Fan Heat and Pump Heat:

Sources and Significance

How these two thermodynamic phenomena—long thought to be similar—occur quite differently in air and hydronic systems

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Editor's note: This is Part 2 of a two-part series. Part 1 was published in August.

Last month, we learned that, in a typical duct system, the temperature of an air stream undergoing frictional and dynamic losses does not change. What's more, we learned that, in a constant-area duct system, the temperature of an air stream experiencing frictional pressure loss actually decreases slightly (but, for all practical purposes, immeasurably). This month, we will look at the thermodynamic characteristics of fans, where energy is introduced to a system, and where temperature rise is experienced in an air stream. In addition, we will examine the parallel concept of pump heat in hydronic systems and address the significance of both it and fan heat in modern HVAC-system design and analysis.

FAN ENERGY

The fan is the source of energy input (or, more accurately, power input) in an HVAC air system, increasing pressure to overcome the effects of friction, turbulence, and flow obstructions (filters, coils, etc.) in the ductwork. Figure 5(a) depicts a centrifugal fan mounted in a section of supply and return ductwork. For simplicity, the inlet and outlet air velocities are equal.

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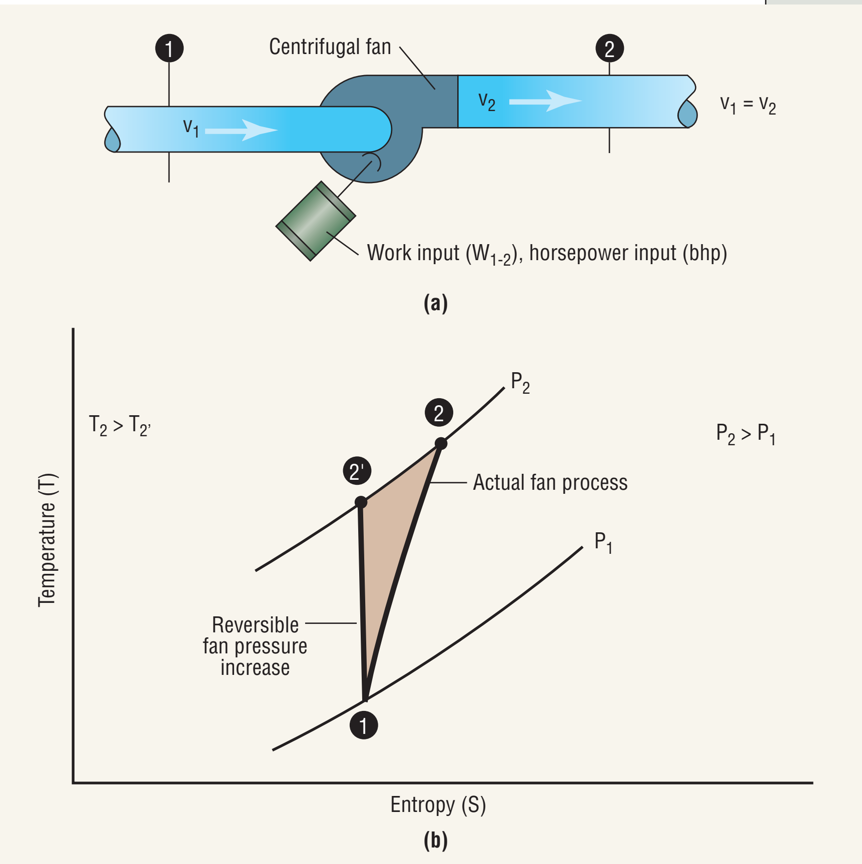

FIGURE 5. A centrifugal fan mounted in a section of supply and return ductwork (a) and a temperature-entropy diagram showing fan processes (b).

(A difference between the inlet and outlet air velocities would have a minimal effect on calculations, as the heat associated with velocity-pressure changes across a fan is quite low.[7]) Input power appears at the fan as heat of compression attributable to the elevation of air-stream pressure from P_1 to P_2, as well as heat rise resulting from frictional losses attributed to fan-wheel inefficiency. Heat of compression for an adiabatic, reversible (isentropic, or frictionless) pressure increase from P_1 to P_2 is given by Equation 10, shown by process 1 to 2'. The ideal (isentropic) horsepower required for the process[8] is given by the equation:

$$(hp)_{ideal} = \frac{k}{k-1}\left(\frac{P_1 cfm_1}{33{,}000}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \quad (14)$$

The actual process, including friction and inefficiencies, is given by the process 1 to 2. The actual horsepower (fan bhp) is defined by the following equation:

$$\eta_f = \frac{(hp)_{ideal}}{bhp} = \frac{T_{2'} - T_1}{T_2 - T_1} = \frac{\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right]}{\left[\frac{T_2}{T_1} - 1\right]} \quad (15)$$

The total temperature rise of the actual process is given by the equation:

$$\Delta T_{1-2} = \frac{T_1}{\eta_f}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right] \quad (16)$$

The total temperature rise across the fan in terms of the total pressure rating is given by the equation:[9]

$$\Delta T = \frac{(TP_2 - TP_1)}{36\ \rho_1\ \eta_f} = \frac{\Delta TP}{36\ \rho_1\ \eta_f}$$

For air at standard conditions, this equation reduces to:

$$\Delta T_{1-2} = \frac{(TP_2 - TP_1)}{2.7\,\eta_f} = \frac{\Delta TP}{2.7\,\eta_f} \quad (17)$$

To summarize, the energy input delivered to the fan system (fan bhp) appears as heat only at the location of the fan. The total work input delivered to the fan shaft is converted to heat, resulting in a temperature rise across the fan, as indicated by Equation 17. For a 70-percent-efficient fan, the rise in temperature is 0.53 F per inch water gauge of total pressure rise across the fan. If the electric motor and drive assembly are located in the conditioned-air stream as well, their heat contribution (motor inefficiencies and drive losses) will appear as additional fan-heat rise. Although motor losses should be determined for the specific motor utilized, they can be approximated from ASHRAE Handbook–Fundamentals.[10] Drive losses may be approximated based on the drive-system power-transmission characteristics described in AMCA Publication 203-90, *Field Performance Measurement of Fan Systems*.[11]

HYDRONIC-SYSTEM FRICTION LOSSES AND PUMP HEAT

Because the fluid flowing in a hy-

Nomenclature

Q_{1-2}	Heat added to a system (fluid) between conditions 1 and 2 (British thermal units per pound or British thermal units)
$U, \Delta U$	Internal energy or change in internal energy in a process (British thermal units per pound or British thermal units)
W_{1-2}	Work done by a system between states or locations 1 and 2 (foot-pounds per pound or foot-pounds)
J	Mechanical equivalent of heat (778.2 ft-lb per British thermal unit)
$T, \Delta T$	Fluid temperature or temperature change in a process (degrees Fahrenheit or degrees Rankine)
C_p	Specific heat capacity at constant pressure (British thermal units per pound, degrees Rankine)
C_v	Specific heat capacity at constant volume (British thermal units per pound, degrees Rankine)
m	Mass of fluid (pounds)
V	Volume (cubic feet)
P	Fluid pressure (pounds per square foot)
ρ	Fluid density (pounds per cubic foot)
v	Fluid specific volume (cubic feet per pound)
Z	Elevation of fluid above a datum plane (feet)
V	Fluid velocity (feet per second)
g	Acceleration of gravity (32.2 ft per second squared)
$h, \Delta h$	Fluid enthalpy or enthalpy change (British thermal units per pound or British thermal units)
k	Ratio of specific heat capacities ($C_p \div C_v$) (dimensionless)
$S, \Delta S$	Fluid entropy or change in fluid entropy (British thermal units per pound, degrees Rankine)
R	Gas constant (for air, 53.3 ft-lb per pound, degrees Rankine)
hp	Horsepower (33,000 ft-lb per minute)
cfm	Volumetric airflow (cubic feet per minute)
$TP, \Delta TP$	Total pressure or rise in total pressure (inches water gauge)
η_f	Fan efficiency (dimensionless)
ΔP	Head loss in water stream (feet) or air stream (inches water gauge)
bhp	Actual shaft horsepower input to a process
Hd	Head across a pump (feet)
η_p	Pump efficiency (dimensionless)
Q	Volumetric flow rate (gallons per minute)

dronic pumping system is not a gas, but essentially incompressible liquid water, the manner in which energy input appears as heat gain is fundamentally different than it is in an air-duct system. However, the divergence lies only in the location at which the heat appears.

Figure 6 depicts a basic hydronic system, including a water pump and a section of discharge piping, operating against the effects of friction loss (with no elevation differences). Applying the steady-flow energy equation (Equation 2) along the piping system, between stations 2 and 3:

$$\frac{W_{2-3}}{J}+Q_{2-3}+\frac{P_2 v}{J}+U_2+\frac{V_2^2}{2gJ}=\frac{P_3 v}{J}+U_3+\frac{V_3^2}{2gJ} \qquad (18)$$

With a constant-area discharge pipe, no external work input, and adiabatic flow with no external heat transfer ($V_2 = V_3$, $W_{2\text{-}3} = 0$, $Q_{2\text{-}3} = 0$), this equation reduces to:

$$\frac{P_2 v}{J}-\frac{P_3 v}{J}=\left(U_3-U_2\right) \qquad (19)$$

In this equation, the term $(U_3 - U_2)$ represents the mechanical energy that is converted to thermal energy because of the viscous dissipation and irreversibilities associated with frictional flow in the fluid stream. This term (h_L) represents the amount of mechanical energy converted to heat because of the frictional-loss process. This loss relates directly to the loss of pressure, which occurs in the fluid system as follows:

$$h_L=U_3-U_2=\frac{v}{J}\left(P_2-P_3\right) \qquad (20)$$

In the absence of heat transfer, the work done against friction leads to an increase in internal energy. For a liquid, this increase in internal energy is associated with an increase in temperature. With the change in internal energy between stations 2 and 3 defined as $C_v (\Delta T)$, the temperature rise attributed to flow across a frictional pressure drop

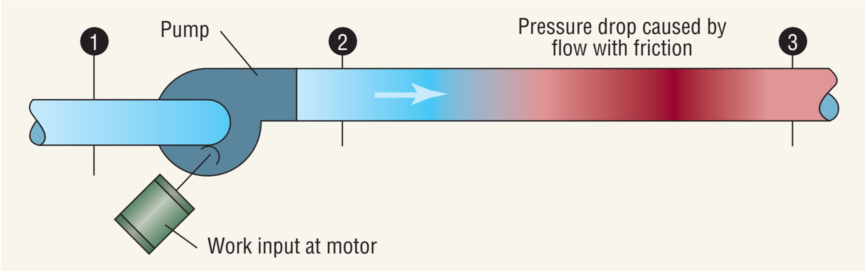

FIGURE 6. A basic hydronic system operating against the effects of friction loss.

is given by the equation:

$$\Delta T_f = \frac{v}{JC_v}\left(P_2 - P_3\right) \qquad (21)$$

Thus, quite unlike in an air system, heat rise attributed to frictional losses in a hydronic system occurs at the point of frictional pressure drop. In a run of piping with a progressive pressure drop, heat rises gradually in proportion to pressure drop. Because of water's density, water-stream temperature rise is proportionately smaller than air-stream temperature rise. For a chilled- or heating-water system with water as the working fluid and frictional pressure drop given in feet of head, the temperature rise attributed to friction is given by the equation:

$$\Delta T_f = \frac{\Delta P}{777} \qquad (22)$$

Thus, for a frictional pressure drop of 200 ft of head, a temperature rise of 0.26 F will be experienced.

The centrifugal pump installed between points 1 and 2 in Figure 6 supplies energy to increase fluid pressure head and overcome losses and elevation differences. The energy added to the fluid by the pump is obtained from the steady-flow energy equation, rewritten in terms of fluid head per pound of fluid flowing:[12]

$$W_{1-2} = \frac{\left(V_2^2 - V_1^2\right)}{2g} + \frac{\left(P_2 - P_1\right)}{\rho} + \left(Z_2 - Z_1\right) \qquad (23)$$

The increase in head across the pump results largely from pressure head. Ideal power is given by the equation:

$$\left(hp\right)_{ideal} = \frac{\rho Q W_{1-2}}{550} \qquad (24)$$

With water at 70 F, volumetric flow rate in gallons per minute (gpm), and work input in feet of head (Hd) (foot-pounds per pound flowing), ideal horsepower input is given by the equation:

$$\left(hp\right)_{ideal} = \frac{\left(gpm\right)\left(Hd\right)}{3,960} \qquad (25)$$

The total power actually added to the fluid by the pump impeller is greater than the ideal horsepower because of inefficiencies in the process. These inefficiencies are the result of secondary flow within the pump impeller and passages, flow separation, friction, and flow turbulence. Actual horsepower input, then, is based on overall pump efficiency (η_p) as follows:

$$\eta_p = \frac{\left(hp\right)_{ideal}}{bhp} \qquad (26)$$

The energy lost because of inefficiency within the pump is converted to thermal energy and transferred to the liquid passing through the pump, increasing the temperature of the liquid. Using equations 21 and 22 and with pump head expressed in feet:

$$\Delta T_{across\ pump} = \left(\frac{Hd}{777}\right)\frac{1-\eta_p}{\eta_p} \qquad (27)$$

Thus, the temperature rise attributed to impeller inefficiencies measured across a 70-percent-efficient pump rated at 200 ft of head is 0.11 F. With the 0.26-F rise attributed to flow friction, the total temperature increase is 0.37 F.

SUMMARY AND SIGNIFICANCE

Table 1 summarizes properties related to fan heat and pump heat. In an air system, temperature rise attributed to energy input occurs almost entirely at the fan. Temperatures within the duct system remain constant, despite friction- and turbulence-induced pressure drops. In a hydronic system, temperature rise attributed to energy input occurs in the piping system at the location of frictional pressure drops and at the pump because of inefficiencies within the pump impeller.

The significance of fan heat is apparent in system design. In a blow-through cooling-fan-system configuration, fan heat is equivalent to a sensible-heat gain in the return air and, thus, shows up as increased coil entering-air temperature and an additional load imposed on the cooling coil. In a draw-through cooling-fan-system configuration, fan heat is equivalent to a sensible-heat gain in the space by directly increasing the supply-air temperature and reducing the ΔT available for space cooling. With either configuration, if the motor and drive are located within the air stream, these losses will add to the fan heat. If a draw-through cooling system is designed for a temperature differential (room temperature minus supply-air temperature) of 20 F, a 4-in.-wg pressure requirement in the fan system would increase the design sensible-cooling load by more than 10 percent. For a constant-volume system, this load is constant. At the extreme of low space load—say, 10 percent—the air-handling system will increase the imposed load by 100 percent, or, if the average load is 60 percent, the fan heat

System type	Total temperature rise, F	Location and amount of temperature rise attributed to energy input, F: At fan/pump	Location and amount of temperature rise attributed to energy input, F: In ductwork/piping system
Air	$\frac{\Delta TP}{(2.7)\eta_f}$	$\frac{\Delta TP}{(2.7)\eta_f}$	0
Hydronic	$\frac{Hd}{(777)\eta_p}$	$\frac{Hd}{777}\left(\frac{1-\eta_p}{\eta_p}\right)$	$\frac{Hd}{777}$

TABLE 1. Heat-rise summary.

will increase the system cooling energy introduced to the refrigeration compressor by 17 percent.[13]

Building-system designers must be aware of the load imposed by not only the supply fan, but the return fan. The psychrometric impact of fan heat must be considered when establishing space air quantities and determining actual entering- and leaving-air temperatures in cooling-coil selection.

With high-pressure fan systems, rather striking improvements can be made with small changes in required airflow because of the large magnitude of the fan-heat component. Consider a fan system operating at 10,000 cfm and 10-in. total pressure with an 85-percent-efficient motor in the air stream, a 70-percent-efficient fan, a 52-F fixed temperature leaving the cooling coil with the fan in a draw-through position, and a desired space temperature of 75 F. Table 2 compares the actual cooling capacity with changes in airflow quantity (as the flow requirement decreases, so does the fan horsepower and, thus, the fan heat by the third power of the air-volume reduction). With higher-pressure systems, the effect is more pronounced.

To illustrate this effect, Figure 7 compares fan heat at design airflow and at 20-percent above and 20-percent below design airflow. At design airflow, temperature rise across the fan is given by Equation 17. Because, in this example, the motor is in the air stream, losses from the motor must be included. (Drive losses are not included in this example.) Therefore, temperature rise across the fan is:

$$\Delta T = \frac{10}{2.7 \times 0.7 \times 0.85} = 6.2\ F$$

The useful sensible cooling available to the space is calculated with the sensible-heat-capacity equation. The same procedure was used for airflow 20-percent above and 20-percent below design; however, system total-pressure requirements were assumed to vary by a system characteristic curve of $\Delta TP \sim (Q_v)^2$. For example, at 12,000-cfm airflow, the system total-pressure requirement is:

$$10.0 \times \left(\frac{12,000}{10,000}\right)^2 = 14.4\ in.\ wg$$

Percent change in airflow	Percent change in fan power	Percent change in useful sensible cooling
+20	+72.8	+0.4
+10	+33.1	+1.4
Design (10 in. wg)	—	—
-10	-27.1	-3.6
-20	-48.8	-9.3
-30	-65.7	-16.7

TABLE 2. Changes in fan power and space sensible cooling vs. changes in airflow above and below design.

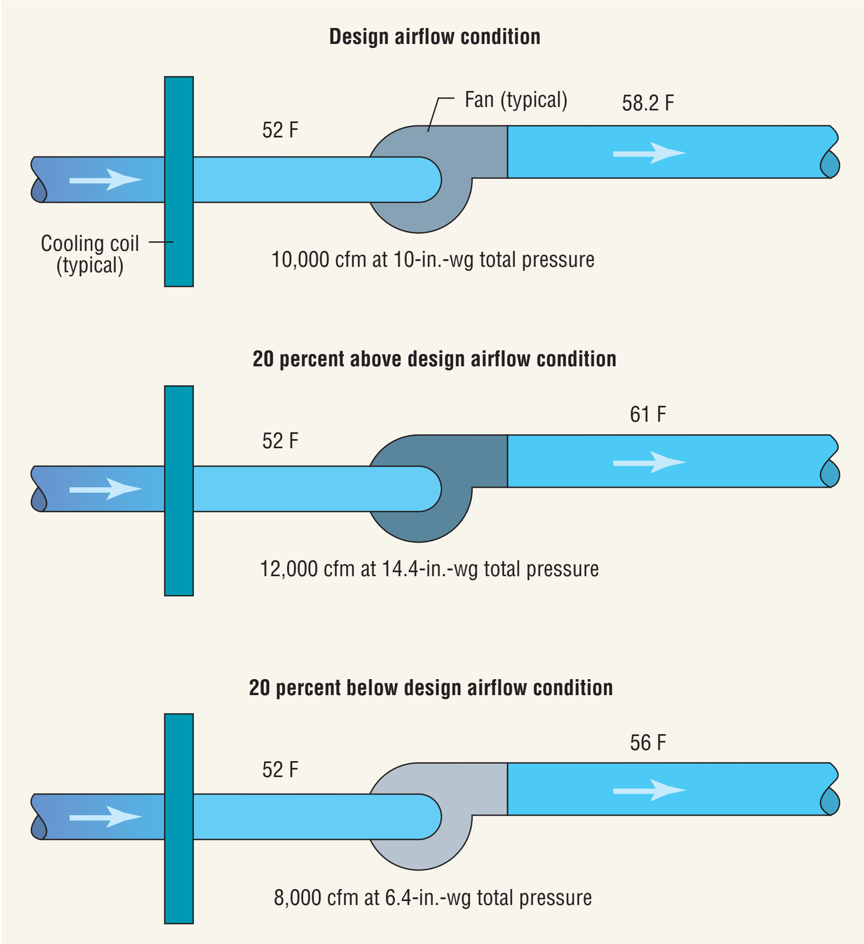

FIGURE 7. Fan-heat comparison.

Although the temperature rise produced in a hydronic system is proportionally smaller than that in an air system, the heat input still must be considered in overall energy requirements. For a chilled-water-system application with 1,000 full-load hours of cooling per year, 2-gpm-per-ton chilled water circulated at 75-ft pump head, and 70-percent overall pump efficiency, pump heat accounts for about 10 percent of the total ton-hours of cooling furnished by the chiller plant.

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