Save That Older STEAM SYSTEM

How variable-vacuum-differential heating control can improve performance and save money

Variable-vacuum differential heating-control technology has been around in various forms for some 70 years and permits significant improvement to older, inefficient systems without requiring the removal of existing pipes and mains. These controls can yield savings of up to 40 percent when compared to an uncontrolled two-pipe condensate-return system (atmospheric return) and up to 25 percent when compared with a controlled system.

Let's examine the case of a 15,000 equivalent-direct-radiation (EDR) capacity system (broken down as 9,750 EDR terminal units plus 2,250 EDR piping heat gain plus 3,000 EDR internal heat gain), where we desire a 70-F room temperature with a 0-F outside temperature at design load and assume 15-percent supply-and-return pipe heat gain and 20-percent internal heat gain (occupied cycle=8 hours, convector radiation, individual auto control valves). We can chart the steam-heat usage, on an annual basis, as shown in Table 1.

TABLE 1. ***Heat wasted due to uncontrolled steam temperature.***

Outside temperature range (F)	A Necessary Btuh for mean of temperature range	B Number hours per year between this temperature range	A × B Necessary Btu per year for this temperature range (M)	C Wasted Btuh for mean of temperature range	B × C Necessary wasted Btu per year for this temperature range (M)
0-5	$12{,}000 \times \frac{52.5}{55}$	36	412	$2{,}250 \times \frac{2.5}{45}$	4
5-15	$12{,}000 \times \frac{45}{55}$	156	1532	$2{,}250 \times \frac{10}{45}$	78
15-25	$12{,}000 \times \frac{35}{55}$	449	3429	$2{,}250 \times \frac{20}{45}$	449
25-35	$12{,}000 \times \frac{25}{55}$	1332	7265	$2{,}250 \times \frac{30}{45}$	1,998
35-45	$12{,}000 \times \frac{15}{55}$	1324	4333	$2{,}250 \times \frac{40}{45}$	2,648
45-55	$12{,}000 \times \frac{5}{55}$	1166	1272	$2{,}250 \times \frac{1}{2}$	1,312
55-70	$12{,}000 \times 0$	2117	0	2,250	4,763
	Total		18,243		11,252

The total annual steam usage then becomes (A x B) + (B x C) =18,243,000 + 11,252,000 = 29,495,000 Btu per year. And the percentage of wasted steam heat is 11,252,000 ÷ 29,495,000 = 38 percent.[1]

In a more recent boiler control retrofit of a 231,000-sq-ft building in the Northeast using variable-vacuum-differential heating control, a payback of approximately 11 months was documented.[2] Of course, the greatest opportunity for savings is in those climes where the total heating degree days are not extremely high, because this type of control is of most benefit on a mild day during the heating season.

To understand the variable-vacuum-differential heating method of steam heat control, it is important to understand the fundamentals of steam-heating. Saturated steam has, of course, a certain temperature associated with it at any given pressure. Steam tables[3] and Mollier diagrams show the temperatures of saturated steam to vary from 32 F to 705 F for sub-critical pressure values ranging from 0.089 psia to 3,208 psia. So-called "low-pressure" steam systems, those operating at 0 to 15 psig, will produce steam having temperatures of approximately 212 F to 250 F. Because of the energy savings available by returning heated condensate, condensate pumps are frequently used in these systems and over the years have frequently replaced vacuum return systems.

The vacuum return systems were commonly used on low-pressure steam heating systems until the 1970s, when their popularity—for whatever reason—began diminishing. One leading supplier reports the sale of more than 60,000 vacuum pumps for steam heating systems since 1921, with most (about 50,000) purchased prior to 1980. This is unfortunate because heating systems with vacuum pumps will heat more quickly and evenly at (obviously) lower steam pressures than those without them. And by controlling the level of vacuum, a relatively precise temperature control of the steam can be obtained. As a practical matter, vacuum return systems in steam heating will generally provide vacuum down to 25 in. Hg. So for an operating range of, say, 25 in. Hg vac to 2 psig, the temperature of the steam can be controlled at any value between 130 F and 220 F, which is very similar to the temperature range of a hot water heating system. There is, however, an important distinction between steam and hot-water over this range of temperatures: The steam contains a relatively constant amount of latent heat over this entire temperature range, while the hot water contains only sensible heat. So boiling water under a vacuum produces lower-temperature steam without sacrificing heat transfer. Because the volume of the steam under vacuum is expanding, the terminal units (e.g., radiators and convectors) will be filled with this lower-temperature steam. The rate of heat transfer is directly proportional to the ΔT, as shown in the equation below, which results in a reduction of external heat losses:

$q = hA\Delta T_m$

where: q = rate of heat flow in Btu per hour

h = coefficient of heat transfer in Btu per hour-sq ft-F

A = area of the path normal to the direction of heat flow in sq ft

ΔT_m = logarithmic mean temperature difference

If we then control this vacuum order to balance the production of steam to the real time demand for heat in the building (by monitoring both the outside temperature and the actual heat losses from the structure), we can require the steam boiler to produce only that amount of steam actually required to replace the heat lost from the building. We have now effectively converted the old, inefficient heating system to a highly efficient demand basis, with temperatures being controlled over essentially the same comfort range as that of a new hot water heating system. And we've done it at a significant savings in cost.

Some of the more obvious efficiency gains come from the reduction in external heat piping losses; the significant reduction in the operating pressure of the steam, which results in shorter cycle times with an "on-off" burner and lower firing rates in a modulating burner; lower standby losses (due to the lower boiler temperature); and increased steam-trap life – again, because of

Candidates for Variable-Vacuum Control

So how can you determine if your building is a good candidate for upgrading with this type of system? The most important criteria are that: (1) the existing heating is a two-pipe, low-pressure steam system; (2) the integrity of the existing boiler(s), piping, and terminal units is sound; and (3) the climatic conditions are such that the system does not continuously operate at (or near) the design point all season. Variable-vacuum differential heating controls will generally find their most effective application in:

- Schools, where outside ventilation air is being introduced, the occupancy and heat gain in individual rooms is varying, there are periodic requirements for larger quantities of outside air (during milder-temperature conditions), and draft prevention is important.
- Buildings with occupancy zoning, such as an office building with retail stores on the ground level, where the heat to the offices can be reduced at 5 or 6 p.m., but the stores require heating until later in the evening and on the weekends.
- Industrial buildings, where warehouse space requires minimal heating, manufacturing space requires somewhat more comfort heating, and the offices require a significantly greater amount of heat.
- Hospitals, where both varying-occupancy and temperature-setback issues may be encountered
- Large multiple-occupancy housing complexes, particularly in high-abuse locations where customer satisfaction helps to reduce unauthorized adjustment, tampering, and damage to equipment.

the lower operating temperatures. Also, if a burner retrofit is part of the steam-system renovation, then replacement with a less costly, lower-turndown burner may also be appropriate because there is no inherent efficiency advantage in using burners capable of high turndown ratios for this type of application.[4]

Perhaps the simplest way to describe this variable-vacuum-differential control system is with an example. Figure 1 is a graph based on a two-pipe system with condensate return at atmospheric pressure.

The system is capable of producing approximately 4,800 lb per hr of steam or 15,000 sq ft of EDR. If we assume that a room temperature of 70 F is desired and that the outside temperature is 0 F, this becomes our maximum heating requirement scenario. Clearly, then, if the outside temperature warms to 70 F, our heating requirement for this case becomes nil. Line D graphically represents this “ideal” situation because it depicts the varying heat loss from the building—as a function of outside ambient temperature —being balanced by a varying heat input controlled by our variable-vacuum-differential heating control.

The horizontal line representing the internal sensible heat gained from transmission, sunlight radiation, occupants, lights, electrical devices, process heat, and infiltration, is assumed for this case to be 3,000 EDR. It intersects Line D at Point F. If the heat gain from piping (assumed to be 2,250 EDR) is disregarded, this point represents the outside-air temperature at which the heating system will be shut down. If the heat gained through the piping is considered, that line intersects Line D at Point E and corresponds to the outside-air temperature at which heating will be required for those areas not affected by the piping heat.

The steam temperature capability of the example system, with a rising outdoor-air temperature, is represented by Line A. Note that the steam temperature increases as the outside ambient temperature increases. The on-off cycling of the boiler, being controlled by a pressure switch, is clearly shown by the variation in terminal-unit temperatures. If this switch was set to fire at 2 psig and shut down the burner at 5 psig, the cycle is obviously less frequent as the outside temperature becomes milder because there is less total steam required by the system. Because the steam rate is reduced, basic friction-loss calculations for flowing fluids tell us that the supply-line pressure drop will also be reduced. This results in the highest-pressure steam and, therefore, the highest-temperature steam at the terminal units on the mildest days. If Line B, which represents the average room temperature for our system with pressure-switch control only, is compared to our “ideal” Line D, the amount of overheating to be expected is clearly evident.

If individual room control valves are incorporated in the system, the peaks are obviously reduced. The point on the graph labeled 78 corresponds to an outside temperature of 35 F and represents actual field data collected for an office building utilizing the system being illustrated herein, with individual room control by means of thermostats and pneumatically operated control valves. At 35-F outside temperature, the average room temperature was 78 F with the thermostat set for 70 F. Based on this 8 F of overheating, with the control valves cycling, Line C represents the average room temperature with this system. Clearly, the same phenomena seen in the system with only pressure-switch control is exhibited here, but to a lesser degree. As the flow to the control valve decreases, with a reduced demand for steam, the pressure drop through the valve decreases and the pressure at the inlet of the valve gradually increases, reaching system pressure when the valve is fully closed. When the valve opens, higher-temperature steam is initially admitted to the terminal unit, resulting in hotter steam on milder days. And because the terminal units are receiving hotter steam to heat warmer air, the cooling-off (heat decay) rate is rela-

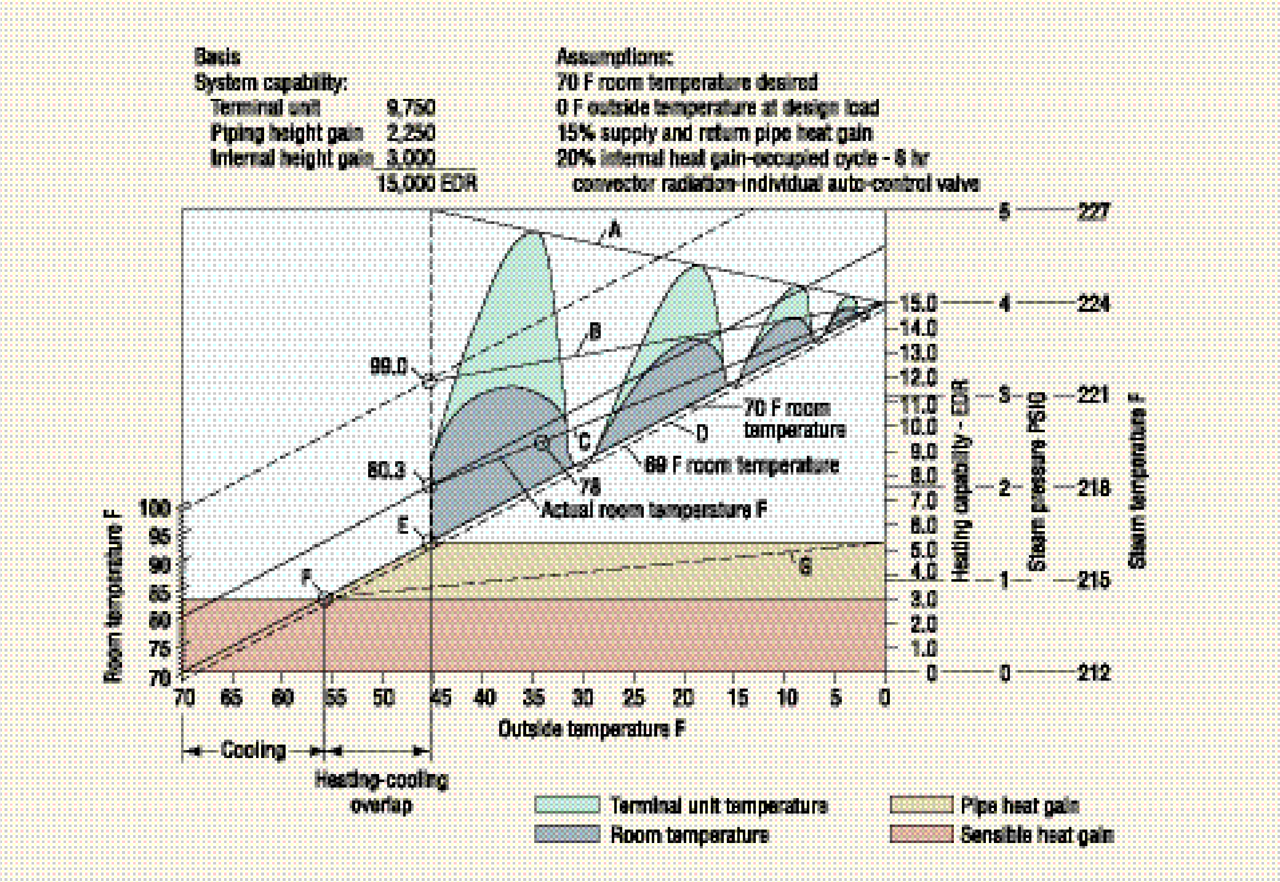

Figure 1. ***Variable-vacuum-differential control system applied to a two-pipe system with condensate return at atmospheric pressure.***

tively long.

Comparing lines B and C shows the marked improvement brought about by adding individual room controls. And, although in our example the valves used were air-operated, lower-cost “self-contained” valves are just as effective as the more expensive air- or electricity-operated valves. Because self-contained thermostatic valves have an inherently high valve flow coefficient (Cv), +/- 2 F control sensitivity is readily achievable.

It should be noted, however, that this pressure drop becomes very significant when these valves are incorporated into a variable-vacuum-differential heating-control system because one of the advantages of such a system is its low overall pressure drop. Because variable-vacuum- differential heating controls the temperature of the steam in the supply lines, as well as in the terminal units and supply lines, the individual regulating valves at the radiators must be calibrated for accurate proportioning of steam flow. Line G illustrates the performance characteristics of a variable-vacuum-differential heating control on our example system. Because neither of the other described control schemes—pressure switch only or pressure switch and room-control valves—incorporated a vacuum pump, the atmospheric pressure at the returns limit the temperature of the steam to no less than 212 F. Without vacuum control, the heat input to the building is essentially uncontrolled.

System Requirements

So what are the required elements of a variable-vacuum-differential heating-control system? Obviously, the control components themselves are essential, as are the calibrated proportioning valves and zoned steam-supply-line control valves previously mentioned. And, just as obviously, there must be a vacuum pump to sustain the required vacuum. Because the vacuum pump exhausts air (and other non-condensables) from the system, it also promotes steam circulation and minimizes warm-up time. There must also be devices for measuring/balancing the rate of heat flow and for sensing indoor and outdoor temperatures. Finally, the steam traps in the system must be suitable for operation under deep vacuum and the traps (and piping) must not leak. Thermostatic traps utilizing bimetal elements are clearly not suitable in this application because they sense only temperature without regard to the system operating pressure. Although there are a number of steam traps employing fluid-filled elements and advertised as being suitable for operation to 25 in. Hg

Outdoor temp.(F)	Value position (%)	System pressure	Climate data (%)
70	0		
65	0	N/A	32.2
60	0		
55	0	25.0 in. Hg	
50	9	25.0 in. Hg	17.7
45	18	25.0 in. Hg	
40	27	25.0 in. Hg	20.12
35	36	22.0 in. Hg	
30	45	18.0 in. Hg	20.2
25	54	14.3 in. Hg	
20	63	10.6 in. Hg	6.8
15	72	7.0 in. Hg	
10	81	3.3 in. Hg	2.4
5	90	0.2 psig	
0	100	2.0 psig	0.5
			100%

***TABLE 2.* Typical calibration of variable vacuum differential heating control.**

vac, if they are not filled and sealed under a vacuum, they may fail to perform satisfactorily in a differential vacuum condition. Steam traps operating under differential vacuum must have the ability to operate consistently at a fixed (subcool) temperature below the saturation temperature of the steam, regardless of the system pressure. Keep in mind that the steam traps in conventional vacuum return systems are seeing vacuum only on one side.

In addition to the obvious economic benefits that this type of retrofit can provide to the building owner (and management/maintenance personnel), there are a couple of important, intangible benefits for the building's occupants: even heat distribution, with room temperatures generally fluctuating less than +/- 1 F and quiet operation because the noise often associated with system warm-up is eliminated.

As previously mentioned, the piping and components must not be leaking excessively because they should be capable of holding a relatively deep vacuum. In other words, the steam traps must be working properly, and the volume of air inleakage through pipe joints, couplings, valves, etc. must not be greater than the air-handling capacity of the vacuum pump. Ideally, the system will be tight enough to hold a vacuum of 25 in. Hg, the operating condition at which the greatest potential savings can be realized. However, even in a system capable of reaching only 18 in. Hg vac, the savings can still be in double digits. For example, for the typical case of 40-percent annual fuel savings with this system compared with a two-pipe atmospheric condensate return with boiler-only pressure control, we will calibrate the variable-vacuum-differential controls according to Table 2. This is based on "ideal" assumptions for actual climatic conditions in a Midwest city (percent shown in table is the percentage of total heating hours per year per outdoor temperature range), noting that these are only approximations because no two systems will be identical.

As can be seen in the table, approximately 30 percent of the control hours required are in the range of 0 to 30 F, where the system is operating at or above 18 in. Hg. Therefore, in this instance, the savings is approximately 30 percent of the 40-percent potential, or a 12-percent total annual fuel savings from a "leaky" system. There will also be additional savings in the 30 to 55 F range because the air inleakage is not 100 percent of the volume, and there is steam entering the piping. Although even in a system with moderate air leaks, some fuel savings can be achieved, it is still important to repair the leaks. Operating under conditions where the vacuum pump must run just to handle the additional air volume, is certainly not desirable, and having high oxygen content in the system is, of course, potentially damaging to all of the ferrous components.

Conclusion

Prior to making a decision on converting a conventional steam heating system to variable-vacuum differential control, a complete survey of the building should be accomplished by qualified personnel working with an individual who has knowledge of the building's piping arrangement, its past remodeling or renovation projects, current heating complaints (or lack thereof), and the intent of the existing controls.

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