

The following article was published in ASHRAE Journal, January 1998. © Copyright 1998 American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. It is presented for educational purposes only. This article may not be copied and/or distributed electronically or in paper form without permission of ASHRAE.

Equipment Oversizing Issues With Hydronic Heating Systems

By **i. B. Kilkis, Ph.D.**
Member ASHRAE

An important development in the late 19th century was the widespread use of central heating systems. In 1854, Joseph Nason and Jones Walworth filed a patent called “Improvement in warming houses by steam.” At first, low pressure steam, single pipe-cast iron radiator systems dominated the heating industry. By the turn of the century, however, many types of radiators were being manufactured (Holohan, D., 1992). Since then, space heating systems coupled to primary energy resources have improved little in terms of exergetic efficiency.

In its simplest and practical way of definition, exergetic efficiency is a measure of how effectively we utilize the available heat derived from an energy resource at a certain temperature. There is a large temperature difference between the primary energy resource (i.e. flame temperature of a natural gas burner) and the space heating system. The result is a very low exergetic efficiency, as shown in *Figure 1* (Rosen and Dincer, 1996).

Today, as a response to the high cost of primary energy resources and environmental concerns, low-temperature district and central heating systems have been developed which permit the use of formerly unclaimed energy resources. In a low-temperature district heating system the fluid supply temperature is less than 120°F (49°C) (ASHRAE, 1991). As a result, these systems have higher exergetic efficiencies, relying on temperature differences as low as 20°F (11°C) between the heat source and the space heating equipment (such as a low enthalpy geothermal district energy system) (Kilkis, i. B., 1996). The standard condition for most of the hydronic heating equipment still remains at an effective temperature of 215°F (102°C)—inherited from practices based on steam.

Therefore, with some exceptions such as hydronic radiant panels, heating equipment needs to be oversized when coupled or retrofitted to low temperature energy resources. In this case a careful optimization between temperature peaking and equipment oversizing becomes a necessity, as does more detailed temperature drop and energy loss calculations in the hydronic circuitry.

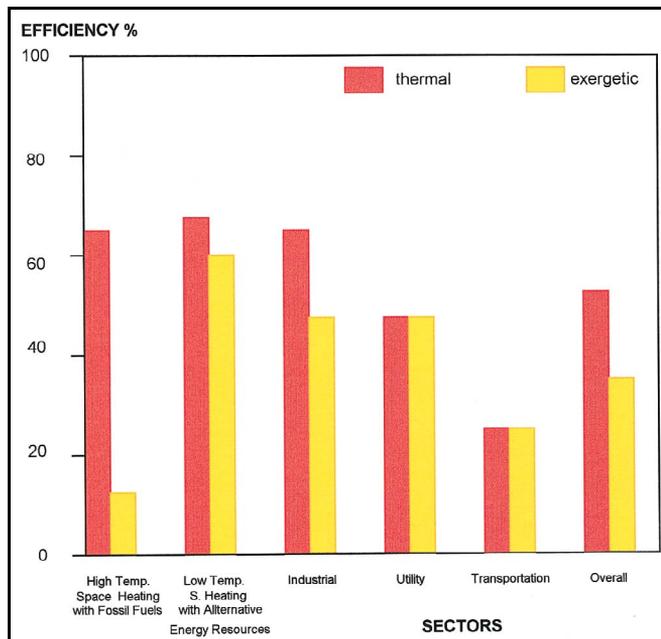


Fig. 1: Thermal and exergetic efficiencies in typical sectors.

Fundamentals

Figure 2 and *Equation 1* show the relationship between the heat output of heating equipment and the effective fluid temperature in the equipment.

$$q = c(t_f - t_a)^n \tag{1}$$

q is the actual heat output of the equipment in Btu/h, t_f is the effective fluid temperature in °F and t_a is the temperature of the air contacting the heat transfer surface of the equipment.

Usually, c and n are determined experimentally. Coefficient c is the specific heat output of the equipment, at a unit temperature difference between the fluid and the air. In certain equipments, c' may define the specific heat output for a characteristic portion of the unit. For example, c' applies to a single section of a cast iron radiator, or a unit panel surface area of a floor heating system, such that:

$$q = [c'N(t_f - t_a)^n] \tag{2a}$$

[cast iron radiator] or;

$$q = c'A_p(t_f - t_a)^n$$

[radiant panel]

N is the number of sections in a cast iron radiator unit, and A_p is the area of the panel surface. *Equation 2a* becomes identical to *Equation 1* when N and A_p are known. Then:

About the Author

i. B. Kilkis, Ph.D., is director of the Research and Development Department at Heatway Radiant Floors and Snowmelting in Springfield, Mo. He has a doctorate in mechanical engineering from Middle East Technical University, Turkey. He is a member of three ASHRAE technical committees and several standards project committees.

$$c = c'N \text{ or } c = c'A_p \quad (2b)$$

Power n is independent of the system of units used, and lies between 1.0 and 1.5. For a finned-tube unit, n varies with air and fluid temperatures. It is customary to take it as 1.25. The relationship is almost linear for radiant panels: power n is 1.1 for floor heating and 1.0 for ceiling heating. To give a brief order of magnitude, c' is about 0.45 Btu/h-ft²·°F for a thin-slab floor heating panel with a total thermal resistance of about 1 ft²·h·°F/Btu (non-metallic tubes laid at 9 in. on centers, 2 in. concrete over pour with light carpet).

For an aluminum, suspended ceiling heating panel with attached metal tubing at 6 in. on centers, c' is 0.89 Btu/h-ft²·°F. By manipulating Equation 1, heating capacity loss at effective fluid temperatures below 215°F (102°C) can be determined. For example, a cast-iron radiator group in a 70°F (21°C) room will deliver only 25% of its rated output when the effective fluid temperature t_f is 120°F (49°C). As a consequence, the equipment has to be oversized by a factor of four, in order to restore the full heating capacity. t^* is the effective temperature for full heat output capacity of the original equipment size:

$$\begin{aligned} \text{Oversize Factor, OF} &= \left[\frac{(t^* - t_a)}{(t_f - t_a)} \right]^n \\ &= \frac{(215^\circ\text{F} - 70^\circ\text{F})^{1.3}}{(120^\circ\text{F} - 70^\circ\text{F})^{1.3}} = 4 \end{aligned} \quad (3)$$

Figure 2 indicates that a linear control algorithm for part load management of a great deal of heating equipment may not be sufficient.

Much literature indicates that the effective fluid (water) temperature may be expressed in terms of the arithmetic average (mean) of the supply (inlet) and return (exit) temperatures (ASHRAE, 1996) if the temperature drop is not above 30 to 40°F (17° to 22°C).

$$\begin{aligned} q_w &= \rho_w c_p Q_w (t_s - t_r) = q = c(t_f - t_a)^n \\ &= c[(t_s + t_r)/2 - t_a]^n \end{aligned} \quad (4)$$

In the hydronic loop of the equipment, q_w [Btu/h] represents the rate of heat delivered by the fluid with density ρ_w [lb/ft³] and specific heat c_p [Btu/lb·°F], and flowing at a volume rate Q_w [gallons per minute]. Temperatures t_s and t_r [°F] are the supply and return fluid temperatures, respectively. For water at 180°F (82°C), q_w is equal to $[495 \cdot Q_w \cdot (t_s - t_r)]$. In low-temperature applications, the coefficient drops slightly. For example, it is 485 at 110°F (43°C). While the equipment performs according to the right-hand side of Equation 4, the hydronic loop experiences a temperature drop ΔT_w which is equal to $(t_s - t_r)$. For a required heat output and “preselected” ΔT_w , the only design unknown Q_w can be solved from Equation 4.

Yet, this equation is unable to solve a design problem which involves more than one unknown.

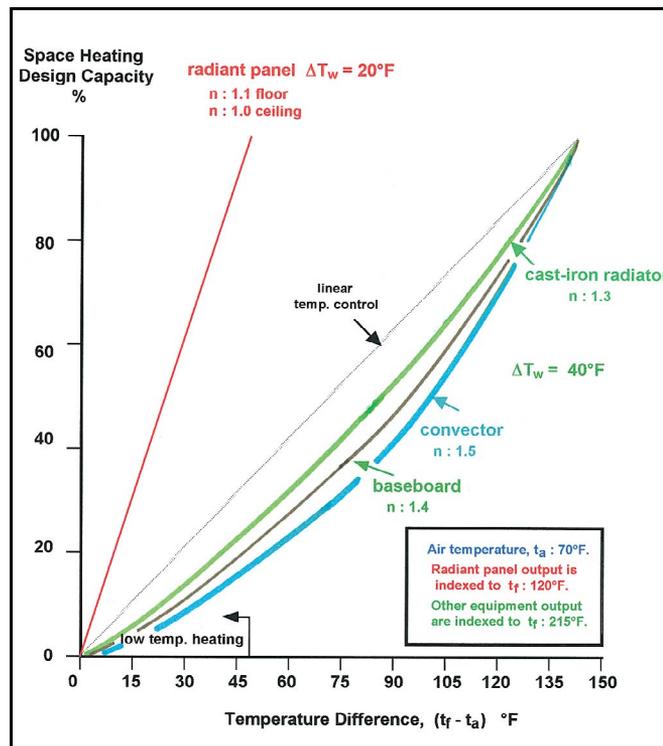


Fig. 2: Comparison of the space heating capacity of different equipment at reduced fluid temperatures.

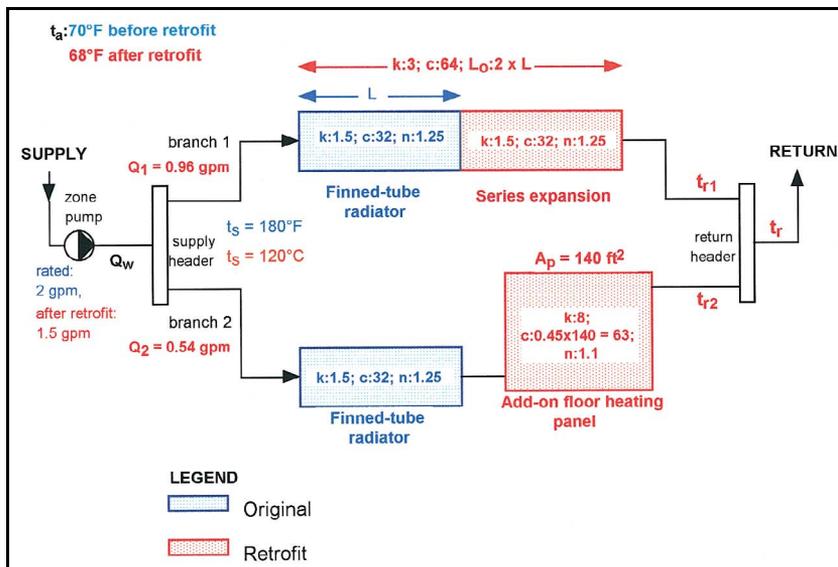


Fig. 3: Zone retrofit for low temperature hydronic heating.

This is typical in hybrid HVAC systems and retrofit applications, especially when parallel branches and a diversity of equipment with different characteristics are involved in a low-temperature heating system. Except for some sophisticated computer codes (Amistadi, H., 1994), analytical design tools with a simplicity comparable to Equation 4 are few (Siegenthaler, J., 1995).

Figure 3 shows a simple retrofit case. The original hydronic heating system serving a small zone, consisted of two identical

finned-tube units, in parallel. Design heating load is 20,000 Btu/h (5862 W), at an indoor design air temperature of 70°F (21°C). A zone pump delivers 2 gpm (0.13 L/s) to the supply header, for a design ΔT_w of 20°F (11°C). Each piece of finned-tube equipment is rated at 10,000 Btu/h (2931 W) at a 180°F (82°C) supply water temperature. Other design parameters are shown in Figure 3.

In this case, Equation 4 is sufficient because the volume flow in each branch may be assumed to be identical. If the units had different flow resistances, normal practice would size the pump according to the branch with the highest flow resistance and add a flow balancing valve to the other branch. This is an oversimplification of the design and its price is usually an oversized zone pump. Now this system is going to be retrofitted with a district heating system which can only deliver water at a supply temperature of 120°F (49°C). In order to restore the heat supply to the zone, one of the finned-tube units will be doubled (expanded in series). On the existing floor, a thin-slab radiant panel which obeys Equation 2, with $A_p = 140 \text{ ft}^2$ (13 m²) will be tailed-in to the second finned-tube unit (add-on in series). The original zone pump will be retained. Because of the special attributes of radiant panel heating, the indoor design temperature will be 68°F (20°C) instead of 70°F (21°C) (Kilkis, i. B., 1993-a). The corresponding heating load will be 19,000 Btu/h (5569 W). In order to predict the performance of this retrofit, we need to calculate the following:

- a. Actual flow rate and the energy (head) loss in each branch with the existing pump;
- b. Temperature drop and the effective fluid temperature in each unit;
- c. Heat output of each unit; and
- d. Fluid temperature at the return header. This information is required for the calculations regarding the district heating loop.

Equation 4 is now insufficient, because the volume flow rate temperature drop and the energy loss in each branch has to be simultaneously calculated.

Volume Flow and Energy (Head) Loss in Parallel Circuits

Pressure loss is the correct term for steam systems and open hydronic systems. In closed hydronic systems, the proper term is energy loss due to pipe friction and singular flow resistance. This term also eliminates the problem of applying the correct density, which varies with the fluid temperature in the circuit (Hansen, E. G., 1985). Figure 4 shows the generalized model which is used to derive the volume flow and energy loss equations. There are (m) parallel branches, tiered to the same supply and return headers. Every branch may have (p) different units in series.

Then the energy loss Δh_j in terms of feet of head across a branch (j), with a volume flow Q_j , will be:

$$\begin{aligned} \Delta h_j &= \sum_{i=1}^p (k_i Q_j^r + K_i Q_j^r) = \sum_{i=1}^p (k_i + K_i) Q_j^r \\ &= \sum_{i=1}^p k_i Q_j^r \end{aligned} \tag{5}$$

Here, k_i' is the pipe loss coefficient and K_i is the singular loss coefficient, each indexed to the volume flow rate, and attributed to the unit (i) on branch (j). k_i is the total loss coef-

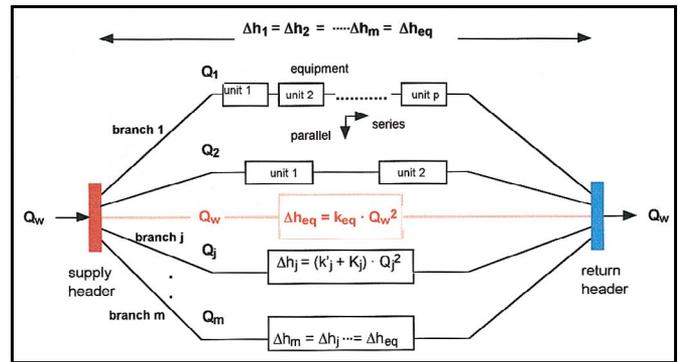


Fig. 4: Energy loss in parallel circuits with unequal flow resistance (r:2).

ficient for a given piece of equipment. In Equation 5, r varies with the degree of dependence of the pipe friction on the Reynolds number. If the pipe friction factor is independent of the Reynolds number (fully rough flow), then r is equal to 2. In radiant panel circuits, the pipe friction factor is generally Reynolds number dependent, and r is less than 2; approximately 1.75 (Siegenthaler, J., 1995). However, its effect on the following analysis is negligible. Then, with the pressure continuity at the supply and return headers (flow nodes) will be:

$$\begin{aligned} \Delta h_1 &= \Delta h_2 = \Delta h_j = \dots = \Delta h_m = \Delta h_{eq} \\ &= k_{eq} Q_w^2 \leq \text{Pump head (r:2)} \end{aligned} \tag{6}$$

k_{eq} is the equivalent loss coefficient for a single imaginary unit, on a single imaginary branch, which delivers the total volume flow Q_w across the headers with an equal energy loss of real branches. This term replaces all the units and branches.

By using the flow continuity:

$$Q_w = \sum_{j=1}^m Q_j \tag{7}$$

and Equation 6 (Hansen E.G., 1985):

$$k_{eq} = \frac{1}{\left(\frac{1}{\sqrt{k_1}} + \frac{1}{\sqrt{k_2}} + \dots + \frac{1}{\sqrt{k_m}} \right)^2} \tag{8}$$

Then the volume flow rate in each branch will be:

$$Q_j = (k_{eq}/k_j)^{1/2} Q_w \quad j:1 \text{ to } m \tag{9}$$

It must be noted that k_{eq} and L_{eq} reduce to k_1 and L_1 respectively, if there is only one branch and one unit in the system.

If the entrance and exit losses are small when compared to the pipe losses, as in radiant panels, then k can be expressed in terms of the characteristic length L . If this is valid for all units and branches in the hydronic circuit, then k may be replaced with L in Equations 8 and 9 (Kilkis, i. B. 1993-b). The characteristic length L of an equipment may be either the total length of the serpentine pipe within the equipment, the length of pipes clustered in parallel or the length of a section such as the height of a cast-iron radiator. In a retrofit, the number of pieces of

equipment or equipment sections in the hydronic circuit increases. Retrofit may be either in parallel, series or both.

For example, adding radiator sections to an existing radiator unit is a parallel retrofit. Similarly, introducing new equipment to the hydronic circuit in parallel is a parallel addition. Figure 5 shows typical parallel and series equipment retrofit and additional arrangements for space heating equipment. Depending on the arrangement, either the characteristic length changes to L_o , or the number of equipment or equipment sections change to N_o , or both change.

In this example, there are two parallel branches and there will be three pieces of equipment after the retrofit. Therefore; (m) is 2, (p) is 1 in branch one, and (p) is 2 in branch two. Using the data shown in Figure 3, after the retrofit:

$k_1: 2 \times 1.5$ ft of head/gpm² (twice the original finned-tube length in series); and

$k_2: (1.5 + 8)$ ft of head/gpm² (finned-tube unit of original size plus tailed-in floor panel).

From Equation 8;

$$k_{eq} = \frac{1}{\left(\frac{1}{\sqrt{3}} + \frac{1}{\sqrt{9.5}}\right)^2} =$$

$$1.23 \text{ ft of head/gpm}^2$$

By referring to a specific pump curve from the manufacturer, the supply volume flow rate Q_w can be predicted after the retrofit. In this example, it is 1.5 gpm (.10 L/s). The corresponding loss is 2.77 ft of head (1.23 x 1.5²).

By using Equation 9 and k_{eq} :

$$Q_1 = (1.23/3)^{1/2} \times 1.5 = 0.96 \text{ gpm};$$

$$Q_2 = (1.23/9.5)^{1/2} \times 1.5 = 0.54 \text{ gpm}$$

Note that Q_1 and Q_2 add up to Q_{wo} . Without flow balancing valves, the next step will be to predict the temperature drop in each piece of equipment.

Temperature Drop: Exact Equations

Provided that the volume flow rate is precisely determined and the supply fluid temperature is known, Equation 4 may be iteratively solved by assuming a linear temperature drop. This will determine the heat output and the fluid temperature drop regarding a given piece of equipment. When the characteristic length becomes too long—especially in a series retrofit—or many pieces of equipment are tailed-in in a single pipe system, the nonlinearity in ΔT_w will be cumulative and therefore cannot be ignored. In order to exactly solve the non-linear temperature profile $t(x)$ and the temperature drop in one step, the following equations were derived (Kilkis, i. B., 1993-b):

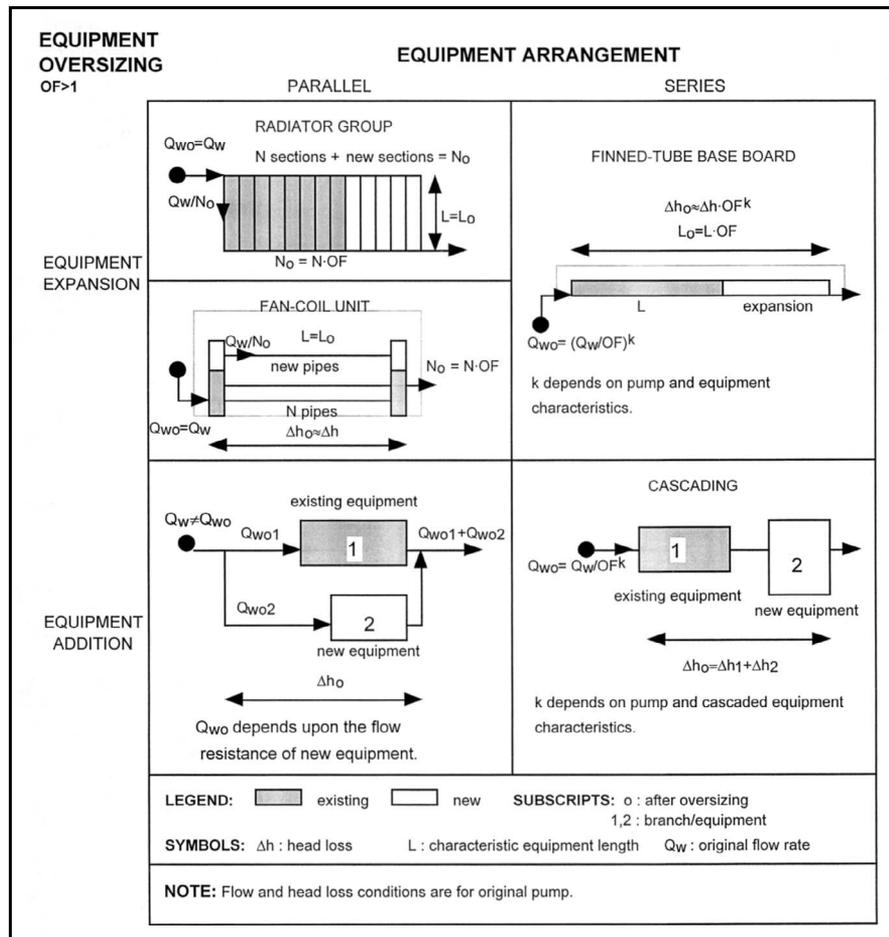


Fig. 5: Series and parallel equipment oversizing arrangements.

$$t(x) = e^Y + t_a \quad n \neq 1 \quad (10)$$

Y depends upon the equipment (c, n, L), the volume flow rate Q_i through the equipment (i), the supply fluid temperature (t_s) and the air temperature (t_a):

$$Y = \frac{\ln \left[(t_s - t_a)^{1-n} + \frac{(n-1)c}{485 Q_i} \cdot (x/L) \right]}{1-n} \quad n \neq 1 \quad (11)$$

x is the distance from the inlet, measured on the characteristic length L . If there are parallel pipes in the unit, Q_i must correspond to the volume flow in a single pipe (See Figure 5). The return (exit) temperature t_r is determined with the condition $(x/L) = 1$. If n is equal to 1, as in radiant ceiling heating, Equations 10 and 11 may not be used. Equation 12 was derived in order to handle this condition.

$$\frac{t(x) - t_a}{t_s - t_a} = e^{-\left[\frac{c}{485 Q_i} \cdot \left(\frac{x}{L} \right) \right]} \quad n = 1 \text{ (ceiling panel)} \quad (12)$$

These equations assume that effective water temperature is around 110°F (43°C). For temperatures other than 110°F (43°C), the coefficient (485) may be revised by using the

expression ($\rho_w \cdot 8.02$), where ρ_w is the fluid density at the given temperature.

The new algorithm will be applied now to the retrofit example as shown in *Figure 3*.

Branch 1:

This branch serves the finned tube radiator, which is oversized by a factor of two. Therefore, the characteristic length after the retrofit, L_o is $2 \times L$, and the heat output coefficient is $2 \times c$. It is a common practice to take the temperature of air entering a finned tube radiator as 5°F (3°C) below the design indoor air temperature. However, the actual difference between the two temperatures depends upon the existing indoor conditions and the radiant, convective heat output split. For a conservative design, t_a may be equated to the indoor air temperature.

Using *Equations 10* and *11*, the return (exit) water temperature is calculated as 104.5°F (40°C). Therefore, the temperature drop at retrofit conditions will be 15.5°F (9°C). By using the actual flow rate and the actual ΔT_w , the exact heat output from the left-hand side of *Equation 4* is calculated as 7,217 Btu/h (2115 W). These calculations indicate that equipment oversized by increasing L has diminishing returns on the heat output, especially when the circuitry pumping is not modified accordingly. Therefore, instead of an excess oversized in series, the fluid temperature may be peaked and/or the retrofit be performed in parallel circuits. *Figure 6* shows the temperature profile of the fluid flow in the oversized finned-tube radiator. The effective fluid temperature t_f is calculated from the right-hand side of *Equation 4*; It is 111.8°F (44°C). The non-dimensional area above the curve (A) in *Figure 6* also approximates t_f :

$$t_r/t_s \approx 1 - 2A [0 \leq A \leq 0.5] \quad (13)$$

Branch 2:

This branch serves the original finned-tube radiator and the new radiant floor panel. Using the same algorithm, their exit temperatures are 105.9°F (41°C) and 95°F (35°C), respectively. The exact heat output of the two pieces of equipment is 3,693 Btu/h (1082 W) and 2,855 Btu/h (837 W). The total heat output of this branch will be 6,548 Btu/h (1919 W). The return temperature of branch 2 is the exit temperature of the radiant panel, which is 95°F (35°C).

The mixed water temperature at the return header will be a flow averaged value between branches 1 and 2:

$$t_r = \frac{Q_1 t_{r1} + Q_2 t_{r2}}{Q_w} = \frac{0.96 \times 104.5 + 0.54 \times 95.0}{1.5} = 101^\circ\text{F} \quad (14)$$

The total heat output after the retrofit will be 13,765 Btu/h (4035 W). This capacity corresponds to 72% (13,765/19,000) of the design heat load, which is a typical base load figure for the heating season. In this example of a typical geothermal district heating retrofit, oversized will be limited by the base load, and temperature peaking at a back-up (peaking) plant will complement the system for peak loads. The same algorithm reveals that during the peak load the supply water temperature needs to be boosted from 120°F (49°C) to about 140°F (60°C) (Kilkis i.B. and Eltez, M., 1996).

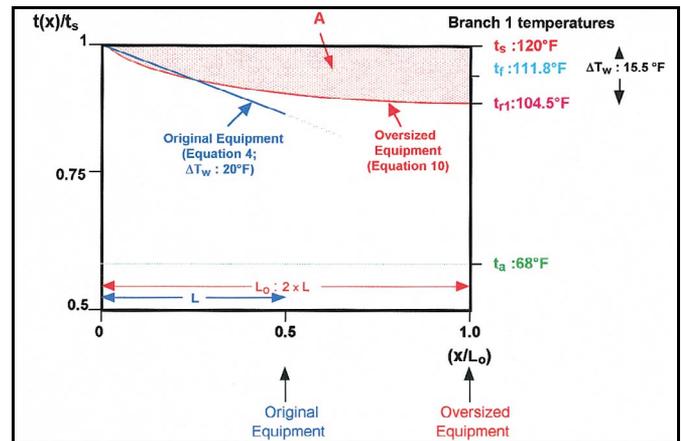


Fig. 6: Temperature profile of the flow in the series oversized equipment (without circuitry oversized).

This example indicates that the optimum split between oversized and temperature peaking options in a typical space heating application with low temperature energy resources can be determined by a careful manipulation of this algorithm.

Parallel Versus Series Circuits

Early steam heating systems were usually operated on a double-pipe, gravity return, parallel flow configuration (Holoan, D., 1992). This custom continued in conventional hydronic heating systems with direct- and reverse-return two pipe systems (Chapter 12, *1996 ASHRAE Handbook—HVAC Systems and Equipment*), which tend to provide a more consistent supply temperature to each unit. Single pipe series circuits became popular mainly due to the advantage of lower piping costs. Yet this advantage may diminish with higher pumping requirements. In a simple series circuit configuration, design calculations are easy because the water flow rate is the same, and the loss and temperature drop are additive.

In a diverting series system, the designer may enjoy a similar simplicity by prescribing the design flow rate in each unit. It is this simplicity, rather than technical and economical difficulties, which tempted the engineers to avoid parallel circuits. The progress of radiant panel heating systems which are arranged in parallel circuits and the growing necessity of employing hybrid space HVAC equipment in certain applications (Kilkis et al, 1995) are two of the many reasons why parallel circuits—usually with unequal flow resistance—are recently becoming an important design issue.

The presumption that the branch with the highest flow resistance will suffer substantial capacity loss because of low volumetric flow is in fact questionable. Flow (pressure) balancing usually results in pump oversizing. One must acknowledge that there are certain factors which need to be closely monitored in designing a parallel system. These are:

- a. Flow velocity should not be less than 1 ft/s in any branch, and the flow should be turbulent;
- b. Amount of capacity loss in the circuit with lowest fluid flow should be small; and
- c. There should be no short circuiting. Short circuiting is a matter of header design, and a suitable design can avoid it (Hansen, E. G., 1985).

Example:

Three radiant floor panel circuits with circuit lengths equal to 100 ft (30 m), 150 ft (46 m), and 200 ft (61 m) are connected to the same supply and return headers. Their respective panel surface areas are 70 ft²(7 m²), 105 ft²(10 m²), and 140 ft² (13 m²) at a 0.5 in. (13 mm) I.D. tube spacing of 9 in. (229 mm) on centers. Panels obey Equation 2a, and *c* equals 31.5, 47.3 and 63 Btu/h·°F, respectively. The supply volume flow rate is 3 gpm (0.2 L/s) at 120°F (49°C). Indoor design air temperature in the zone is 68°F (20°C). Determine the actual flow rate, the temperature drop and the heat output for each circuit. In this example, *L* replaces *k* (Kilkis, 1993-b):

$$L_{eq} = \frac{1}{\left(\frac{1}{\sqrt{100}} + \frac{1}{\sqrt{150}} + \frac{1}{\sqrt{200}}\right)^2}$$
$$= 15.7 \text{ ft}$$

Using Equation 9:

$$Q_1 = (15.7/100)^{1/2} \cdot 3 = 1.188 \text{ gpm}$$

In a similar fashion:

$$Q_2 = 0.970 \text{ gpm}; Q_3 = 0.842 \text{ gpm}$$

[Note that they add up to 3 gpm].

Using Equations 10 and 11:

$$\Delta T_1 = 4^\circ F; \Delta T_2 = 7.1^\circ F; \Delta T_3 = 10.5^\circ F$$

Consequently, from the left-hand side of Equation 4,

$$q_1 = 2, 328 \text{ Btu/h}; q_2 = 3, 340 \text{ Btu/h}; q_3 = 4, 303 \text{ Btu/h}$$

Total heat output in the zone will be 9,971 Btu/h (2922 W).

In a traditional attempt, one would first assume balanced flow conditions, and then by using both sides of Equation 4 would solve *t_r*, and determine the heat output of each circuit. If the zone pump would be selected according to the longest run at balanced flow conditions, the required pump head would be overestimated by about 42% when compared with the unbalanced case. If the flow in each circuit were adjusted identically to 1 gpm, the heat output in each circuit would be 2,303, 3,374 and 4,383 Btu/h, (675, 989, 1284 W) respectively. In this case, the total heat output will be 10,060 Btu/h (2949 W). These figures indicate that without any balancing the maximum drift in the total heat output at design conditions will be only 89 Btu/h (26 W).

In the longest radiant panel circuit, the drift will be less than -2%. This rather unexpected result shows that there is no need to add balancing valves for the sole purpose of resuming the design heat output. Without flow balancing, the flow velocity in the longest circuit with 0.5 in. (13 mm) I.D. tubing will be about 1.4 ft/s (0.43 m/s). The corresponding Reynolds number is about 9,000 at an effective water temperature of 114.5°F (46°C).

Discussion and Conclusions

Analytical equations presented in this article provide a complete package for the design and analysis of parallel and mixed type circuits, serving any kind of equipment. This algorithm is applicable to any degree of circuit complexity and equipment diversity, as long as the equipment performance may be expressed in terms of Equations 1 and 5. This algorithm also

provides a precise check for the flow velocity and helps to fine tune other equipment and circulators.

Acknowledgment

The information presented herein is part of the research project funded by the Turkish Scientific and Technical Research Council (TÜBİTAK). This international support is greatly appreciated.

References

- Amistadi, H. Selecting piping system software. *Engineered Systems*. June, 1994, pp. 57- 62.
- ASHRAE. 1991. *Terminology of HVAC&R*. Atlanta, Georgia: ASHRAE.
- ASHRAE. 1996. *ASHRAE Handbook — HVAC Systems and Equipment*. Atlanta, Georgia: ASHRAE.
- Hansen, E. G. 1985. *Hydronic system design and operation*. New York: McGraw-Hill.
- Holohan, D. 1992. *The lost art of steam heating*. Bethpage, New York: Dan Holohan Associates.
- Kilkis, i. B. 1993-a. "Advantages of combining heat pumps with radiant panel heating and cooling systems," Newsletter. Sittard: IEA Heat Pump Centre, Vol. 11, No. 4, pp. 28-31: The Netherlands.
- Kilkis, i. B. 1993-b. "Computer-aided design and analysis of radiant floor heating systems." Proceedings, paper No. 80, Clima 2000 Conference, November 1-3: London, UK.
- Kilkis, i. B., S. R. Suntutur, and M. Sapci. 1995. "New trends and alternatives in HVAC systems, a case study about museum and library environments." *ASHRAE Journal*, Vol. 37, No. 12, p. 23-28. Atlanta, Georgia: ASHRAE.
- Eltez, M., and Kilkis, i. B. 1996. "Low Enthalpy Geothermal District Heating System in Nevsehir, Turkiye." 31st Intersociety Energy Conversion Engineering Conference. IECEC 96. *Proceedings*. Vol. 3. p. 1642-1646.
- Kilkis, i. B. and Eltez, M. 1996. "Advances in geothermal energy use, hybrid cycle/integrated district HVAC systems." *ASHRAE Journal*, Vol. 38, No. 10, p. 40-48. Atlanta, Georgia: ASHRAE.
- Rosen, M. A. and Dincer, I. 1996. "Energy and Exergy Analyses of Sectoral Energy Utilization: An Application for Turkey." Proceedings. First International Trabzon Energy and Environment Symposium. Trabzon. Vol. 3. p. 1059-1065.
- Siegenthaler, J. 1995. *Modern Hydronic Heating*. Boston: Delmar Publishers. ■

Please circle the appropriate number on the Reader Service Card at the back of the publication.

Extremely Helpful	450
Helpful	451
Somewhat Helpful	452
Not Helpful.....	453