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# Application of Component Models for Standards Development

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## ABSTRACT

*This paper describes a simulation model and life-cycle cost study used to develop a fan speed control requirement for cooling towers, evaporative condensers, and air-cooled condensers as part of* ASHRAE/IESNA Standard 90.1-1999, Energy Standard for Buildings Except Low-Rise Residential Buildings. *A joint working group consisting of members from ASHRAE TC 8.6 and ASHRAE/IESNA SSPC 90.1 undertook this study, employing a national energy model developed by a national laboratory and a computer model of a central chilled water plant. The results from the study supported development of a new requirement for fan speed control on all fans over 5 hp (3.7 kW). The SSPC90.1 subsequently expanded the model to evaluate the impacts of controls on multiple cell towers and condensers with multiple fans. These requirements were adopted as part of the published standard in 1999.*

## OBJECTIVE

In 1995, a joint working group of ASHRAE/IESNA Standing Standards Project Committee 90.1 (SSPC90.1), Energy Standard for Buildings Except Low-Rise Residential Buildings, and ASHRAE Technical Committee 8.6 (TC 8.6), Cooling Towers and Evaporative Condensers, collaborated on a study to determine the life-cycle cost justified cooling tower efficiency and fan speed controls. The purpose of this working group was the development of new requirements for ASHRAE/IESNA Standard 90.1-1999 (Standard 90.1). This paper describes the fan-speed control portion of this study and the application of component chiller and tower models to achieve the working group's objectives.

## BACKGROUND

At the time of this study, Standard 90.1-1989 was in major revision: the scope had been expanded to include modifications and additions to existing buildings, the standard was rewritten in code language, new economic criteria were developed and applied to existing requirements, and new requirements were being added. This revision of Standard 90.1 spanned ten years. This study documents the development of one of the requirements in the new standard.

The making of standards, like the proverbial sausage, is not pretty. Requirements in Standard 90.1 were created through life-cycle cost analysis, consensus of experts, or, in many cases, a combination of the two. Through discussions at committee meetings and through the public review process, requirements are revised to account for issues of market equity, applications overlooked by the drafters, and often an attempt to appease competing agendas from different segments of the industry. The two-speed fan requirement described in this study is an example of a technically based requirement that was developed with industry input. Its collaborative roots were a key to its success.

## METHOD

Following the method established by the SSPC90.1, the working group developed a life-cycle cost analysis for the fan-speed control requirement. The analysis had three major sections: a simulation model to determine the energy usage and relative energy costs, estimation of the installed cost premium for alternative fan-speed controls, and a life-cycle cost model to evaluate the results. Each of these elements is described in detail below.

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## Hourly Simulation Model

The working group employed an hourly simulation model as the basis of the energy calculations. The group performed this work in two phases—one for thermal cooling loads and a second for the central cooling plant. The thermal cooling load model had been previously developed by the staff of a national laboratory to support the analysis of heating and cooling equipment efficiencies. The central plant model was custom built by the working group for this study. The challenge was to create models that broadly represent the entire range of non-residential buildings that have central cooling plants in all the climates of the United States.

## Thermal Cooling Load Model

Researchers at a national laboratory developed the thermal cooling load models from a series of 22 BLAST simulations. These models form a common basis for many of the requirements in Standard 90.1-1999: prior to their application in this study, the SSPC90.1 used the hourly loads from these simulations to determining life-cycle costs for efficiencies of mechanical equipment. The BLAST simulations represent two occupancies—an office building and a retail building—each of which was customized for 11 different climate locations in the United States. The model used a common three-story, 48,000 $ft^2$ (4,460 $m^2$) commercial building envelope and geometry for both occupancies; however, they modified the operation schedules, base building internal loads, and ventilation requirements as appropriate to represent each occupancy. The model employed a building shell, lighting, and HVAC elements that just met the draft prescriptive requirements of ASHRAE 90.1-1989R and ASHRAE 62-1989. Two mechanical system models were developed—a packaged single-zone system and a central VAV system. The working group selected the VAV system model as appropriate for this study.

The thermal cooling load model employed 11 cities and 2 occupancies to represent the bulk of construction activity in the United States. The researchers grouped similar climates by applying statistical clustering of available climate data from the National Oceanographic and Atmospheric Administration (NOAA)[1] as follows:

- Select climate parameters, such as heating degree-days, base 65°F (18.3°C), that are unique in their influence on energy use in commercial buildings.
- Identify groups of climate stations that are similar in these climate parameters.
- Define the geographic extent of each climate group.

This analysis yielded 16 unique climate regions, each represented by a specific climate station. They then considered new construction activity in each of these geographical regions. They used commercial building construction and population data from the U.S. Census[2] and aggregated the data for each climate-based geographical region. Based on this examination, they dropped five climate regions because of low population and/or little expected new construction activity. Construction volume data for these regions were assigned to the next most similar climate region. The final 11 representative climate stations, and the relative construction volume in both the retail and office building categories, are shown in Table 1. A complete description of the clustering process is reported in a paper presented at the CLIMA 2000 conference (Hadley and Jarnagin 1993).

**TABLE 1**
**BLAST Simulation Cities and Construction Activity Weighting Factors**

| City | Office | Retail |
|---|---|---|
| Denver, Colo. | 0.9% | 1.8% |
| Detroit, Mich. | 9.6% | 10.4% |
| Fresno, Calif. | 1.3% | 2.1% |
| Knoxville, Tenn. | 10.2% | 9.0% |
| Los Angeles, Calif. | 6.2% | 7.3% |
| Minneapolis, Minn. | 0.9% | 1.5% |
| Orlando, Fla. | 3.1% | 5.3% |
| Phoenix, Ariz. | 0.7% | 1.7% |
| Providence, R.I. | 5.3% | 6.8% |
| Seattle, Wash. | 2.2% | 1.7% |
| Shreveport, Mich. | 5.2% | 6.8% |
| **Total** | **45.6%** | **54.4%** |

## Central Plant Model Overview

The working group converted the results of the BLAST runs into text files for use as input to the central plant. Having obtained a model for the thermal cooling loads, the working group's next task was to define the attributes of the central plant model. This was a process based on the collective experience of the working group members, and decisions were achieved through discussion and consensus. The final plant model included a single centrifugal chiller with inlet-vanes, a single one-cell cooling tower, and the following three forms of cooling-tower fan control:

- Single-speed fan (on-off)
- Two-speed fan or pony motor sized for 2/3 capacity (100%-67%-off)
- Two-speed fan or pony motor sized for 1/2 capacity (100%-50%-off)

1. They considered the 212 locations in the continental United States for which Typical Meteorological Year (TMY) weather data files were available.

2. The source of these data was not recorded. To the best of our recollection, these were data available from the U.S. Census bureau on construction valuation that was available for commercial buildings up to 1993. It may have been based on building permits.

These cooling tower controls were selected as being representative of the bulk of commercial applications. Given the aggressive schedule for this working group (several months), the group reached a consensus to leave out variable-speed drive controls. They made this decision largely to avoid controversy in this new area of energy-standard requirements—at the time of the study, two-speed and pony motor controls were widely applied and accepted by the marketplace, but variable-speed fan controls were a relatively small part of the market. Variable-speed controls would not be required; however, they would be accepted as meeting or exceeding any requirement based on two-speed controls. They left the evaluation of variable-speed controls to subsequent study as an addendum to the standard at a later date.

The group developed the plant model in the macro language of a spreadsheet application. The assumptions used in this model were as follows:

- Size the chiller to just meet the peak cooling load.
- Base the chiller part-load performance on a centrifugal machine with inlet vanes.[3]
- Set the chilled water supply temperature to 44°F (6.7°C).
- Set the chiller full-load efficiency to 6.1 COP at ARI Standard 550-92 (ARI 1992).[4]
- Size the cooling tower to provide a 10°F (5.6°C) approach at the full heat rejection load, a 10°F (5.6°C) range, and the maximum wet-bulb temperature.
- Control the cooling tower to provide condenser water supply equal to the lesser of the maximum wet-bulb temperature in the climate file or the design approach temperature.
- Base the cooling tower available capacity on the default DOE2 (ver2.1E) curves.
- Set the cooling tower efficiency to 20 gpm/hp (1.7 L/s-kW) at CTI Standard 210-96 conditions (CTI 1996).[5]
- Set the 2/3-speed fan operation to 67% of the design capacity at 30% of the design power.[6]
- Set the 1/2-speed fan operation to 50% of the design capacity at 13% of the design power.
- Set the no-fan (induced draft) operation to 15% of design capacity with 0% of the design power.

3. The curve coefficients were developed by the SSPC90.1 Mechanical Subcommittee and published as default curve coefficients for centrifugal chillers in Appendix J of the first public review draft of Standard 90.1-1989R (ASHRAE 1996).

4. This efficiency rating was the requirement for the ≥300 ton (1055 kW) category of centrifugal chillers in the first public review draft of Standard 90.1-1999 (ASHRAE 1996).

5. This efficiency rating was the requirement for centrifugal fan cooling towers in the first public review draft of Standard 90.1-1989R (ASHRAE 1996).

6. The low speed power for both the 2/3- and 1/2-speed controls is based on the ideal fan law with power proportional to the cube of the speed.

## Cooling Tower Component Model

The cooling tower model used in this study was based on the DOE2 (ver2.1e) simulation program (Winkelman et al. 1993). At the time, the group selected this model because it was readily understood and available. In subsequent studies (Benton et al. 2002), this model was tested and proven reliable over a wide range of manufacturers' data.

The cooling tower model utilized has two curves that together represent the variation of available cooling tower capacity (condenser water flow) as a function of desired approach, range, and wet-bulb temperatures. This model is shown in Equations 1 through 6 below. The two regression curves, FRA and FRB, combine to create a capacity function as a function of the three variables. The regression curve coefficient values are in Table 6.

$$t_R = t_{cwr} - t_{cws} \tag{1}$$

$$t_A = t_{cws} - t_{owb} \tag{2}$$

$$t_A = a + b \times t_R + c \times t_R^2 + d \times FRA + e \times FRA^2 + f \times t_R \times FRA \tag{3}$$

$$FRA = \frac{-d - f \times t_R + \sqrt{(d + f \times t_R)^2 - 4 \times e \times \left(a + b \times t_R + c \times t_R^2 - t_A\right)}}{2 \times e} \tag{4}$$

$$FWB = a + b \times FRA + c \times FRA^2 + d \times t_{owb} + e \times t_{owb}^2 + f \times FRA \times t_{owb} \tag{5}$$

$$q_{available} = q_{rated} \times FWB \times \left(\frac{t_R}{10}\right) \tag{6}$$

where

| | | |
|---|---|---|
| $t_R$ | = | range, °F |
| $t_{cwr}$ | = | condenser water return temperature, °F |
| $t_{cws}$ | = | condenser water supply temperature, °F |
| $t_A$ | = | approach, °F |
| $t_{owb}$ | = | outside wet-bulb temperature, °F |
| *FRA* | = | regression function for tower capacity as a function of range and approach |
| *FWB* | = | regression function for tower capacity as a function of FRA and wet-bulb temperature |
| *a* to *f* | = | regression coefficients (see Table 6 for values) |
| $q_{availble}$ | = | available heat rejection capacity from tower, mbh |
| $q_{rated}$ | = | heat rejection capacity from tower at rated conditions, mbh |

At a given hour, if the available heat rejection capacity exceeds the condenser heat, the model modulates the fans. Figure 1 depicts the percentage power versus percentage heat rejection capacity of single-speed, two-speed 100%/50%, and two-speed 100%/67% fan controls. In addition, Figure 1 shows a curve for variable-speed fan control. It is important to

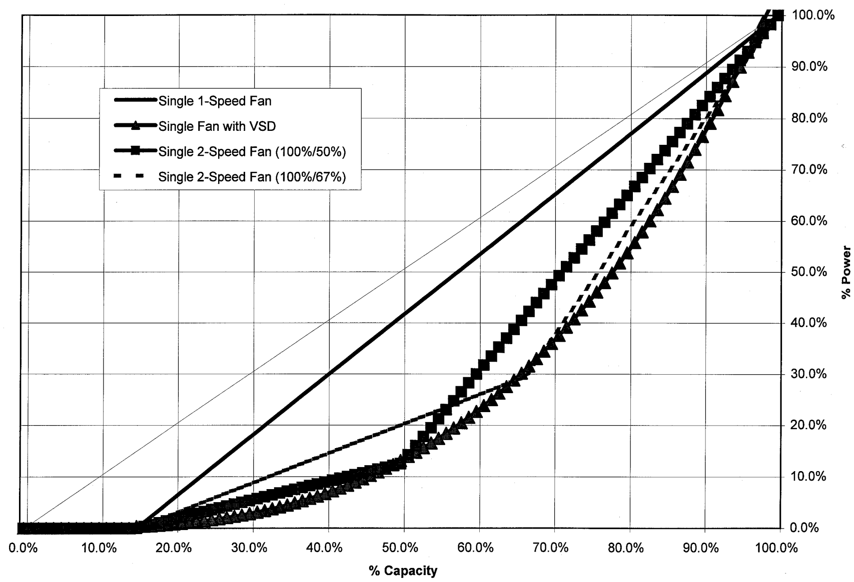


***Figure 1*** *The percentage power versus percentage heat rejection capacity of single-speed, two-speed 100%/50%, and two-speed 100%/67% fan controls.*

note that the two-speed and variable-speed fan models cluster closely together in their part-load performance. The one-speed fan uses significantly more energy as the load decreases. Table 2 summarizes the results at 100%, 75%, 50%, and 25% capacities. At 75% capacity, a single-speed fan uses 20% to 40% more power than the two-speed or variable-speed fans. At 25% capacity, a single-speed fan uses 50% to 85% more power than the two-speed or variable-speed fans. The actual savings will depend on dynamics of the cooling load and coincident wet-bulb temperature as it affects the hours spent at full- and part-load conditions.

It should be noted that the group developed the low-speed operation of these fan curves using the perfect fan laws and did not account for the degradation of motor efficiency at low speed. The variable-speed curve depicted in Figure 1 assumed 5% degradation in motor and drive efficiency at all speeds.

## Electric Chiller Component Model

The electric chiller component model is also based on DOE2 (ver 2.1E) (Winkelman et al. 1993). Like the cooling tower model, the group selected this model because it was readily available and easy to use. In addition, the SSPC90.1 Mechanical Subcommittee had already developed performance curves in this model format to represent all the major compressor and condenser classes of electric chillers. Subsequent studies (PEC 1998; Hydeman et al. 1997) have tested these curve formats using both manufacturers' and field-monitored data and found the curve format to be quite accurate (3% to 5% CVRMSE).

The component chiller model uses the following three regression curves to represent chiller part-load and off-design performance:

**TABLE 2**
**Cooling Tower Fan Model at Part Load: Percent Power as a Function of Percent Load**

| Percent Full Capacity | One-Speed Fan | Single Fan with VSD | Two-Speed Fan (100%/50%) | Two-Speed Fan (100%/67%) |
|---|---|---|---|---|
| 100.0% | 100.0% | 105.0% | 100.0% | 100.0% |
| 75.0% | 70.6% | 44.3% | 56.3% | 47.2% |
| 50.0% | 41.2% | 13.1% | 12.5% | 20.1% |
| 25.0% | 11.8% | 1.6% | 3.6% | 5.7% |

- CAPFT: A curve that represents the variation in capacity with evaporator and condenser temperatures.
- EIRFT: A curve that represents the variation in efficiency with evaporator and condenser temperatures.
- EIRFPLR: A curve that represents the variation in efficiency with part-load conditions. This curve uses an intermediate variable for part-load ratio (PLR).

These curves are represented in Equations 7 through 11. The coefficient values are in Table 5.

$$CAPFT = a_{CAPFT} + b_{CAPFT} \times t_{CHWS} + c_{CAPFT} \times (t_{CHWS})^2 + d_{CAPFT} \times t_{CWS} + e_{CAPFT} \times (t_{CWS})^2 + f_{CAPFT} \times t_{CHWS} \times t_{CWS} \tag{7}$$

$$EIRFT = a_{EIRFT} + b_{EIRFT} \times t_{CHWS} + c_{EIRFT} \times (t_{CHWS})^2 + d_{EIRFT} \times t_{CWS} + e_{EIRFT} \times (t_{CWS})^2 + f_{EIRFT} \times t_{CHWS} \times t_{CWS} \tag{8}$$

$$PLR = \frac{q}{q_{rated} \times CAPFT} \quad (9)$$

$$EIRFPLR = a_{EIRFPLR} + b_{EIRFPLR} \times PLR + c_{EIRFPLR} \times (PLR)^2 \quad (10)$$

$$P = P_{rated} \times CAPFT \times EIRFT \times EIRFPLR \quad (11)$$

where

$CAPFT$ = regression function for chiller capacity
$EIRFT$ = regression function for chiller efficiency
$EIRFPLR$ = regression function for chiller efficiency
$PLR$ = ratio of load to available capacity for chiller
$t_{CHWS}$ = chilled water supply temperature, °F
$t_{CWS}$ = condenser water supply temperature, °F
$a$ to $f$ = regression coefficients (see Table 5 for values)
$q$ = present capacity of chiller, tons
$q_{rated}$ = capacity of chiller at rated conditions, tons
$P$ = present power draw from chiller, kW
$P_{rated}$ = full-load power draw from chiller at rated conditions, kW

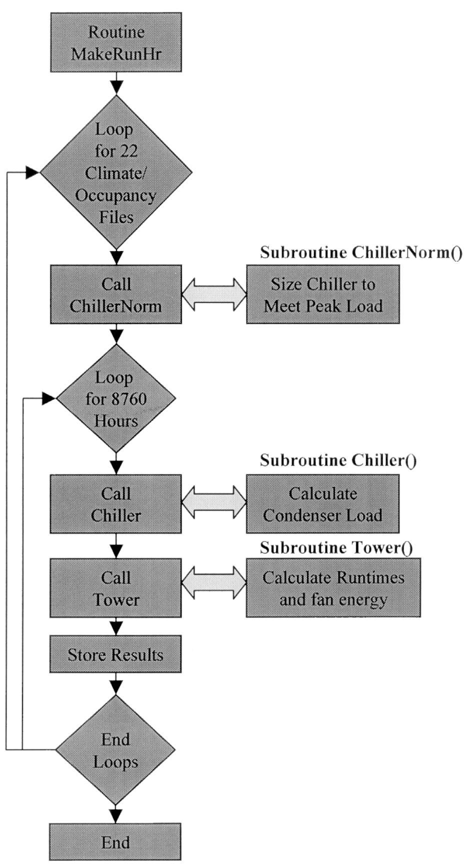


***Figure 2*** *The central plant computer model calculation flow diagram.*

### Central Plant Computer Model Flow Diagram

The central plant computer model calculation flow diagram is represented in Figure 2. This model is composed of a main routine (MakeRunHr()) and three subroutines (ChillerNorm(), Chiller(), and Tower()). The main routine has two loops—one to cycle through each of the 22 climate and occupancy models and another to cycle through each hour of the year. For each of the 22 climate and occupancy files, the subroutine ChillerNorm() is called to size the chiller for the peak load at a condenser water temperature corresponding to the approach and the peak wet-bulb temperature (noncoincident). In the hourly loop, the subroutines Chiller() and Tower() are called to estimate the hourly heat rejection and cooling tower fan power usage.

### Cost Estimate

A member of TC 8.6 developed the cost premiums for fan speed controls. These costs included the premium for a two-speed motor, the starter, wiring, installation, and controls. A 25% factor was added to account for contractor markup and safety. These numbers are presented in Table 4.

### Life-Cycle Cost Model

The life-cycle costs were based on the criteria developed by SSPC90.1 and consistent with the other requirements of that standard. These include

- 0.08$/kWh for electricity (flat rate),
- 20-year project life,[7] and
- 12.1% real discount rate (7.4 scalar).

The calculation of life-cycle cost-effectiveness is shown in Equation 12. A positive number represents a cost-effective investment (or positive net present worth) based on the discount rate and project life. In this equation, ΔFLEOH represents the reduced full-load equivalent operating hours of a form of fan control, HP is the horsepower of the motor, and PW is the present worth.

$$PW = \Delta FLEOH \times HP \times 0.7457 \times ElectricRate \times Scalar - \Delta InstalledCost \quad (12)$$

## RESULTS

The detailed results of the simulation model are presented in Table 3. This table presents the full-load equivalent operating hours (FLEOH) of the two-speed 100%/50%, two-speed 100%/67%, and one-speed cooling tower fan controls. Identified in the left column are the file codes representing the city and occupancy. The construction activity weighted averages are presented in the final row. These averages were used in the

[7] Table 3 in chapter 33 of the *1995 ASHRAE Handbook—HVAC Application* lists the service lives of wood and galvanized cooling towers as 20 years, electric motors as 18 years, and motor starters as 17 years (ASHRAE 1995).

**TABLE 3**
**Hourly Simulation Results by Climate and Occupancy**

| File | Weight | Cooling Tower (Two-Speed 100%/50%) | | | Cooling Tower (Two-Speed 100%/67%) | | | CT One-Speed |
|---|---|---|---|---|---|---|---|---|
| Occupancy and Climate Code | | FLEOH (h/yr) | Run Time (Full speed h/yr) | Run Time (Half speed h/yr) | FLEOH (h/yr) | Run Time (Full Speed h/yr) | Run Time (2/3 Speed h/yr) | Run Time (Full Speed h/yr) |
| Office DEN | 0.89% | 1027 | 767 | 2082 | 1061 | 525 | 1808 | 1624 |
| Office DET | 9.55% | 981 | 730 | 2011 | 1006 | 482 | 1771 | 1558 |
| Office FRS | 1.31% | 1644 | 4273 | 2962 | 1689 | 925 | 2580 | 2493 |
| Office KNX | 10.15% | 1813 | 1556 | 2060 | 1802 | 1229 | 1934 | 2404 |
| Office LAX | 6.23% | 1993 | 1560 | 3466 | 2036 | 1131 | 3053 | 2987 |
| Office MNP | 0.91% | 867 | 651 | 1727 | 897 | 454 | 1495 | 1362 |
| Office ORL | 3.14% | 3163 | 2793 | 2955 | 3127 | 2288 | 2833 | 4010 |
| Office PHX | 0.71% | 2377 | 1920 | 3654 | 2434 | 1492 | 3180 | 3425 |
| Office PRV | 5.30% | 911 | 684 | 1812 | 945 | 483 | 1558 | 1430 |
| Office SEA | 2.21% | 763 | 562 | 1612 | 796 | 388 | 1377 | 1225 |
| Office SHR | 5.18% | 2420 | 2129 | 2326 | 2404 | 1754 | 2192 | 3087 |
| Retail DEN | 1.82% | 1145 | 928 | 1735 | 1159 | 699 | 1551 | 1642 |
| Retail DET | 10.39% | 1074 | 852 | 1777 | 1086 | 612 | 1600 | 1584 |
| Retail FRS | 2.09% | 1655 | 1348 | 2452 | 1682 | 1040 | 2169 | 2358 |
| Retail KNX | 9.03% | 1852 | 1622 | 1836 | 1854 | 1356 | 1682 | 2378 |
| Retail LAX | 7.25% | 2001 | 1614 | 3095 | 2046 | 1245 | 2705 | 2889 |
| Retail MNP | 1.47% | 898 | 697 | 1609 | 925 | 512 | 1394 | 1359 |
| Retail ORL | 5.27% | 3209 | 2902 | 2457 | 3175 | 2472 | 2372 | 3914 |
| Retail PHX | 1.74% | 2294 | 1931 | 2905 | 2328 | 1567 | 2556 | 3127 |
| Retail PRV | 6.77% | 954 | 739 | 1725 | 983 | 541 | 1494 | 1449 |
| Retail SEA | 1.74% | 798 | 607 | 1530 | 829 | 442 | 1308 | 1237 |
| Retail SHR | 6.84% | 2367 | 2111 | 2041 | 2365 | 1806 | 1884 | 2952 |
| **Weighted Average** | | **1689** | **1413** | **2205** | **1702** | **1113** | **1988** | **2322** |

**TABLE 4**
**Life-Cycle Cost Results**

| Motor Size | Run Hours Single-Speed | Run Hours Two-Speed | Cost Premium | Markup | Installed Cost Premium | Operating Cost Savings | Life-Cycle Cost Savings |
|---|---|---|---|---|---|---|---|
| hp | h/yr | h/yr | $Premium | % | $ | PW$ | PW$ |
| 3 | 2322 | 1689 | $745 | 125% | $931 | $840 | $(91) |
| 5 | 2322 | 1689 | $882 | 125% | $1103 | $1401 | $298 |
| 7.5 | 2322 | 1689 | $993 | 125% | $1241 | $2101 | $860 |
| 10 | 2322 | 1689 | $1348 | 125% | $1685 | $2801 | $1116 |

final life-cycle cost analysis (Table 4). Note that there was very little difference between the run hours of the two-speed 100%/50% and two-speed 100%/67% controls (1689 and 1702 h/yr, respectively). Both of the two-speed controls saved significant amounts of energy compared to the one-speed model (2,322 h/yr). As previously noted, these efficiency differences are reflected in Figure 1.

*Note:* The full-load equivalent operating hours (FLEOH) and weighted averages were calculated as shown in Equations 13 to 15.

$$FLEOH_{100/50} = RT_{100} + \left(\frac{1}{2}\right)^3 \times RT_{50} \quad (13)$$

$$FLEOH_{100/67} = RT_{100} + \left(\frac{2}{3}\right)^3 \times RT_{67} \quad (14)$$

$$RT_{weighted\_avg} = \frac{\sum_i (RT_i \times Weight_i)}{22} \quad (15)$$

The results of the life-cycle cost analysis are presented in Table 4. Note that this study indicates a positive present worth (benefits outweighing costs) for 5 hp (3.7 kW) motors. By consensus of both the working group and, subsequently, the SSPC90.1, the requirement was set at 7.5 hp (5.6 kW) to provide a margin for safety. The group considered a conservative approach prudent as this requirement regulates products that previously were not covered by any energy code.

It should be noted that no credit was taken for the increased motor and drive life that a multiple speed arrangement will have due to the reduced fan cycling.

Based on these results and two rounds of public review, the final requirement in Standard 90.1-1999 is as follows (ASHRAE 1999):

> **6.3.5 Heat Rejection Equipment.**
>
> **6.3.5.1 General.** This subsection applies to heat rejection equipment used in comfort cooling systems such as air-cooled condensers, open cooling towers, closed circuit cooling towers, and evaporative condensers.
>
> **Exception to 6.3.5.1:** Heat rejection devices included as an integral part of the equipment listed in Tables 6.2.1A through 6.2.1D.
>
> **6.3.5.2 Fan Speed Control.** Each fan powered by a motor of 7.5 hp (5.6 kW) or larger shall have the capability to operate that fan at two-thirds of full speed or less, and shall have controls that automatically change the fan speed to control the leaving fluid temperature or condensing temperature/pressure of the heat rejection device.
>
> **Exceptions to 6.3.5.2:**
>
> (a) Condenser fans serving multiple refrigerant circuits.
>
> (b) Condenser fans serving flooded condensers.
>
> (c) Installations located in climates with greater than 7200 CDD50 (4000 CDD10).
>
> (d) Up to 1/3 of the fans on a condenser or tower with multiple fans where the lead fans comply with the speed control requirement.

It is important to note that packaged units with air-cooled or evaporatively cooled condensers are exempt from these requirements when the condenser fan is part of the overall unit efficiency ratings covered in the equipment efficiency tables in the standard. Also, equipment for commercial or industrial process cooling, such as condensers for reach-in coolers, are exempt.

## DISCUSSION

There were 47 comments on the first public review draft of this requirement and 5 comments on the second public review draft. In response to these comments, the SSPC90.1 added the four exceptions to 6.3.5.2. The first and second exceptions were added based on increased complexity of controls for systems with multiple refrigerant circuits and flooded condensers that were not considered in the original life-cycle cost analysis. The climate exception (c) represents the reduced savings seen in the calculation for cool climates, such as Seattle.

The multiple fan exception (d) represents the fact that we only performed analysis on a single-cell tower with a single fan. Many cooling towers and air-cooled condensers have multiple fans that cycle to maintain control. The capacity versus power chart for towers with two and three fans are presented in Figures 3 and 4. Consider, for example, an air-cooled condenser configured with either a single one-speed fan or two one-speed fans installed where the air is not isolated between the two fans. Figure 3 indicates that the action of multiple fans with one-speed motors is actually worse than that of a single one-speed fan cycling on the entire load. This phenomenon is due to the fact that a single fan during cycling will provide half of the cooling capacity with less than half of the power (due to the free cooling effect when the fan is off). Two fans with one-speed motors will produce half of the cooling with exactly half of the power. Savings are only accrued by having two-speed controls on all fans. Recognizing that there is an increased cost for each additional fan control (the law of diminishing returns), the SSPC 90.1 voted to limit the speed control requirement to two of every three fans.

## CONCLUSIONS

Five years after the study, the fan speed control requirement was published in the final version of Standard 90.1-1999. The fact that this requirement, a new provision for heat rejection equipment, survived through two rounds of public review is a testament to the collaborative process in which it was born. This requirement has persisted because it was based on the foundation of a thorough LCC study and developed with the input and support of the cognizant technical committee.


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- Bob Shriver, EVAPCO, TC 8.6
- Jon Edmonds, Edmonds Engineering Co., TC 8.6

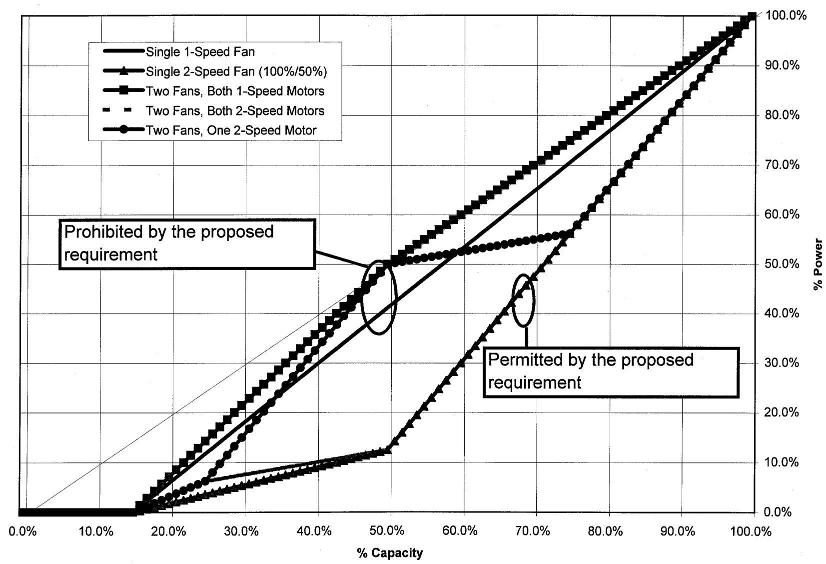


*Figure 3*

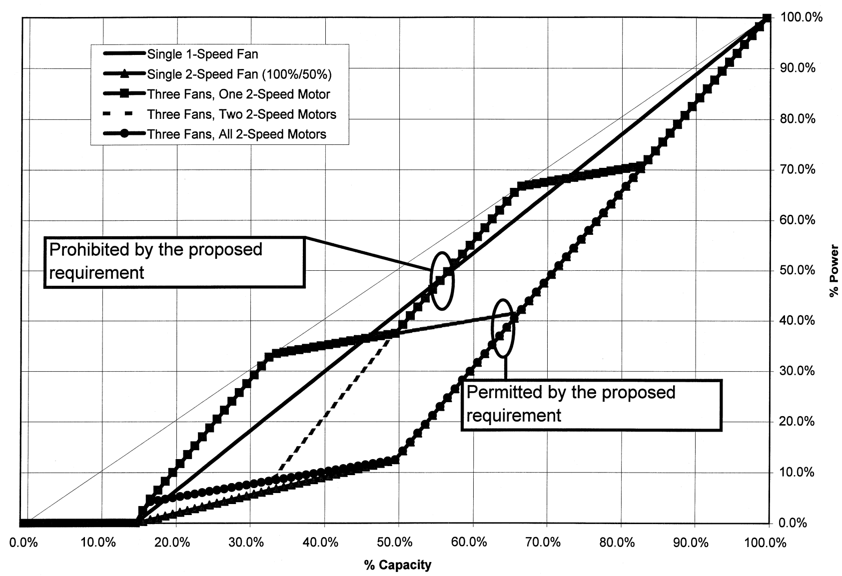


*Figure 4*


- Keith Emerson, Public Service Company of Colorado, SSPC 90.1
- Paul DuPont, PAE Consulting Engineering, SSPC 90.1
- Mick Schwedler, Trane Company, SSPC 90.1

and the leadership provided by the following chairs of both TC 8.6 and Standard 90.1:

- Tom Carter, Baltimore Air Coil, TC 8.6 Chair
- Ron Jarnagin, Battelle Pacific Northwest National Laboratories, SSPC 90.1 Chair


## REFERENCES


ARI. 1992. *ARI Standard 550-92, Standard for centrifugal and rotary screw water-chilling packages.* Arlington Va.: American Refrigeration Institute.

ASHRAE. 1995. *1995 ASHRAE Handbook—HVAC Applications*, p. 33.4. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1996. *BSR/ASHRAE/IESNA 90.1-1989R, Energy standard for buildings except low-rise residential buildings*. First public review draft, March 1996. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ASHRAE. 1999. *ANSI/ASHRAE/IESNA 90.1-1999R, Energy standard for buildings except low-rise residential buildings*. Atlanta: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.

Benton, D.J., C.F. Bowman, P. Miller, and M. Hydeman. 2002. An improved cooling tower algorithm. *ASHRAE Transactions* 108(1).

CTI. 1996. *CTI Standard 201-96, Standard for certification of water cooling tower thermal performance*. Houston Tex.: Cooling Tower Institute.

Hadley, D.L., and R.E. Jarnagin. 1993. *Selecting representative climate stations for use in a building energy model*, CLIMA 2000 Conference.

Hydeman, M., K. Gillespie, and R. Kammerud. 1997. PG&E's CoolTools project: A toolkit to improve evaluation and operation of chilled water plants. Presented at the Cool$ense National Forum on integrated chilled water retrofits. Berkeley, Calif.: Lawrence Berkeley National Laboratories.

PEC. 1998. CoolTools™ Project. http://www.hvacexchange.com/cooltools/index.html. San Francisco Calif.: Pacific Energy Center.

Winklemann F.C., et al. 1993. *DOE-2 Supplement, Version 2.1E*. Berkeley, Calif.: Lawrence Berkeley National Laboratories.

## BIBLIOGRAPHIES

BLAST. See website http://www.eren.doe.gov/buildings/tools_directory/software/blast.htm.

DOE2. See website http://www.eren.doe.gov/buildings/tools_directory/software/doe-2.htm.

CTI. 1997. *CTI ACT 105-97, Acceptance Test code for water cooling towers.* Houston Tex.: Cooling Tower Institute.

## APPENDIX

**TABLE 5**
**Chiller Curve Coefficients**

| | a | b | c | d | e | f |
|---|---|---|---|---|---|---|
| CAPFT | 0.57215645 | 0.03922805 | (0.00087059) | (0.00708085) | (0.00033479) | 0.00091124 |
| EIRFT | 1.00951828 | (0.00430107) | (0.00010280) | (0.00309426) | 0.00006078 | 0.00005418 |
| EIRFPLR | 0.16102213 | 0.58578139 | 0.25319648 | | | |

**TABLE 6**
**Cooling Tower Curve Coefficients**

| Coefficient | FRA | FWB |
|---|---|---|
| a | –2.22888899 | 0.60531402 |
| b | 0.16679543 | –0.03554536 |
| c | –0.01410247 | 0.00804083 |
| d | 0.03222333 | –0.02860259 |
| e | 0.18560214 | 0.00024972 |
| f | 0.2451871 | 0.00490857 |