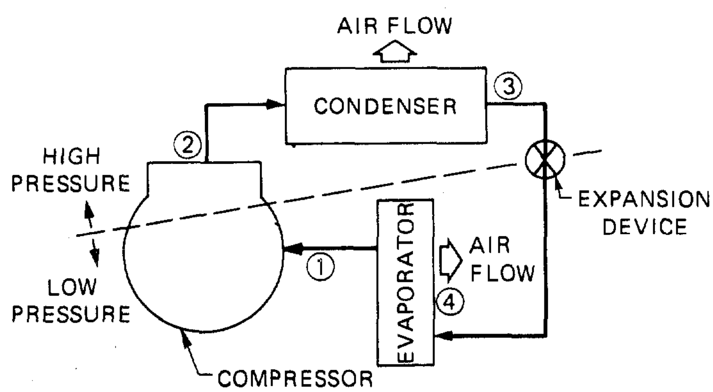


1 — LOW-PRESSURE, LOW-TEMPERATURE, SLIGHTLY SUPERHEATED VAPOR

2 — HIGH-PRESSURE, HIGH-TEMPERATURE, SUPERHEATED VAPOR

3 — HIGH-PRESSURE, INTERMEDIATE-TEMPERATURE LIQUID

4 — LOW-PRESSURE, LOW-TEMPERATURE, LIQUID-VAPOR MIXTURE

Fig. 2. Simple refrigeration cycle: schematic flow diagram.

a high-pressure, intermediate-temperature liquid (point 3). The heat of condensation is dissipated into the atmosphere by air blown through a coil usually located outdoors. The liquid then travels to the expansion device through which it is then admitted to the adjacent evaporator. When the refrigerant enters the low-pressure region of the evaporator, enough liquid flashes into vapor to cool the remaining liquid to the prevailing saturation temperature (point 4). In the evaporator the remaining liquid is vaporized (by absorbing heat from the air being cooled) to the original slightly superheated vapor starting point (point 1) to repeat the cycle. Because heat travels from a higher to a lower temperature, the conditioned space heat is, therefore, transferred to the refrigerant in the evaporator and then rejected to the outdoor air by the condenser. When the unit is operating, it is this heat, plus the heat of compression of the refrigerant, that is rejected by the outdoor unit.

## 3.2 Other Cooling Systems

Although an air-to-air refrigeration system has been described, the same principles apply to other cooling systems using the compression cooling cycle. For medium-size systems, where the cost of water is not prohibitive or where a source of outdoor air is not convenient, the condenser can be cooled by city water that is usually dumped down a drain. An air evaporator coil may still be used to cool the indoor air.

In large buildings, a hydronic system is usually employed for heating and cooling. The heat removal from the condenser coil may employ large fan units or use water. For water-cooled units, cooling towers are normally used to cool an inner loop of water that is heated by the condenser. For cooling, chilled water surrounding the evaporator is piped throughout the building. Heat is removed by blowing the indoor air through coils cooled by the chilled water, which is then cycled back to the evaporator. For heating, the hydronic system's water is heated.

## 3.3 Heat Pump

The heat pump uses the refrigeration cycle for both heating and cooling. When there is a demand for heat, the indoor coil becomes the condenser through a reversing valving arrangement. The liquid refrigerant is expanded as a vapor through the expansion valve and evaporates in the outdoor coil. Heat is supplied from the atmosphere for the vaporizing action and is then transferred to the interior of the building by compressing the vapor and condensing the refrigerant to a liquid. During periods of extreme cold weather when the heat pump does not have sufficient capacity to maintain the desired indoor temperature, auxiliary electric resistant heater strips are employed.

## 3.4 Furnace

The visit to the Army installations indicated a predominance of gas furnaces. Because of the limited scope of this report, only the forced-air, atmospheric gas-fired type furnace will be discussed. Figure 3 is a cutaway drawing of this type of furnace with the air filter removed from the cold-air return. Gas is mixed with air and burned in a chamber at atmospheric pressure. The outer surface of this chamber is the heat exchanger that heats the interior air, which is then distributed throughout the building through ducts. The main gas valve is controlled by the building thermostat. The blower motor is activated by another thermostat that senses when the heat-exchanger temperature is hot enough. Heat can continue to be extracted from the hot heat exchanger long after the flame has been extinguished. To conserve energy, the blower motor is deactivated when a cool-enough temperature is reached.

The amount of gas to the amount of primary-air mixture in the combustion process is factory set for most small units and is usually not adjustable. For larger units this ratio can be adjusted for optimum combustion efficiency. The steady state efficiency is measured by analyzing the flue-gas temperature and the concentrations of certain combustion products. The flue-gas temperature should be as low as possible, ~350°F, without condensation forming in the flue venting system and causing corrosion. Higher flue temperatures waste energy up the stack. The optimum fuel-air mixture is achieved with stoichiometric (complete) combustion.

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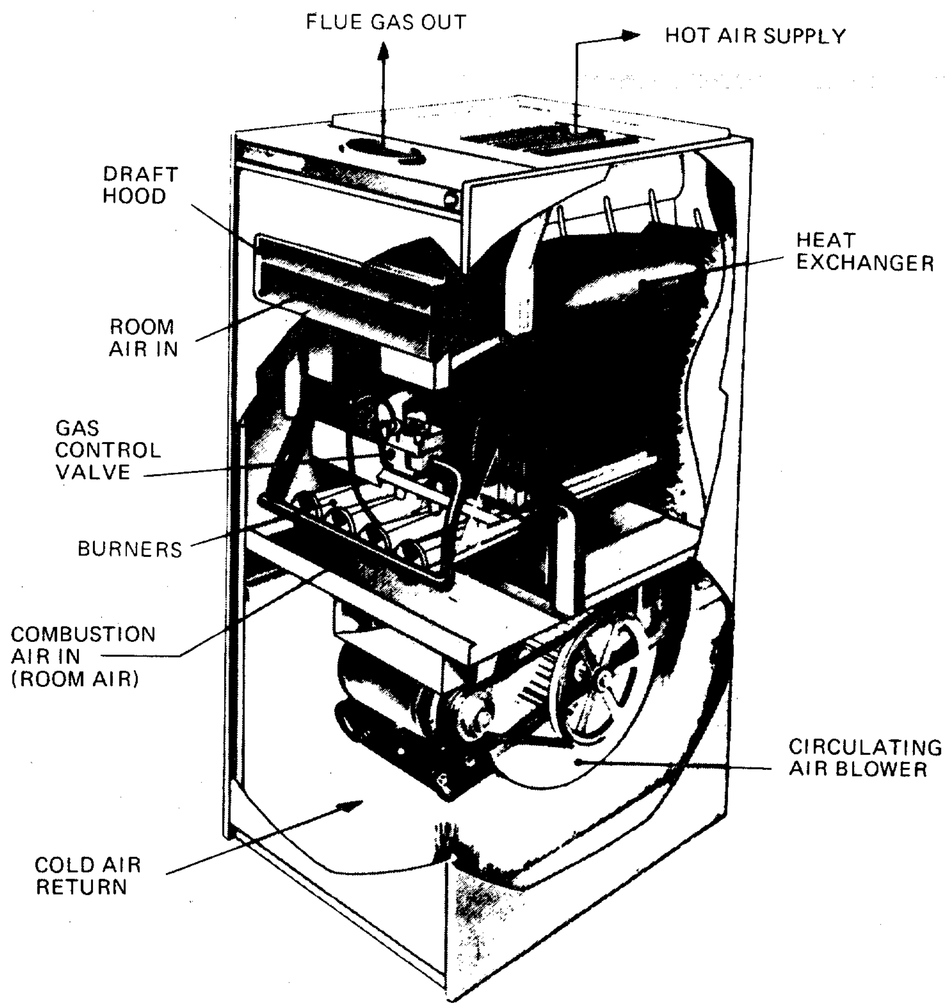


Fig. 3. Atmospheric combustion furnace — central, upflow, forced circulation.

## 3.5 Air Handler

### 3.5.1 Blower assembly and motors

The blower assembly is composed of the fan, usually centrifugal, and housing. Impellers are forward curved, radial or straight, and backward curved [Figs. 4(*a*)–(*c*)]. Typical performance curves are shown in Figs. 5(*a*)–(*c*). The highest efficiencies and static pressures usually occur around one-half of the wide-open volume. For the forward-curved and the radial impeller fans, the power rises continuously toward the free-air delivery, and it is possible to overload the motor if a

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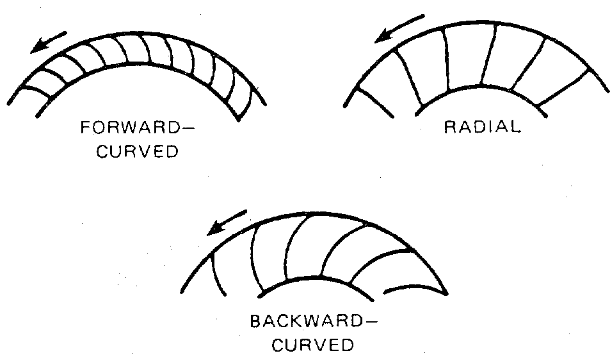


Fig. 4. Fan impellers (rotation is in the direction of the arrows). *Source:* Adapted with permission from *Carrier System Design Manual. Part 6: Air Handling Equipment*, Carrier Air Conditioning Company, Syracuse, N.Y., 1963, Fig. 5, p. 6-3.

reasonable capacity margin has not been provided. For the backward-curved impeller fan, the horsepower curve peaks at high capacity and does not overload at any point on the curve at any fan speed if the speed is maintained.

For a given type of fan, the characteristic performances[6] are that

1. the capacity varies directly with the fan speed,
2. the static discharge pressure varies as the square of the fan speed, and
3. the horsepower or energy requirement varies as the cube of the fan speed.

The air filter is always installed upstream of the blower assembly and the heat exchanger to keep these components clean and the air filter from becoming heated.

### 3.5.2 Dampers

On medium-to-large HVAC systems automatic dampers are installed to allow the entry of fresh outdoor air. These dampers operate by closing to the minimum position in the summer and winter and by opening fully during intermediate seasons. During the intermediate season the use of outdoor air may be sufficient to maintain the indoor-air comfort level. If the dampers are not closed to minimum position during hot or cold

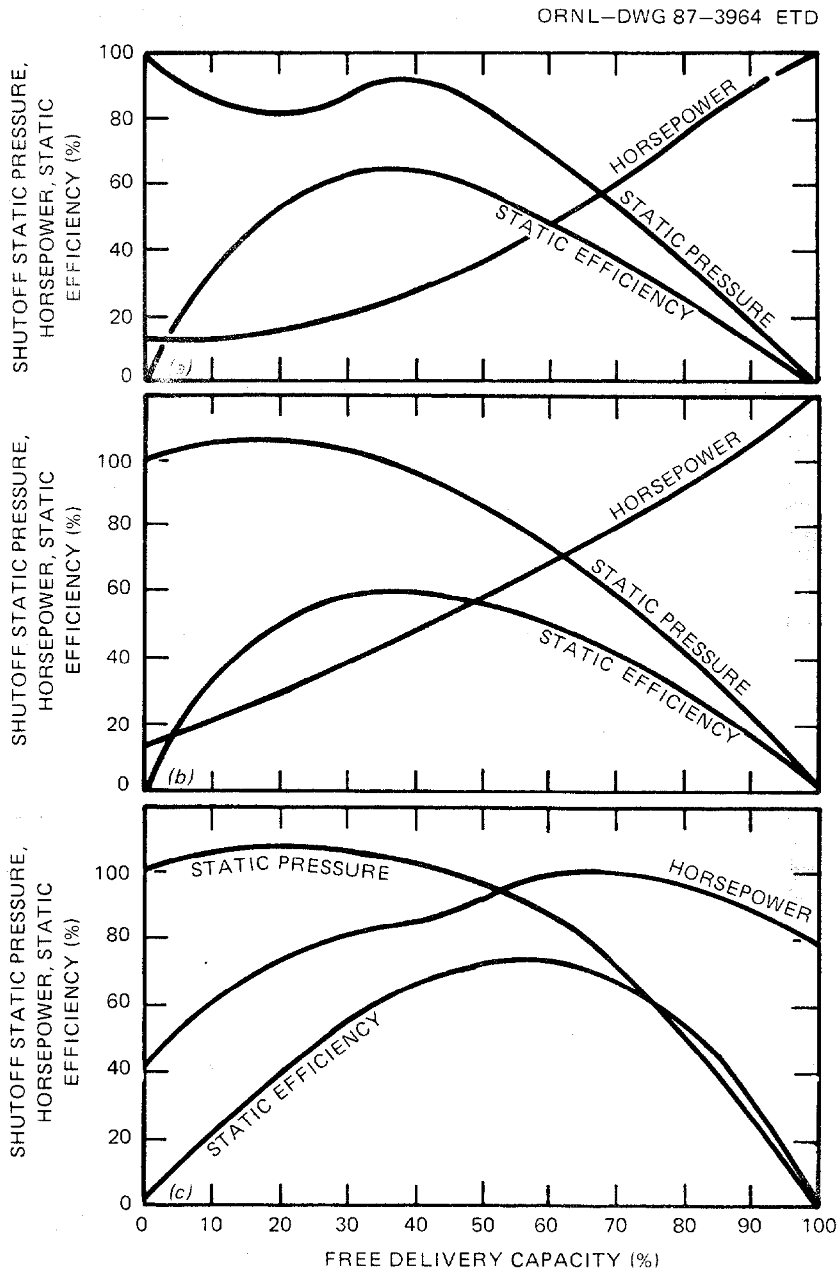


Fig. 5. Fan impellers' performances. (*a*) Forward-curved fan impeller performance, (*b*) radial fan-impeller performance, (*c*) backward-curved fan-impeller performance. *Source*: Adapted with permission from *Carrier System Design Manual. Part 6: Air Handling Equipment*, Carrier Air Conditioning Company, Syracuse N.Y., 1963, Figs. 6–8, pp. 6-3 and 6-4.

outdoor temperatures, energy is wasted by bringing the air to the desired comfort level. On large units the air is usually filtered near the entry.

# 4. EFFECTS ON HVAC SYSTEMS' PERFORMANCES

## 4.1 Refrigeration Systems

In cooling systems the only useful refrigeration is due to the heat transferred by the evaporator. The maximum performance of the cycle is obtained by operating at the maximum possible evaporating temperature and at the lowest possible condensing temperature.[7] Any deviation in these temperatures will affect the air-conditioner's performance and energy consumption. As the air filter becomes clogged, the airflow across the evaporator is reduced, and heat transfer from the air to the evaporator is inhibited. A dirty evaporator surface also lowers the heat-transfer coefficient so that less heat can be transferred to the refrigerant. When either the filter becomes clogged or the evaporator surface becomes dirty, the evaporator temperature and the suction pressure are lowered, thus reducing the cycle efficiency.

Single-layered condenser coils, although not filtered, are not prone to get dirty if properly installed. Multilayer condenser coils are more likely to clog because of debris becoming trapped between the coils. When installed at the ground level, the unit should be placed on a concrete pad of sufficient size so that rain cannot splash dirt onto the coils; ground cover or certain bushes also can be effective in preventing splashing. If possible, no trees should be in close proximity. If nearby trees are unavoidable, the coils should be checked periodically for accumulation of leaves, seed pods from trees, or pine needles, especially in the fall and spring. Units in a high-dust area with heavy particles, such as those from a coal-fired plant, also should be checked periodically. Mowers should be cautioned not to blow grass clippings into the coils. If the condenser coils become dirty or clogged with debris, the discharge temperature and pressure will rise. Unless protected by a high-pressure limit switch, the compressor can be damaged.

In refrigeration systems there are two basic types of expansion devices. First, the thermostat expansion valve has a thermostatic bulb attached to the suction line from the evaporator coil. This bulb senses any increase in the coil temperature and causes the valve to open wider to admit more refrigerant. The pressure of the refrigerant on the low-pressure side of the valve tends to close the valve. When the valve is functioning properly, no liquid can get to the compressor. The second expansion device is a small-diameter capillary tube sized to reduce the condenser pressure to the evaporator pressure. The amount of refrigerant passing through the tube is dependent upon the pressure difference across it and the amount of subcooling of the high-pressure liquid entering the tube. Although the tube tends to be self-adjusting, the combination of less heat being transferred by the evaporator and a higher liquid pressure caused by an abnormally high outdoor temperature or a dirty condenser coil can overcome the compensating effect and produce a flood-back condition in which some liquid reaches the compressor. Because liquid is virtually incompressible, damage to the compressor is likely if enough liquid enters.

As related by one air-conditioner manufacturer, flood-back has been observed during tests conducted on their units with a capillary tube and an evaporator airflow reduction of 55% of the unit's rating at an outside air temperature of 105°F for its single-packaged units and at 112°F for its split systems. Thus, the combination of the air conditioner's being taxed during extremely hot weather and having reduced airflow through the evaporator may produce a flood-back condition, causing damage to the compressor. This set of operating conditions may be considered one limiting criterion beyond which the air conditioner should not be operated. This limit will also be used in the analysis. Therefore, for units with a capillary tube expansion device, the importance of keeping the air filter and the evaporator coil clean during peak cooling periods cannot be overemphasized.

Since the mid-1950s, the capillary tubes, easier and cheaper to manufacture, have become widely used in small- and middle-size units. These tubes also tend to give no problem because there is no moving part. If the refrigerant becomes dirty, the tube can plug, but replacement is relatively inexpensive. However, under the flood-back condition the more-expensive compressor can be damaged.

### 4.1.1 Self-contained refrigeration-unit correlation

The primary effects of clogged filters on self-contained refrigeration units are a reduction in the quantity of air that flows through the evaporator coil and the change in the load placed on the blower motor. The reduction in airflow depends upon the amount of dirt collected on the filter and on the evaporator coil.

A correlation for the airflow reduction vs the capacity or the ability of the air-conditioner system to remove heat has been developed and used for many years by a major manufacturer.[8] The correlation is

$$\frac{Q_{DIRTY}}{Q_{CLEAN}} = \left(\frac{Cfm_{DIRTY}}{Cfm_{CLEAN}}\right)^{1/6} ,$$

where

$Q_{DIRTY}$ = the heat-removal capability of the air-conditioner system after the filter becomes clogged,

$Q_{CLEAN}$ = the heat-removal capability of the air-conditioner system with a clean filter,

$Cfm_{DIRTY}$ = the amount of airflow (cubic feet per minute) through the evaporator coil after the filter becomes clogged,

$Cfm_{CLEAN}$ = the amount of airflow (cubic feet per minute) through the evaporator coil with a clean filter.

The one-sixth power correlation is a good indicator that the cooling capacity is not greatly affected by a dirty filter.

Their method of simulating the airflow reduction is by incremental addition of extra filters. Normally, five filters will reach the upper limit of the pressure drop while maintaining the same velocity distribution profile through the evaporator as caused by a clogged filter. This correlation does not apply to the change in fan power usage or to heat pumps in the heating mode. This manufacturer conservatively estimates the correlation to be in the 15 to 20% accuracy range; however, when applied to a collection of published values even at an extreme range for many widely different manufacturers' units, the disagreement is not >2%.

By setting a limit for the airflow through a dirty filter to 55% of the flow through a clean filter, a simpler, approximate expression can be derived. Although this simpler expression may seem to be applicable only to units with a capillary tube, it will be seen later to be suitable for use with most types of air-conditioning units. The expression is

$$\frac{Q_{DIRTY}}{Q_{CLEAN}} = 0.2 \times \frac{Cfm_{DIRTY}}{Cfm_{CLEAN}} + 0.8 \ , \quad \text{for } \frac{Cfm_{DIRTY}}{Cfm_{CLEAN}} > 0.55 \ .$$

When the mass flow limit is substituted, the lower limit of the heat-removal capacity factor $Q_{DIRTY}/Q_{CLEAN}$ is 0.91.

How valid is 0.91 as a lower limit? During one of the times that filters were being changed at Fort Bragg, some of the used filters were saved for testing. The degree of clogging varied from light to extremely heavy, and there was no doubt that the dirtiest filter had exceeded its useful life. Also, the manufacturer's suggested maximum allowable pressure drop across that filter was exceeded. In a fabricated test rig, the airflow measured by Fort Bragg's Engineers was 1150 $ft^3$/min for a clean filter and 700 $ft^3$/min for the dirtiest filter. These measurements translate into a reduced mass flow factor $Cfm_{DIRTY}/Cfm_{CLEAN}$ of 0.61, which is considerably higher than the 0.55 chosen. Applying either correlation, the capacity factor $Q_{DIRTY}/Q_{CLEAN}$ is 0.92. Therefore, regardless of what type of expansion device is used, the value of 0.91 appears to be a valid lower limit for the reduction in the heat-removal capacity because of a very dirty filter. The limit also corresponds with that condition where flood-back can occur.

### 4.1.2 Air-conditioner performance evaluation

In addition to a reduction in the airflow through the evaporator coil caused by a clogged filter, dirt on the evaporator coil will reduce the heat-transfer capability of the refrigerant to extract heat from the airstream. Even a thin film of dirt on any metal surface will affect the heat-transfer coefficient. For a dirty evaporator a reasonable estimated drop in the refrigerant's temperature is ~10°F; a much larger change is not expected. If the refrigerant becomes slightly colder, ice will likely form on the evaporator coil.

Likewise, a change in the heat-transfer coefficient because of a dirty condenser is expected to increase the condensing temperature ~10°F. If there is airflow blockage, the temperature could rise higher. These estimated temperature limits do not represent the worst possible case but are reasonable expected limits because of reduced airflow or heat transfer. Long before the maximum limits are reached and especially during hot weather, the occupants should be complaining about inadequate cooling, or the unit may have malfunctioned.

The compressor's performance data can be used to determine the magnitude of the effects of dirty filter, dirty or clogged evaporator coil, and dirty or clogged condenser coil.[9,10] The capacity and power requirement for compressors operating at various suction and condensing temperatures are available from the manufacturers. The suction temperature, nearly equal to the temperature of the refrigerant leaving the evaporator, will be assumed to be the same. Using manufacturer's data, the compressor's capacity for different units over all sizes was found to be nearly proportional to any other unit at the same operating condition for the commonly used R-12 and R-22 refrigerants. However, the power requirement may differ slightly, especially when older units are compared with modern units with more-efficient motors or when there is a comparison between motors sealed within the compressor or located externally.

From the performance data three performance factors — capacity, power, and energy — will be defined. These factors are derived from ratios of compressor capacity and power requirements at some suction temperature $T_S$ and condensing temperature $T_C$ to the corresponding quantity at a selected baseline temperature condition $T_{SBL}$ and $T_{CBL}$.

The capacity factor is

$$\text{Capacity Factor} = \frac{Q(T_S, T_C)}{Q(T_{SBL}, T_{CBL})} ,$$

where the quantity Q is the cooling capacity at the designated temperatures.

The power factor is

$$\text{Power Factor} = \frac{P(T_S, T_C)}{P(T_{SBL}, T_{CBL})} ,$$

where the quantity P is the compressor power requirement at the designated temperatures.

The energy factor is

$$\text{Energy Factor} = \frac{\text{Power Factor}}{\text{Capacity Factor}} .$$

These performance factors are valid only when sufficient capacity exists to maintain the desired indoor temperature. When the capacity is not sufficient, the compressor will have to run continuously. Another limitation is that the unit must be properly charged. A low charge can cause icing of the evaporator coil that affects both the airflow and the heat-transfer coefficient.

The capacity factor is the same term as the previously defined heat-removal capacity factor $Q_{DIRTY}/Q_{CLEAN}$. For each of the performance factors, values less than unity indicate a reduced effect; values greater than unity have an increased effect. To remove the same amount of heat when the heat-removal capacity factor is less than one, the air conditioner must operate longer than when operating at the baseline condition. If the power factor is greater than one, more energy will be consumed over the same operating time of the baseline condition. The fractional change in the total energy consumed in providing the same heat removal is indicated by the energy factor.

A set of compressor performance data will be used for demonstration. As stated previously, differences that reflect the more-efficient motors are expected in the power and energy factors for modern compressors. Because the compressors in the U.S. Army are typically "pre-energy crisis" in age, an older set of data (Table 1) will be used. The data[9] are for a compressor rated at 15 tons using R-22.

To develop the performance factors in Table 2 for an air-to-air refrigeration unit, baseline operation conditions of 45°F for the suction

Table 1. Capacity and power demand for various suction and condensing temperatures for a 15-ton compressor operating on R-22[a]

| Suction temperature (°F) | Capacity and power at various condensing temperatures | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 85°F | | 95°F | | 105°F | | 115°F | |
| | Tons | bhp | Tons | bhp | Tons | bhp | Tons | bhp |
| 0 | 6.8 | 10.3 | 6.1 | 9 | 5.6 | 10.9 | | |
| 5 | 7.5 | 10.8 | 7.1 | 11.3 | 6.3 | 11.7 | 5.7 | 12.1 |
| 10 | 8.9 | 11.2 | 8.1 | 11.9 | 7.3 | 12.4 | 6.6 | 13 |
| 15 | 10.1 | 11.7 | 9.3 | 12.4 | 8.4 | 13.2 | 7.6 | 13.8 |
| 20 | 11.4 | 12.1 | 10.5 | 13 | 9.6 | 14 | 8.7 | 14.5 |
| 25 | 12.9 | 12.4 | 11.9 | 13.4 | 11.0 | 14.5 | 9.9 | 15.3 |
| 30 | 14.4 | 12.5 | 13.3 | 13.8 | 12.4 | 14.9 | 11.3 | 15.8 |
| 35 | 16.1 | 12.5 | 14.9 | 14 | 13.8 | 15.3 | 12.7 | 16.4 |
| 40 | 17.7 | 12.6 | 16.6 | 14.1 | 15.4 | 15.6 | 14.2 | 16.9 |
| 45 | 19.6 | 12.7 | 18.3 | 14.3 | 17 | 15.9 | 15.6 | 17.5 |
| 50 | 21.5 | 12.7 | 20 | 14.4 | 18.6 | 15.9 | 17.3 | 17.7 |

[a]Reproduced by permission of The Trane Company, La Crosse, Wisconsin.

*Source*: R. W. Roose, comp.-ed., *Handbook of Energy Conservation for Mechanical Systems in Buildings*, copyright 1978 by Litton Educational Publishing, reprinted by permission of Van Nostrand Reinhold, New York, all rights reserved, Table 43-3, p. 281.

temperature $T_{SBL}$ and 95°F for the condensing temperature $T_{CBL}$ were elected because of the following considerations.

1. To generate a cooling sensation, the air temperature leaving the evaporator should be at least 20°F cooler than the indoor temperature. If 75°F is the indoor thermostat setting, the air leaving the evaporator should be ~55°F, which would require that the suction temperature be ~45°F.
2. Cooling needs are expected when the ambient outdoor temperature reaches ~85°F and hotter. This temperature translates into a condensing temperature of ~95°F, which is our reference temperature. A water-cooled unit would probably have a condensing temperature of 105°F, and a similar table could be constructed. A check showed that these factors differ slightly, but not significantly, from the factors of the above baseline temperatures.

Table 2. Factors for compressors operating with R-22

| Case | Condition | Effect[a,b] | Capacity factor | Power factor | Energy factor |
|---|---|---|---|---|---|
| 1 | Very dirty filter | 5°F Colder evaporator | 0.91 | 0.99 | 1.09 |
| 2 | Dirty evaporator | 10°F Colder evaporator | 0.81 | 0.98 | 1.20 |
| 3 | Very dirty filter, dirty evaporator | 15°F Colder evaporator[c] | 0.73 | 0.97 | 1.33 |
| 4 | Dirty condenser | 10°F Hotter condenser | 0.93 | 1.11 | 1.20 |
| 5 | Dirty condenser, airflow blockage | 20°F Hotter condenser | 0.85 | 1.22 | 1.44 |
| 6 | Very dirty filter, dirty condenser | 5°F Colder evaporator<br>10°F Hotter condenser | 0.84 | 1.09 | 1.30 |
| 7 | Very dirty filter, very dirty condenser | 5°F Colder evaporator[d]<br>20°F Hotter condenser | 0.78 | 1.18 | 1.52 |
| 8 | Dirty filter, dirty evaporator, very dirty condenser | 10°F Colder evaporator[c,d]<br>20°F Hotter condenser | 0.69 | 1.15 | 1.65 |
| 9 | Very dirty filter, dirty evaporator, airflow blockage, dirty condenser | 15°F Colder evaporator[c,d]<br>20°F Hotter condenser | 0.62 | 1.10 | 1.79 |

[a]The baseline operational test conditions of 45°F for the suction temperature and 95°F for the condensing temperature were used.

[b]The fan-power change is not considered.

[c]Icing on the evaporator is likely.

[d]Flood-back is likely for units with a capillary tube.

These baseline operating conditions are used in developing the performance factor table. From Table 1 the net capacity $Q(T_{SBL}, T_{CBL})$ is 18.3 tons with a power requirement $P(T_{SBL}, T_{CBL})$ of 14.3 brake horsepower (bhp). If a dirty filter is assumed to decrease the suction temperature by 5°F, then the capacity $Q(T_S, T_C)$ is reduced to 16.6 tons, and the power requirement $P(T_S, T_C)$ is also reduced to 14.1 bhp. The capacity factor is 0.91, the power factor is 0.99, and the energy consumption is increased by a factor of 1.09.

Fortunately, our lower limit for the capacity factor of 0.91 was reached with the assumed 5°F temperature drop in the suction temperature. Were this not the case, a different temperature would have been sought to give this result. Therefore, case 1 in Table 2 will represent the condition of a very dirty filter.

In the manner discussed, the other values in Table 2 can be constructed if reasonable estimates concerning the temperature effects are made. For each case, an operating condition is described, and the temperature effect to produce the condition is selected. Interpolation between values is permissible; extrapolation should not be done. The effects shown are extreme conditions, and intermediate conditions are not considered. The effects should be noticeable by the occupants and may act as a warning that something is awry.

Before proceeding to the other cases, note that the impact of a very dirty filter is not very large in reducing the heat-removal capacity of the air-conditioning unit. However, this small reduction should not be interpreted as meaning that the filter can be used after the filter capacity is exceeded. Eventually, the dirt collected on the front face of the filter will bleed through and be deposited on the evaporator coil.

Because a dirty evaporator can cause icing, the temperature drop was selected just short of ice forming on the evaporator. For case 2 a drop of 10°F was used. From the performance data, $Q(T_S, T_C)$ is 16.1 tons, and $P(T_S, T_C)$ is 12.5 bhp. The capacity factor is reduced to 0.81, the power factor is reduced to 0.98, and the net increase in energy consumption is 20%. Based on these assumptions, a dirty evaporator increases the energy consumption a little more than twice the increase caused by a very dirty filter.

If the airflow over the evaporator is decreased further, ice will begin forming between the fins and will eventually block the airflow. Case 3 represents such a situation.

Small temperature effects on the condenser will not be felt by the occupants. After all, the change in the ambient temperature will produce a larger effect. For effects on the condenser coil, Table 2 considers only incremental changes of 10°F.

Case 4 is for a dirty condenser with an assumed effect of a 10°F increase in the condensing temperature. The net capacity $Q(T_S, T_C)$ is 17 tons, and the power requirement $P(T_S, T_C)$ is 15.9 bhp. The capacity factor is 0.93, the power factor is 1.11, and the energy consumption increases by 20%.

Combined conditions are possible and may appear as a single effect. The filter can be dirty and/or the surface of the evaporator and condenser coils can be dirty and the condenser coil can be blocked by debris. These effects are shown as cases 3 and 5 through 9. Also, case 7 is the condition described by the manufacturer for flood-back with a very dirty filter but with a clean condenser and an ambient temperature of 105°F.

Although cases 8 and 9 will likely cause a mechanical failure for units with a capillary tube, they are included to demonstrate the extreme condition to which a unit with a thermostatic expansion valve could degrade and still function. Before the energy crisis, there was a tendency for air-conditioner installers to greatly oversize units to cut down on the number of callbacks and complaints. Often, the amount of oversizing was about twice the needed capacity. In the cases presented, none of the capacity is reduced by one-half. However, icing on the evaporator coil may eventually form, and the unit might not handle the cooling demand. Case 8 is marginal for icing.

Additional energy consumption because of the longer run times for the blower and condenser-fan motors is not considered. The increased consumption is small compared with the energy required to operate the compressor. For a very dirty filter, case 1, the increase is about ~2%. For the worst conditions, cases 7 through 10, the increased consumption is <10% of the total.

### 4.1.3 Other cooling systems

In large hydronic systems that pipe chilled or heated water to many points of use, the effect of a dirty filter is localized. If there is insufficient airflow over any individual water coil, the heat transferred can be reduced to a level at which the desired comfort cannot be maintained. The water, when returned for recycling, is mixed with the much larger total volume. A single or a small number of units having dirty filters or coils does not greatly affect the suction temperature, and the effect on the system is small. A large number of affected units will be required to have a significant impact.

Air-cooled condensing units should experience the same temperature effects as discussed for the self-contained units. Fortunately, the large units have a tendency to reject debris buildup, and hardly any problem is expected because of a dirty or clogged coil.

## 4.2 Heat Pumps

In the cooling mode the heat pump acts exactly as an air conditioner. The previously described effects and correlation are applicable.

In the heating mode the condenser coil becomes the evaporator coil and vice versa. With a set of performance data and different baseline temperatures, a performance factor table could be constructed. However, the available data cover only a limited range and do not include the type of effects sought. Only a few effects can be calculated.

For a dirty filter with a 20% reduction in the mass airflow, the gross heating capacity factor is reduced slightly by 0.98 with an increase in the power factor to 1.02. The result is an increase of 4% in the energy consumption. With the same reduction in the mass air-flow, the gross cooling capacity is reduced by 0.96, which is less than when the same unit is in the heating mode. This trend applies to all of the different units checked.

An estimate was made for a dirty or clogged outdoor coil by assuming a 10°F drop in the suction temperature. The capacity factor was reduced to 0.81 with a drop in the power factor to 0.95. The energy factor increased to 1.17. The outside coil is not likely to get dirty while in the heating mode. Moisture from the air will condense and will tend to keep the fins washed. However, debris may have accumulated during the fall.

While the unit is in the heating mode, any dirt fouling the inside coil will be baked onto the surface, resulting in a reduction in the heat-transfer capability. If not cleaned before the heating season, the coil will become more difficult to clean.

During extreme cold weather when the auxiliary heater strips are employed, the dirty filter could reduce the airflow enough so that insufficient heat is removed and the temperature rises in the vicinity of the heater strips. A safety cutout switch will cause the heater strips to operate intermittently. There would be little increase in the energy cost, and there may be a decrease because not enough heat is being supplied for comfort.

## 4.3 Furnace

As the air filter becomes clogged, the airflow over the furnace heat exchanger will be reduced, causing the flue gases to be exhausted at a higher temperature. Some of the excess energy content in the exhaust that could have been used to heat the interior is carried away. The composition of the combustion products is not expected to change significantly. As a result, almost all of the effect is centered on the higher flue-gas temperature.

In Ref. 2 the percentage of the energy wasted because of the reduced air velocity across the furnace heat exchanger was calculated. The results are tabulated in Table 3. The calculation considered the change in

Table 3. Percent of energy wasted because of reduced air velocity across furnace heat exchanger

| $V_o$[a] (%) | Energy wasted (%) |
|---|---|
| 100 | 0 |
| 90 | 2.8 |
| 80 | 6.1 |
| 70 | 10.2 |
| 60 | 14.8 |
| 50 | 27.2 |

[a] $V_o$ = initial velocity.

the heat exchanger's effectiveness as a result of the temperature increase and the change in the heat-transfer coefficient for forced convection over parallel plates. The calculated results are used.

## 4.4 Air Handler

### 4.4.1 Blower assembly and motors

Another component affected by a dirty filter is the fan-coil motor. Referring to Figs. 5(*a*) and (*b*) for the forward-curved and the radial impeller fans, there is a decrease in the power required as the filter becomes loaded. However, the reduced energy consumption is offset by the longer run times as a result of the reduced capacity. As shown in Fig. 5(*c*), the backward-curved impeller fan power rises and peaks near a fully clogged filter.

Instead of cycling, the motor could run continuously, thus consuming more energy and adding its heat to the cooling load. If operated long enough, the motor could burn out and require replacing. In many older air conditioners the blower and fan motors should be lubricated annually. Most motor failures are caused by lack of lubrication. Some newer air conditioners have permanently lubricated motors.

### 4.4.2 Dampers

Instead of a dirty filter's affecting dampers, the situation is reversed. A malfunctioning damper adversely shortens the life of filters and significantly increases the direct energy cost. Excess outside air that leaks by the damper blades passes through the filter and must be heated or cooled. Leaking or malfunctioning dampers have been estimated to waste as much as 30% of a plant's energy cost.[11]

# 5. ENERGY SAVINGS VS FILTER-REPLACEMENT SCHEDULING OPTIONS

## 5.1 General

The main question to be answered by this study is "When is it most economical to replace the filter while conserving energy on a HVAC system?" As long as the filter is serving its intended purpose and has not exceeded its useful life, it can be left in place. Obviously, the filter must be replaced when fully clogged. However, there may be some advantages in replacing the filter before clogging. To develop a filter-replacement schedule, the analysis will consider periods less than the filter's lifetime.

Lifetimes for filters can vary greatly. For example, assume a time period during a seasonal change, such as spring, when little heating or cooling is required. If a large source of pollen is close, the filter may become clogged in a relatively short time.

Further, the heating and cooling requirements differ greatly among the many regions of the country. A common practice is running the blower fan by itself to eliminate hot and cold areas caused by air stratification between building levels and to circulate the air for a more uniform temperature distribution. The number of fan hours and the degree of dust in the air — not the amount of energy consumed — will determine how much and how fast the filter will clog. If the amount of airborne dust in two different regions is the same and one of the regions has a much milder climate than the other, the two HVAC systems would still require a filter change at about the same time. Only when a fan is operated in conjunction with the HVAC system's demand would the number of operating hours correlate with the amount of dirt collected on the filter.

For the analytical technique presented, the filter is assumed to be completely clogged after some arbitrary time period. In practice, this time period will need to be determined at the individual site for the various building types, types of occupants, activities, etc. Records or experiences can furnish this necessary information. Normally, this time period is not expected to exceed 3 months.

Although a certain amount of the energy required for heating or cooling can be shown to be conserved by scheduling the filter replacement cycle, another, more-important economic consideration should also be included. A filter replacement schedule should be considered both as an energy conservation measure and as part of the preventive maintenance for the system. A filter that becomes dirty should be replaced, especially before some vital component, such as a motor or compressor, is damaged.

In a life-cycle analysis the cost of replacing the filters is part of the recurring cost of the HVAC system. Nonrecurring cost includes replacement of the fan motor or compressor, either one being much more expensive than the relatively low cost of individual common filters. If the HVAC system is properly maintained, the coils should rarely require the tedious and labor-intensive chore of cleaning. It is easier to keep the coils clean than to have to clean the coils. This analysis will strive to keep these nonrecurring costs low by stressing certain good practices that will show up as constraints in the analysis.

## 5.2 Assumptions and Constraints

Various options are available for analyzing the different schemes in scheduling the filter replacement. These options will be bound by the assumptions that

1. the filter accumulates dirt uniformly over its lifetime and
2. the fan is operated only in conjunction with heating or cooling

and the constraints that

1. the life of the filter ends when the filter is fully clogged and is replaced to prevent damage to the HVAC system and
2. the filter can be replaced anytime before the end-of-life but the period between replacement is at least 1 month. (Realistically, an unusual circumstance, such as remodeling, may shorten this period and require an immediate filter replacement.)

## 5.3 Analytical Technique

The most-accurate and preferred method for this analysis involves the use of periodic energy consumption data for individual buildings or building types. However, on most military installations this kind of information is rarely available. An entire post is usually metered by one meter or by several meters if sections of the post are widely separated. Another method that could be used to generate the energy consumption data would involve the rated capacities of the various systems and estimated operating times. An energy audit for the different types of buildings could be used, but this effort would be tedious.

For the examples shown, the results from a study of the energy-use patterns for certain types of buildings on a military installation will be incorporated.[12] Hourly energy consumption data were analyzed by the U.S. Army Construction Engineering Research Laboratory (CERL) for 70 buildings at Fort Carson, Colorado; Fort Hood, Texas; and Fort Belvoir, Virginia. A regression technique was used on the combined data for classes of buildings. Although the results are not likely to agree for any one particular building, the results are expected to produce fair agreement for a class of building. The slopes from the regression and appropriate heating-degree days (HDD) or cooling-degree days (CDD) values are used to estimate the energy requirements for heating or cooling. An inherent shortcoming is that the many HVAC systems' efficiencies have been included and averaged into the final results and cannot be removed. Also, the survey determined the cooling slopes for only a couple of building types.

For cooling, the incremental increased energy consumption caused by reduced capacity is estimated from the $Q_{DIRTY}/Q_{CLEAN}$ relationship, along with the appropriate airflow rates. For the range of interest, the power required is unchanged. For heating, the values from Table 3 are used. Because a constant flow reduction rate caused by dirt collecting on the filter is assumed, interpolation is done to calculate the intermediate values.

Two examples will be shown for some buildings sited at Fort Bragg. The first example will be for a family housing to demonstrate the effect on cost for single filter of a HVAC system with no labor charges. The assumption is that the occupant will change the filter. The second example will be for a modern barrack that has many filters and a labor charge. Because Fort Bragg purchases bulk filter materials from which the individual replacement filters are cut, no ready cost was available. Therefore, the filter cost and labor charge (recurring nonenergy costs) are taken from Mean's.[13] The filters cost $1.81 each, and the labor rate, including overhead and profit, for an average skilled worker is $227.60/d. Because Mean's costs are based on new construction, for failed components, best guesses were made from a cursory survey of several service shops to determine reasonable repair (nonrecurring) costs.

The tabulated degree-days values (Table 4) for the nearest metropolitan area, Raleigh/Durham, North Carolina, were used.[14] For the transition months this set of calculations assumed no cooling requirements for March and October and no heating requirements for May and September. For electrical consumption the energy cost was calculated by using fuel-cost information supplied by Fort Bragg and the 11,600-Btu/kWh conversion required by the "Energy Conservation Investment Program

Table 4. Heating and cooling degree days for Raleigh/Durham, North Carolina

| Month | HDD, base 65°F | CDD, base 65°F |
|---|---|---|
| Jan | 760 | 0 |
| Feb | 637 | 0 |
| Mar | 500 | 11 |
| Apr | 180 | 14 |
| May | 47 | 122 |
| Jun | 0 | 281 |
| Jul | 0 | 387 |
| Aug | 0 | 356 |
| Sep | 11 | 180 |
| Oct | 185 | 36 |
| Nov | 450 | 0 |
| Dec | 738 | 0 |

(ECIP) Guidance."[15-18] The electricity cost of \$0.02781/kWh converts to \$8.18/million Btu (MBtu). For natural gas the fuel cost is \$5.153/MBtu.

### 5.3.1 Single military housing

From the CERL survey the single family housings' average living space is 1900 $ft^2$. The HVAC system is assumed to be a gas furnace that incorporates an air conditioner in the same unit. The slopes of the fitted data from CERL's regression analysis are 0.00172 $kWh/(ft^2 \cdot CDD)$ for cooling and 16.5 $Btu/(ft^2 \cdot HDD)$ for heating.

The energy consumption, filter cost, and total cost were estimated for various options in replacing the filters at different times and conditions. The results are tabulated in Table 5. The option has a three-

Table 5. Energy, filter, and total cost for a single military family housing of 1900 $ft^2$ at Fort Bragg

| Option[a] | Energy cost (\$) | Filter cost (\$) | Total cost (\$) |
|---|---|---|---|
| 2C3 | 432 | 4 | 436 |
| 3C3 | 426 | 5 | 431 |
| 4C3 | 423 | 7 | 430 |
| 6C3 | 420 | 11 | 431 |
| 1C6 | 435 | 2 | 437 |
| 2C6 | 429 | 4 | 433 |
| 3C6 | 420 | 5 | 425 |
| 4C6 | 419 | 7 | 426 |
| 6C6 | 418 | 11 | 429 |
| 2H3 | 600 | 4 | 604 |
| 3H3 | 582 | 5 | 587 |
| 4H3 | 576 | 7 | 583 |
| 6H3 | 569 | 11 | 580 |

[a]The key to the options is that the first character represents the number of times the filter is changed during the heating or cooling seasons, the letter represents either heating (H) or cooling (C), and the last number represents the number of months required to clog the filter.

character designation of a number, a letter, and another number. The options are grouped by the last two designations and in an increasing order by the first number. The key to the options is that the first character represents the number of times the filter is changed during the heating or cooling seasons, the letter represents either heating (H) or cooling (C), and the last number represents the number of months required to clog the filter. For example, if 3 months are required before the filter is clogged, option 2C3 states that the filter is changed twice during a 6-month cooling season. Because the arbitrary period is 3 months and March was stated as a month that does not require cooling, the filter is changed at the beginning of July. A clean filter is assumed at the start of any arbitrary period. All of the filter changes are spaced equally except for those options that have four filter changes. This option is an attempt to reduce the filter cost by not changing the filter for the mild months. For example, option 4C3 uses the same filter for April and May; the filter is changed at the beginning of the summer months of June, July, and August; the filter is not changed for September. This option also allows a buffer time that can be used to apply preventive maintenance, such as oiling the fan motors, which should be lubricated once a year.

By chance, if the filter does not become completely clogged until the end of a 6-month cooling season, the option with the last character "6" is calculated. An arbitrary period of 6 months is not believed likely for heating because of occupants' tendency to stay indoors more, the air dryness, and the electrostatic charge effects that tend to increase the amount of dispersed dust. Therefore, only one period of 3 months was considered.

For heating, the airflow lower limit is assumed to be the same as that used in the cooling estimates. The filter is completely clogged whether it is used in the heating or cooling cycle. A lower airflow rate is likely to cause the furnace bonnet to overheat, thus activating a safety thermostat. At the beginning of the heating season, October, a clean filter is installed. If a filter is to be replaced at the beginning of April, its cost is added into the cooling option to avoid double costing.

In general and as expected, the energy cost decreases with an increasing number of filter changes. The filter cost naturally follows the number of filter changes. However, none of the total costs are significantly different.

For cooling with a 3-month filter lifetime, option 4C3 was somewhat successful in minimizing the total cost but not enough to warrant recommending. Although options 3C3 and 6C3 are nearly equal in total cost, option 6C3 is recommended because of the greater energy reduction. During the cooling season, option 6C3 is simply monthly replacement of the filter, a schedule that supports the manufacturer's usual recommendation.

The 6-month filter lifetime option demonstrates the effect that if the filter is not replaced often enough, the extra energy cost will be more than the lower filter cost. If the filters are replaced too often, the cost of the filters will be more than the energy cost reduction. Option 3C6 would be recommended because the energy cost is near the minimum and the filter cost is not excessive.

As seen in the heating options 2H3 through 6H3, the heating energy costs are higher than the cooling costs. Although the British-thermal-unit price is lower, a greater amount of energy is consumed. Results of lower energy consumption will show more readily. As expected, there is a steady decrease in the energy costs with each increase in the number of filter replacements. Option 6H3 is recommended because of its lowest energy consumption and total cost. Again, monthly filter replacement is recommended.

For an installed HVAC system these options can be put into practice for cost savings. For proposed systems this technique can be applied in estimating the annual recurring cost in the life-cycle cost analysis (an example is shown in Table 6). Using the example presented here, filters are recommended for replacement every month at an annual cost of $22. When discounted at the required 7% inflation rate over the estimated 15-year lifetime of the equipment, the discounted cost is $198.

Documented studies of compressor failures performed by American Electric Power Company and by Alabama Power[19] imply that compressors will fail and must be replaced on an average of every 8 years. No data

Table 6. Life-cycle analysis summary
Energy Conservation Investment Program (ECIP)

LOCATION: Fort Bragg REGION NO. 4 PROJECT NUMBER Example 1

PROJECT TITLE More Efficient HVAC System FISCAL YEAR 86

DISCRETE PORTION NAME Filter Replacement Program

ANALYSIS DATE Nov. 85 ECONOMIC LIFE 15 YEARS PREPARED BY L. Jung

1. INVESTMENT
   A. CONSTRUCTION COST $______
   B. SIOH $______
   C. DESIGN COST $______
   D. ENERGY CREDIT CALC (1A+1B+1C)X.9 $______
   E. SALVAGE VALUE -$______
   F. TOTAL INVESTMENT (1D-1E) $______

2. ENERGY SAVINGS (+) / COST (-)
   ANALYSIS DATE ANNUAL SAVINGS, UNIT COST & DISCOUNTED SAVINGS

| FUEL | COST $/MBTU(1) | SAVINGS MBTU/YR(2) | ANNUAL $ SAVINGS(3) | DISCOUNT FACTOR(4) | DISCOUNTED SAVINGS(5) |
|---|---|---|---|---|---|
| A. ELEC | $ 8.18 | | $ | | $ |
| B. DIST | $ | | $ | | $ |
| C. RESID | $ | | $ | | $ |
| D. NG | $ 5.15 | | $ | | $ |
| E. COAL | $ | | $ | | $ |
| F. TOTAL | | | $ | | ---------->$ |

3. NON ENERGY SAVINGS(+) / COST(-)
   A. ANNUAL RECURRING (+/-) $ -22
      (1) DISCOUNT FACTOR (TABLE A) 9.11
      (2) DISCOUNTED SAVING/COST (3A X 3A1) $ -198

   B. NON RECURRING SAVINGS(+) / COST(-)

| ITEM | SAVINGS(+) COST (-)(1) | YEAR OF OCCURRENCE(2) | DISCOUNT FACTOR(3) | DISCOUNTED SAVINGS(+) COST(-)(4) |
|---|---|---|---|---|
| a. | $ -800 | 8 | .58 | $ -466 |
| b. | $ -200 | 10 | .51 | $ -102 |
| c. | $ | | | $ |
| d. TOTAL | $ -1000 | | | $ -568 |

   C. TOTAL NON ENERGY DISCOUNTED SAVINGS(+) / COST(-) (3A2+3Bd4) $ -766

   D. PROJECT NON ENERGY QUALIFICATION TEST
      (1) 25% MAX NON ENERGY CALC (2F5 X .33) $______
         a IF 3D1 IS = OR > 3C GO TO ITEM 4
         b IF 3D1 IS < 3C CALC SIR = (2F5+3D1)÷1F= ______
         c IF 3D1b IS = > 1 GO TO ITEM 4
         d IF 3D1b IS < 1 PROJECT DOES NOT QUALIFY

4. AVERAGE ANNUAL DOLLAR SAVINGS 2F3+3A+(3B1d ÷ YEARS ECONOMIC LIFE) $______

5. TOTAL NET DISCOUNTED SAVINGS (2F5+3C) $______

6. DISCOUNTED SAVINGS RATIO (IF < 1 PROJECT DOES NOT QUALIFY) (SIR)=(5÷1F)= ______

were available for fan motors, and a conservative 10 years is used. For a 2 1/2-ton compressor the total replacement cost is \$800. Fan motor costs vary widely; the total replacement cost is estimated at \$200.

Discussions with representatives at Fort Bragg show that one of the benefits already experienced in their aggressive filter replacement program is a reduction in the number of fan-coil motor replacements. Clogged filters do shorten the life of motors by increasing the load and operating times. Temperature is probably the main culprit. If the failure occurs during the unit's early life, the replacement cost for one motor nearly equals the discounted cost for all of the filters throughout the unit's life. As discussed previously, filter abuse can cause a compressor failure for units with capilliary tubes. Regular filter replacement can contribute to a longer life for compressors and motors and can be very cost-effective if early failure is prevented.

### 5.3.2 Modern barrack

A large, modern barrack of 39,500 ft$^2$ will be used as an example. The HVAC system is mounted on the roof. Inside the air handler are eight filters that filter the make-up air and the return air. Conditioned air is ducted to each room and returned through a plenum. Each room has an air return with a filter across the opening. No survey was made of the total room filters; 150 will be assumed. One man-day is assumed to be needed to replace all of the room filters. The slopes of the fitted data from CERL's regression analysis, are 0.00149 kWh/(ft$^2$•CDD) for cooling and 7.4 Btu/(ft$^2$•HDD) for heating. For this analysis, natural gas is assumed for heating.

To estimate the energy cost savings, it is neither practical nor necessary to consider the incremental condition of each filter at various life spans. However, the maximum energy cost savings can be estimated from the cost difference of the energy cost if all of the filters stay clean and the energy cost if all of the filters are clogged. For cooling, the maximum cost is \$8139, the minimum cost is \$7493, and the maximum energy cost savings is \$646. For heating, the maximum cost is \$6107, the minimum cost is \$5203, and the maximum energy cost savings is \$904.

Because all of the conditioned air must pass through the eight filters inside the air handler, these filters are of primary concern. Assuming 1 h of labor is required, each change of these filters is $43. A complete filter change, including the room filters, is $514 with about one-half being the labor cost. Because the maximum energy cost savings for cooling is $646 and for heating is $903, a few complete filter changes will exceed these savings. The air-handler filters filter outdoor air before distribution throughout the building and prevent the room filters from becoming clogged too rapidly. These filters can be changed each month if necessary and not use up the savings. Experience from observation will be necessary to determine when to change the individual room filters. Those filters on the upper levels may not need to be changed as often as those on the lower levels. However, as stated previously, filters should be replaced when they become clogged. If the room filters are clogged too soon, a longer-life pleated filter is an alternative to consider.

## 6. OTHER OBSERVATIONS, CONCLUSIONS, AND RECOMMENDATIONS

Concurrent with the literature search, additional information was sought by following a filter-changing crew and observing the condition of filters and evaporator coils in a large building at one of the Oak Ridge plants. The filters had about 2-months' usage, and the air conditioners were mainly window units and some larger floor units. All of the units have had many years of service. None of the filters was very dirty but, more surprisingly, most of the evaporator coils were shiny clean. Only three coils were dirty, and they are located in rooms that are rarely occupied and are kept locked. Later, two more units with dirty coils were found in occupied offices that keep the doors shut most of the time. All of the other units were in offices or work areas with doors opened to hallways. Although the sample size is small and the observed effects are unexpected, some speculations about cause are made.

The reason for the clean evaporator coils appears to be the wash-down effect from the condensate. During the summer the many large entrances of the building are kept open, and the air infiltration rate is high in the experimental areas. The office areas are separated by closed doors that are opened many times during the day because of people traffic. This traffic is the main reason for the high air infiltration into the office areas. Because the outside air has a high humidity, the condensate appears to be sufficient in keeping the coils clean. The filters by themselves cannot account for the cleanliness. Because of the particle sizes, most of the dust cannot be stopped by the type of filters used.

The search for an explanation for the dirty coils has shown what seems a common denominator. Rarely are the doors to those rooms opened. Thus, moisture in the air is greatly reduced. These evaporator coils do not have the wash-down effect of the other units, and dirt collects. A dirty condenser coil was also found on one of the window units, and the dirt has continued to build.

Some speculations about the mechanism that causes the evaporator coil to become clogged are made. Coils located in areas that can produce sufficient wash-down have been known to clog. Therefore, wash-down is not sufficient to ensure that a coil will not get dirty. There appears to be a correlation between a coil's beginning to get dirty and the fact that a filter has been used too long. As the filter becomes clogged, the airstream will bypass around the edges of the filter. This fact is evidenced by observing that when a dirty evaporator is found, the edges are usually the first place where dirt begins to accumulate. Also, as the filter becomes clogged, the airflow velocity on the face of the evaporator is greatly reduced. There is some bleed-through of the dirt, and the airstream does not have sufficient velocity to blow the dirt off of the evaporator's surface. When wet by the condensate, the accumulation mats and continues to build. If this mechanism is correct, it is important to replace the filter before the evaporator coils become coated with dirt.

These observations have generated two questions: (1) how valid is the concept that high-quality air filters are needed? and (2) what minimum quality of air filter (and what conditions) is necessary to keep the evaporator surface clean? If a lower-quality, inexpensive filter is suitable, additional savings can be realized by reduced blower-fan requirements and by the use of less-expensive filters. A loosely woven fiberglass filter with a resin coating could be used to keep large particle-size pollutants from reaching the evaporator coil. Because the single-family filter analysis recommended replacing the filter monthly, an inexpensive filter should be sufficient because it is to be replaced before any accumulated dust bleeds through the filter.

A general purpose, low-quality air filter may be employed in

1. areas that do not require any high-quality filtered air,
2. areas that experience high humidity,
3. units that do not run the blower fan when the compressor is not operating (if the humidity level and the heat load are sufficient to maintain a high percentage of operating time, the blower may be left running),
4. units that have the evaporator coil mounted vertically so that the condensate drips over the layers of tubes, and
5. areas that have high infiltration rates.

There may be more conditions and limitations that could be determined from experience or testing.

Conditions under which high-quality filters should be left in place are

1. work areas that have a large amount of dust;
2. areas requiring a relatively dust-free condition, such as computer rooms, medical facilities, food service areas, and bowling alleys; and
3. areas near coal- or oil-fired power plants.

It must be emphasized that there is not enough evidence to support these speculations. However, it may be worthwhile to conduct a limited test by monitoring filters and air conditioners to determine if these speculations have merit. A large majority of the filters used might be replaced by these inexpensive filters. If there should be additional follow-up work, the areas suggested for investigating are

1. locating units with dirty evaporator coils and determining the cause and
2. testing the concept of using an inexpensive filter to screen out large particle-size pollutants while maintaining a clean evaporator coil.

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