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CTI Comments â€“ Title 20 Phase II Pre-Rulemaking â€“ Commercial and Industrial

Comments on the Title 20 Phase II Pre-Rulemaking - Commercial and Industrial Fans and Blowers, submitted on behalf of the Cooling Technology Institute.

Additional submitted attachment is included below.

COOLING TECHNOLOGY INSTITUTE
P.O. BOX 681807, HOUSTON, TEXAS 77268-1807
3845 CYPRESS CREEK PARKWAY, SUITE 420, HOUSTON, TEXAS 77068
Phone: 281/583-4087 Fax: 281/537-1721

email: vmanser@cti.org http://www.cti.org

June 16, 2017

California Energy Commission
Docket Unit, MS-4
Re: Docket No. 17-AAER-06
1516 Ninth Street
Sacramento, California 95814-5512

Reference CTI Comments – Title 20 Phase II Pre-Rulemaking – Commercial and Industrial Fans & Blowers [Docket No. 17-AAER-06]

Dear CEC Staff:

These comments are submitted on behalf of the Cooling Technology Institute in response to the California Energy Commission (CEC) Phase II Pre-Rulemaking Invitation to Participate meeting held on May 11, 2017, regarding the proposal to establish minimum efficiency standards for commercial and industrial fans into California's Appliance Efficiency Standards in Title 20 of the California Code of Regulations, Sections 1601 through 1609.

The Cooling Technology Institute (CTI) is a broad based industry association of owners / operators, suppliers, and manufacturers of cooling equipment whose mission is to advocate and promote, for the benefit of the public, the use of all environmentally responsible commercial and industrial cooling technologies, such as wet cooling towers, air-cooled condensers, dry coolers, indirect cooling, and hybrid systems, by encouraging:

- Education on these technologies
- Development of standards and guidelines
- Development, use, and oversight of independent performance verification and certification programs
- Research to improve these technologies

- Advocacy and dialog on the benefits of cooling technologies with Government Agencies and other organizations with shared interests
- Technical information exchange

We thank the CEC for the opportunity to comment on the Phase II Pre-Rulemaking for Commercial and Industrial Fans and Blowers.

CTI Position on CEC Fan Efficiency Rulemaking for Heat Rejection Equipment

After careful evaluation and study, the Cooling Technology Institute requested and was granted an exemption for heat rejection equipment, including air cooled, evaporatively cooled, and hybrid combinations, from the proposed USDOE Fan Efficiency Requirements. In 2015, the CTI was a voting member of the Department of Energy (DOE) Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) Commercial and Industrial Fans and Blowers Working Group formed to negotiate energy conservation standards and test procedures for fans and blowers. Note that the CTI voted for the term sheet, which contained an exemption for heat rejection equipment along with many other consensus-based agreements.

The exempted equipment definitions for heat rejection equipment as shown in the September 24, 2015 Term Sheet (attached) represent a complete and accurate description of the exemptions for both packaged and field erected equipment, including open circuit cooling towers, closed circuit cooling towers, evaporative condensers, air cooled condensers, dry coolers, and hybrid wet / dry versions of these devices. On this basis, the CTI requests an exemption from any future California Title 20 Fan Efficiency Requirements, based on the following justifications:

Low Potential Energy Savings in Both Commercial and Industrial Heat Rejection Applications

From the graph below, taken from a DOE sponsored study, Exhaust Fans, Supply & Return Fans, and Fan Powered Terminal Boxes account for over 93% of the total energy consumed by fans in Commercial Building HVAC Systems in the United States (note that the energy in the graph below that is allocated to pumps has been subtracted out of this total). Increasing the efficiency of these specific fans has the potential to have a major impact on energy use.

In contrast, fans used in outdoor evaporative heat rejection equipment where movement of air is secondary to the main function of the device (such as an air cooled chiller, evaporative condenser, or cooling tower) account for only 6.6% of the fan energy use in Commercial Building HVAC Systems in the United States per the referenced DOE sponsored study.

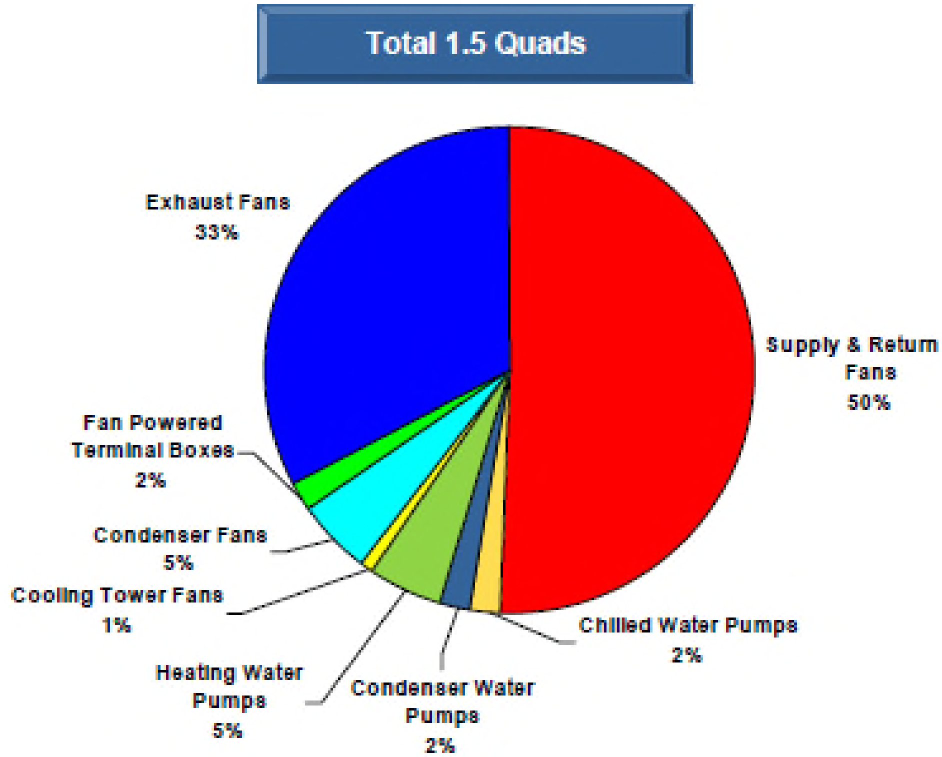


Figure 1-1: Parasitic Primary Energy Use -- Equipment Breakdown

Source: Energy Consumption Characteristics of Commercial Building HVAC Systems Volume II: Thermal Distribution, Auxiliary Equipment, and Ventilation, October 1999; download available at: https://pdfs.semanticscholar.org/d917/85a024eb854e0fb88bb8e886979878d0cf0b.pdf

Specifically, cooling towers account for a little over 1% of the energy use per the graph. Some evaporative condensers may be included in the condenser fans number, which would make the evaporative heat rejection percentage slightly higher. Note that the percentage is slightly smaller if pump energy is included as in the above chart. Fans used on heat rejection equipment for industrial applications would also follow a similar pattern, accounting for a very small percentage of overall fan energy use.

Energy Efficiency of Primary Product Function is More Appropriate

Heat rejection equipment typically utilizes efficiency metrics, such as kW/ton of air conditioning capacity, BTUH/HP, or gpm/HP at standard conditions that take into the account the energy efficiency to deliver the heat rejection performance of the entire unit and not just the individual fan component. These metrics are broadly used in both commercial and industrial industries to evaluate the overall unit efficiency as well as compare one model versus another. In turn, the performance of the heat rejection equipment has a large impact on the energy use of the system to which it is connected, often by several orders of magnitude, such as a cooling tower (0.056 kW/ton or less) coupled with a water cooled chiller (approximately 0.60 kW/ton).

Minimum Energy Efficiency Requirements are Widely Followed in the United States, including California

Fans and their associated housings and drive systems are designed and selected close to their optimum efficiency point in order to maximize the heat rejection value produced as measured by the primary unit efficiency metrics described above. This is also true of other components in heat rejection units, such as heat exchangers, fill (heat transfer media), and pumps. This drive for high component efficiency is required to comply with Federal requirements, energy codes, and standards (such as California Title 24, ASHRAE Standard 90.1, and the International Energy Conservation Code) as well as the needs of a competitive marketplace. Title 24, the IECC, and Standard 90.1 collectively are widely adopted into building energy codes throughout the United States and are generally harmonized as to the requirements for equipment that reject heat to the ambient (though California took the lead and was the first to include requirements for heat rejection equipment along with independent certification of thermal performance).

While these standards may not be uniformly enforced throughout the country, the level of compliance by manufacturers to the strictest of the standards is high and nation-wide, applying across a wide range of markets, including both commercial building and industrial applications. The latest editions of these standards also serve as the "standard of care" for most system designers that utilize heat rejection equipment, independent of local adoption of Standard 90.1 and IECC updates. Furthermore, the thermal performance of the vast majority of heat rejection equipment, such as open and closed circuit cooling towers, produced and sold in the United States are independently certified for thermal performance by independent third party organizations such as the Cooling Technology Institute, assuring full thermal performance is achieved at the rated electric power consumption.

Unintended Consequences of Fan Efficiency Requirements

Specific fan efficiency regulations for heat rejection equipment, including evaporative heat rejection units, will place a large, unnecessary testing and compliance burden on manufacturers for little expected gain, while distracting design teams from the optimization of overall unit performance. Metrics such as Fan Efficiency Ratio (FER), Fan Efficiency Grade (FEG) or Performance Based Efficiency Requirement (PBER) are not relevant for fan based heat rejection equipment. Optimization of these metrics rather than the heat rejection per energy input could actually lead to ***net increases*** in actual fan power consumption for the same heat rejection even if the higher calculated fan efficiency metric is met. Reference the attached fan efficiency example which illustrates this point.

Design Requirements are at Peak Outdoor Conditions

Heat rejection equipment is selected for design day duty, resulting in the equipment being oversized for combinations of "off peak" loads and ambient temperatures less than design. In addition, virtually all fan powered heat rejection devices utilize low cost, highly efficient variable speed drives to modulate the fan speed in response to the system need for cooling. Thus fans are typically operated at reduced speed for much of the year in most climates, significantly reducing the energy consumed by these fans thanks to the fan laws (for instance, at 50% fan speed, only 1/8 of the full load power is drawn). The widespread use of variable speed drives for part load operation further reduces the need for specific fan efficiency requirements and is required by Code, including Title 24-2016, for fan powered heat rejection devices over 7.5 HP per motor (note Standard 90.1-2016 calls for a total connected load limit of 5 HP per device, including the service factor of the motor, for VSD use).

Regulatory Sound Requirements Conflict with Peak Fan Efficiency

Low noise applications are very common in the heat rejection industry, requiring the use of low sound axial fan designs with higher fan solidity (i.e., more and / or wider chord fan blades) to move the same amount of air through the unit with a slower fan speed. By the nature of such designs, these fans have slightly lower fan efficiency but are often required to meet local sound codes. On occasion, the use of centrifugal fan designs with sound attenuation are utilized to meet critical sound requirements. Note that the heat rejection equipment with such low sound fan designs must still meet the overall energy efficiency metric as called for in the energy codes and standards for that equipment.

Application Utility Constrained

The heat exchanger is often a part of the fan housing and can have many possible arrangements, leading to a wide variation of fan entrance and exit conditions. Structural blockage may be necessary to make the product function properly and comply with other standards such as for wind and seismic structural resistance. Testing embedded fans for compliance in such arrangements would be impractical at best. Additionally, this product class is frequently limited in configuration by shipping width. Fan diameters tend to maximize at allowable shipping width increments. For these reasons, the fan design and selection is often incompatible with a pure efficiency focus.

Fan efficiency typically increases with fan diameter, so if the fan diameter is limited for shipping, a way to increase the fan efficiency would be to reduce the heat exchanger size and increase air flow, resulting in an increase in net power required, though higher peak fan efficiency might be realized. The nature of outdoor heat rejection equipment is such that optimization of the sum of all of the parts of the unit together is a desirable path to maximize heat rejection energy efficiency and in turn, overall system efficiency. Forcing a particular efficiency level for an individual component is very likely to have unexpected consequences. For instance, requiring a more efficient but costly fan may force a manufacturer to reduce the size of the heat exchanger in order to reduce the cost for the same thermal performance, negating the positive impact of the improved fan. Thus unit power consumption would likely not go down in the case of heat rejection devices, as appears to be anticipated by the intent of the efficiency rule.

Application in Heat Rejection Equipment is considered to be Severe Duty for Fans

Fans used in heat rejection equipment, which are most often installed outdoors, are subjected to service limitations that limit the peak efficiency that is achievable. Such fans are subjected to high temperatures and very high humidity conditions, especially in regards to evaporative heat rejection units, which limit the types of fans that can be applied both in terms of materials that will survive the environment and in terms of the types of designs that will not trap condensed water and become unbalanced or corrode. In the case of evaporative heat rejection designs, the fan blades also must be able to survive the impact of the small amount of carryover water droplets that hit the leading edge at high velocity. A reduced subset of fan designs can meet these criteria, and fans specifically designed for evaporative heat rejection equipment service are typically utilized.

In addition, outdoor equipment is subject to wind and solar effects that can deflect the housings. Equipment is thus designed with large enough tip clearance between the fan blades and the fan enclosure/shroud that will enable the equipment to operate without the potential for damage and in a worst case, a catastrophic fan failure. Seismic requirements may also require increased tip clearance. Higher tip clearance has an adverse impact on peak efficiency potential but is necessary for public safety.

Diverse Application Points Require Very Diverse Potential Model Configurations

Lastly, heat rejection products are not mass-produced but instead are typically manufactured to order, often with thousands of potential model configurations for a single product line in order to meet a wide range of customer requirements (including sound requirements as mentioned earlier). Note that a high percentage of these unit combinations are not produced in any given year. These product configurations are also applied at many different levels of fan power depending on the system requirements of the building or industrial process in which they are applied, further complicating compliance with specific fan efficiency metrics. On the specific issue of replacement fans for existing heat rejection units in the field, the CTI suggests that like-for-like replacements be allowed for all equipment for reasons of cost and safety.

Summary

Based on the low volume of potential energy savings, the applicability and widespread usage of other, more relevant energy efficiency metrics found in California Title 24, the potential for lack of actual energy savings or even unintended increases in fan and system energy use, and the design and testing challenges detailed above, the CTI recommends that fans used in all heat rejection devices be exempted from any future CEC Title 20 fan efficiency regulations. We have attached a presentation to further illustrate our justifications for this exemption.

The Cooling Technology Institute (CTI) appreciates this opportunity to comment on the proposed rulemaking. If there are any questions on our comments, please feel free to contact me at the phone number or email below.

Sincerely,

Paul Lindahl ( ) for

Bill W. Howard, P.E.
President
Cooling Technology Institute
Office: 720.746.1234
Cell: 303.241.6369
Email: bhoward@ctdinc.com

Attachments

ASRAC Term Sheet – September 25 2015
CTI Heat Rejection Highlights Presentation
Fan Efficiency Example

cc:

Virginia Manser	CTI Administrator
Larry Burdick	SPX / Marley
Paul Lindahl	Cooling Industry Relations, LLC
Frank Morrison	Baltimore Aircoil Company
Sarah Ferrari	Evapco

Attachment

Fan Efficiency Example

Below is an example demonstrating that the overall energy efficiency of the heat rejection process needs to be considered, and not just the fan efficiency, when selecting fans for heat rejection equipment. A more efficient fan does not always result in more energy efficient heat rejection.

It is in the best interest of heat rejection equipment manufacturers to use the most efficient fans available for a given fan diameter. Due to the nature of the equipment, the fan diameter is often limited by the geometry of the product. Generally the largest diameter fans that will fit into a product are utilized in order to get the most airflow through the product. This is because thermal performance is proportional to the flow rate of air through the product – the more air, the greater the heat transfer. The desire to squeeze the largest fan into a product must also be weighed against compromising the uniform air inlet transition from the plenum into the fan, which tends to reduce the fan efficiency from its optimum level. However the benefit of a larger fan is that higher air flow rates can be achieved without wasting energy to accelerate the air to the higher discharge velocity associated with a smaller diameter fan.

Table 1 illustrates how the thermal performance of a cooling tower is impacted by reducing the fan diameter from 9' to 8' diameter, and how much more efficient the 8' fan has to be to yield equivalent thermal performance. For calculation purposes in this example, the design airflow is 135,000 CFM, and the static pressure is 1" W.G., which are representative values for a particular size cooling tower. The results can be expressed in several ways:

- In order to achieve the same thermal performance of the cooling tower (i.e., cooling the same flow rate of water to the same temperature conditions at the same fan motor BHp), the 8' fan total efficiency would need to be 13.75% higher than the 9' diameter fan. This is an unrealistic and costly expectation for a fan efficiency improvement.
- Let us look at the case where the 8' diameter fan is 10% more efficient than the 9' diameter fan (which is still an unrealistic expectation). The impact on thermal performance is proportional to the reduction in air flow for a given power draw. So the thermal performance with the 8' fan is only 98.9% of the performance achieved with the lower efficiency 9' fan. In order to obtain the same thermal performance, the fan speed would need to be increased to move the same amount of air flow through the tower. Because fan power draw increases by the cube of the airflow increase (fan laws), the power draw would increase by 3.4 % compared to the tower with the lower efficiency 9' diameter fan. This example demonstrates that it would be counterproductive to overall energy usage to select a higher efficiency, smaller fan.

- Suppose we now examine a case where the 8' diameter fan is only 5% more efficient than the 9' diameter fan (which would be a more realistic expectation). Again, the impact on thermal performance is proportional to the reduction in air flow for a given power draw. In this case, the thermal performance with the 8' fan is only 97.4% of the performance with the original, lower efficiency 9' diameter fan. In order to achieve the same thermal performance (by moving the same amount of airflow through the tower), the power draw with the 8' diameter fan would be 8.3 % higher than the tower with the lower efficiency, 9' diameter fan. This example further demonstrates that it would be counterproductive to the goal of improved energy efficiency to select a smaller diameter fan, even if the total efficiency is higher.

In conclusion, the overall energy efficiency of heat rejection equipment, whether measured in gpm cooled / Hp, BTUH rejected/ kW, or another similar metric is a result of the complex interaction of many different components such as the heat exchanger, the fan housing, the fan diameter, and the fan itself. In the case of evaporative heat transfer equipment, the effect of the spray distribution system also can play a significant role. The above examples demonstrate that trying to optimize a single component can be detrimental to the efficiency of the heat rejection device as well as the much larger system that it is part of.

Table 1: Thermal performance and power draw impact of selecting a smaller, more efficient fan for a cooling tower													
Fan Selection	**Fan Diam. (ft)**	**Airflow (CFM)**	**Static Pressure ("WG)**	**Fan Discharge Area - assuming 1.5' diameter seal disk (ft^2)**	**Fan Discharge Velocity (ft/min)**	**Velocity Pressure at 0.071 lb/ft^3 air density ("WG)**	**Total Pressure (" WG) See note 1**	**Fan Total Air Hp (Hp) See note 2**	**Fan Motor Brake Hp (BHp) See note 3**	**Fan Total Efficiency See note 4**	**Fan Efficiency Increase Required to Draw the Same Power as Base Fan See note 5**	**Thermal Perf. Impact of using the 8' Fan See note 6**	**Fan Motor Hp Increase to get same airflow (same thermal perf.) as Base Fan See note 7**
9' Fan (Base)	9	135,000	1.000	61.85	2,183	0.281	1.281	27.21	40	68.03%	Base	N/A	Base
New 8' Fan	8	135,000	1.000	48.50	2,784	0.457	1.457	30.95	40	77.38%	13.75%	100.00%	0.00%
New 8' Fan (Same Efficiency as Base 9' Fan)	8	129,326	0.918	48.50	2,667	0.420	1.337	27.21	40	68.03%	0.00%	95.80%	13.75%
New 8' Fan (5% More Efficient than Base 9' Fan)	8	131,447	0.948	48.50	2,710	0.434	1.382	28.57	40	71.43%	5.00%	97.37%	8.33%
New 8' Fan (10 % More Efficient than Base 9' Fan)	8	133,501	0.978	48.50	2,753	0.447	1.425	29.93	40	74.83%	10.00%	98.89%	3.41%

Table 1 Notes:

1. Total pressure = Static Pressure + Velocity Pressure
2. Fan Air Total Hp = Airflow * Total Pressure / 6356
3. Fan Motor BHp is held constant at 40 Hp in order to determine the impact on thermal performance, because thermal performance is rated at a nominal power.
4. Fan Total Efficiency = Fan Air Total Hp / Fan Motor BHp
5. Fan Efficiency Increase is the amount that the fan efficiency would need to increase over the base value (68.03%) in order to draw the same Fan Motor BHp.
6. Thermal Performance Impact = the ratio of the Airflow with the new fan to the Airflow with the original fan.
7. Fan Motor Hp Increase is the % increase in power draw required to obtain the original Airflow through the unit with the new fan; this value is determined by the fan laws. The increase is the ratio of original Airflow to the new Airflow cubed minus 1. – or - (135,000/Airflow)^3 – 1

CTI Heat Rejection Equipment Position Highlights

16-June-2017

Fans Used in Heat Rejection Equipment are a Small Factor in Commercial Building Systems

Cooing Tower Fans account for only 1% of consumption in commercial buildings

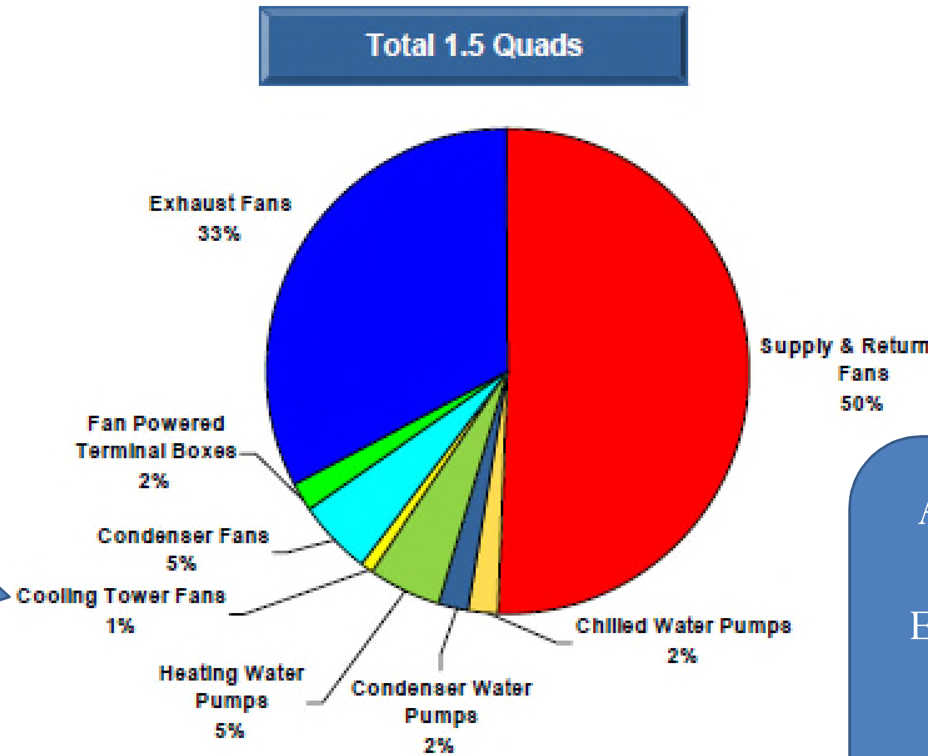


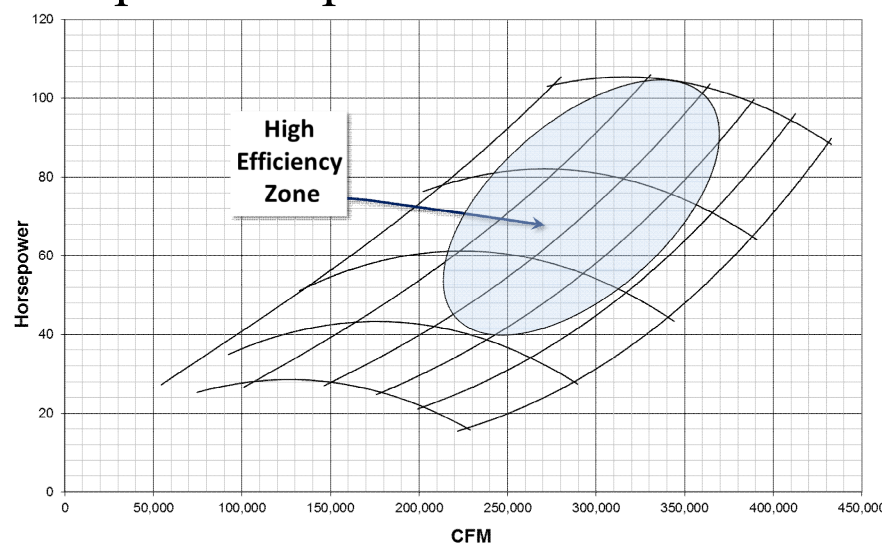
Figure 1-1: Parasitic Primary Energy Use -- Equipment Breakdown

All outdoor Heat Rejection Equipment is only 6% of total fan energy in commercial buildings

Source: Energy Consumption Characteristics of Commercial Building HVAC Systems Volume II: Thermal Distribution, Auxiliary Equipment, and Ventilation, October 1999

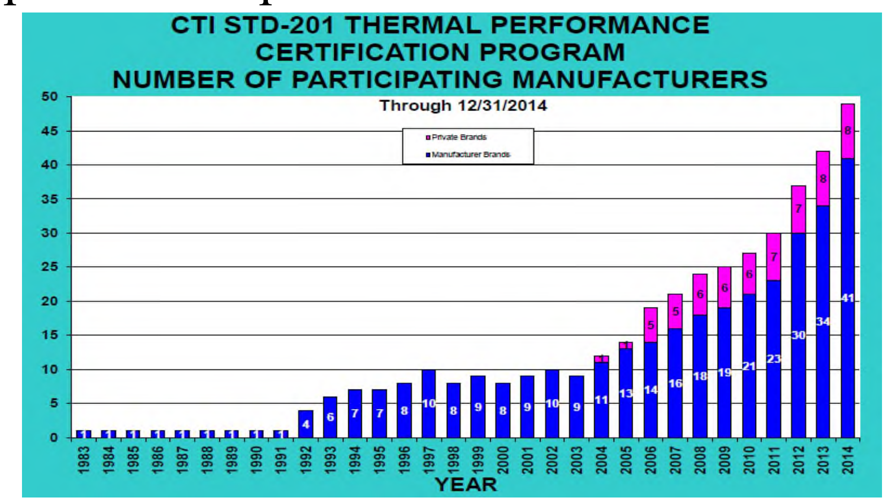
Energy Efficiency

- Estimated Heat Rejection Equipment annual power connection is only 34,500 HP in the U.S. (even less when looking at California only)
  - Low value due to load / weather variation, variable speed drives, and staged operation
- Optimized system designs for both preconfigured and custom tower designs use drive ratios and pitch adjustment to minimize fan power consumption at peak load



Product Energy Efficiency

- Heat Rejection Equipment utilizes metrics with heat rejection divided by energy input (i.e., BTUH / HP)
- All units in the sold in the US are expected to meet or exceed minimum efficiencies mandated by either California Title 24 (in CA), Standard 90.1, or the IECC, all of which are recognized as the "Standard of Care"
- CTI Certification is mandated in the standards referenced above for open and closed circuit cooling tower equipment in the U.S. Verifiable compliance via public CTI website. http://cti.org/certification.php



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Product Energy Efficiency

- Example: Customer needs 3,000 GPM of water cooled from 95°F to 85°F with 78°F entering air wet-bulb temperature:

Axial Fan Open Circuit Cooling Tower Model	Tower Pumping Head (psi)	Total Fan Motor Power (HP)	First Cost Premium to Customer	Estimated Fan Static Efficiency	Fan Diameter (ft)	Required Increase in Fan Efficiency of Base Unit to Match Larger Lower HP Unit	Title 24 Tower Efficiency (GPM/HP)
Tower Z: 12'Wx21.5'Lx20.5'H **Operating weight: 37,520 lbs**	8.44	75	Base	55%	11	Base	47.5
Tower A: 14'Wx24'Lx22'H **Operating Weight: 47,380 lbs**	9.02	40	14%	52%	13	88%	88.8

- The alternate case (Tower A) with a greater heat exchange surface requires only 53% as much fan power, yet offers a reasonable first cost premium over Tower Z. However, note that the alternate tower "Z" pumping head is 7% higher, the operating weight is 26% greater, and the tower plan area for steel support is 30% larger, offsetting some of the energy benefit.
- Increasing the heat exchanger size is the only practical method the reduce energy consumption. Increasing the fan efficiency of the base tower by 85% to achieve the same benefit is not feasible.
- Cooling towers evaluated are galvanized steel with stainless steel cold water basin.
- Annual energy saving estimate (using Tower A versus Tower Z):

(75-40)HP * 0.746 KW/HP * 8760 Hours/yr * 50% duty cycle * $0.134/KWhr

= $15, 325/year Savings

Concerns with Fan Efficiency Focus

- Heat rejection equipment accounts for only 5% to 10% of the total energy use in a typical system, but is critical to the overall energy efficiency of the system
  - For instance, a cooling tower operates at 0.06 kW/ton while a water cooled chiller operates at 0.60 kW/ton (note that an air cooled chiller operates at 1.2 kW/ton)
    - Chiller energy is 10X the tower energy
- Unintended consequences
  - Imposed changes to fan efficiency would likely result in redesign with smaller heat exchangers to manage cost, in turn lowering the efficiency (GPM/HP, BTUH/kW, etc.), which is the true metric for heat rejection equipment efficiency (versus individual component efficiency)

Concerns with Fan Efficiency Focus

- Design requirements are determined at peak outdoor conditions; however:
  - Title 24 mandates heat rejection systems with 7.5 HP and above motors to have variable speed drives*; as a result the vast majority of fans for heat rejection duty are controlled by variable speed drives
  - Lower ambient temperatures reduce heat rejection demand while increasing heat rejection capability. Fan loads can be extremely reduced from design peak through the use of VS drives.
  - "Natural draft" or "fans off" mode is a common practice for lower ambient temperatures for refrigeration and HVAC applications with evaporative cooling
- For these reasons, the actual energy consumption is well below design on annual basis.

*** Title 24-2016 Section 140.4 (h) 2.**

Note that Standard 90.1-2016 calls for units with a total connected HP of 5 HP or more to have variable speed capability, which is an opportunity for California.

Constraints on the Package

- Improved fan efficiency attempted by increasing the fan diameter are often constrained by unit size limitations
- Unit sizes limited to allowable shipping widths and heights
  - Larger blower or fan won’t fit up to the unit
  - Freight shipping limits on large, axial fan heat rejection equipment:
    - 8.5’ legal width limit commonly uses 7.5’ diameter fan.
    - 12’ permit width limit commonly uses 11’ diameter fan.
    - 14’ escort permit width limit commonly uses 13’ diameter fan

Uniqueness of the Fan Design

- Large range in diameter, from 12" up to 40 feet in diameter
- Specific designs to withstand ambient exposure, drift, corrosion, operating environment, pressure rise, and blade stress; safety and low noise requirements also important factors
- Wide use of adjustable pitch and speed optimization to minimize energy consumption
- Application utility constrained.
- <u>Not</u> the same as a Panel Fan

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Uniqueness of the Fan Design cont.

- Adequate test quality using large scaling factors of model fans and fan laws is not possible to validate performance on large diameters
- There is no reliable validation of the airflow and static measurement of fans embedded in equipment and that are larger than can be lab tested
- Heat Rejection Equipment is qualified to the industry using a measure of heat rejection relative to the electrical energy draw (ex. GPM/HP, BTUH/KW etc…)

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Uniqueness of the Fan Design cont.

- Utility of Forward Curved blowers used on Heat Rejection equipment is unique
  - Forced draft blowers on Heat Rejection equipment operate in a moderate pressure rise region with low noise output
  - Forward curved offers similar efficiency as backward curved in this operating region but at lower sound
  - Heat Rejection equipment is commonly placed in sound sensitive locations within the property boundary, therefore minimizing sound is paramount to comply with sound regulations
  - Backward inclined blowers of the same diameter must operate at higher speed (resulting in higher sound level) to deliver the same air duty point. At the same speed a larger diameter BI fan would be required, which would require a unit redesign and may not fit within the allowed enclosure
  - Reduction in the availability of FC fans for other uses can impact their use in heat rejection equipment

Conclusion

- On the basis of these justifications, the CTI, on behalf of its members, has requested an exemption for fans used in heat rejection equipment
  - Reference CTI submission during comment phase
- The CTI looks forward to working with the CEC and other stakeholders on specific wording for this exemption to avoid loopholes for non-exempt applications
- Questions? Contact:

Larry Burdick
Director Global Technical Services
SPX Cooling Technologies
7401 W. 129th Street
Overland Park, KS 66213
TEL +1 913-664-7456
MOB +1 913-636-2037
larry.burdick@spx.com

Appliance Standards and Rulemaking Federal Advisory Committee
Commercial and Industrial Fans and Blowers Working Group
Term Sheet
September 3, 2015 (edited September 24, 2015)

Background

On April 1, 2015, DOE issued a Notice of Intent to Establish the Fans and Blowers Working Group to negotiate a Notice of Proposed Rulemaking (NOPR) for energy conservation standards and test procedures for fans and blowers. 80 FR 17359. This working group is established under the Appliance Standards and Rulemaking Federal Advisory Committee (ASRAC) in accordance with the Federal Advisory Committee Act and the Negotiated Rulemaking Act. The purpose of the working group was to discuss and, if possible, reach consensus on the scope of the rulemaking, certain key aspects of a proposed test procedure, and proposed energy conservation standard for fans and blowers, as authorized by the Energy Policy and Conservation Act (EPCA) of 1975, as amended. The working group was to consist of representatives of parties having a defined stake in the outcome of the proposed standards, and to consult as appropriate with a range of experts on technical issues.

DOE received 25 nominations for membership. Ultimately, the working group consisted of 25 members; including one member from ASRAC and one DOE representative (see Appendix A). The working group met sixteen times. The meetings were held on May 5, May 6, May 18, May 19, June 3, June 4, June 22, June 23, July 21, July 22, August 4, August 5, August 6, September 1, September 2, and September 3, 2015. The working group successfully reached consensus on certain aspects related to scope, test procedures, metric, and aspects of the energy conservation standards related to certification. This document includes the working group's recommendations to ASRAC on the energy conservation standards and test procedure and metric-related recommendations. Appendix E includes items where the working group did not reach consensus.

Scope

Fan Categories "in"
Recommendation 1.

- The scope of the test procedure and energy conservation standards recommended as part of this Working Group will include the following categories of fans: (1) axial cylindrical housed; (2) panel; (3) centrifugal housed, excluding inline and radial; (4) centrifugal unhoused, excluding inline and radial; (5) inline and mixed-flow; (6) radial housed; and (7) power roof ventilators.
- Equipment classes are discussed under Recommendation 30.

Vote results: Consensus (24 yes - 0 no – 0 abstention – 1 absent) on 7/21/2015

Fan and Impeller Categories "out"
Recommendation 2.

- The scope of the test procedure and energy conservation standards recommended as part of this Working Group will exclude the following:
  - (1) Fans of following categories, either standalone or embedded in larger pieces of equipment:
    - Radial housed unshrouded fans with diameter less than 30 inches or a blade width of less than 3 inches;
    - Safety fans as defined in Appendix D.
    - Circulating fans;
    - Induced flow fans;
    - Jet fans;
    - Cross flow fans[1]; as well as

  - (2) Fans embedded in:
    - Regulated Central Air Conditioners and Heat Pumps (Single-Phase, <65,000 Btu/h)
    - Regulated Commercial Air Conditioners and Heat Pumps that are Three-phase and less than <65,000 Btu/h (Air-Cooled)
    - Regulated Consumer Furnaces
    - Transport refrigeration (i.e., Trailer refrigeration, Self-powered truck refrigeration, Vehicle-powered truck refrigeration, Marine/Rail container refrigerant), and fans exclusively powered by internal combustion engines;
    - Vacuums
    - Fans exclusively embedded in Heat Rejection Equipment (as characterized by the Cooling Tower Institute) as:
      - Heat Rejection Equipment is defined as follows:
        - Packaged evaporative open circuit cooling towers: a device which rejects heat to the atmosphere though the direct cooling of a water stream to a lower temperature by partial evaporation.
        - Evaporative field erected open circuit cooling tower: a structure which rejects heat to the atmosphere though the direct cooling of a water stream to a lower temperature by partial evaporation.
        - Packaged evaporative closed circuit cooling towers: a device which rejects heat to the atmosphere though the indirect cooling of a process fluid stream in an internal coil to a lower temperature by partial evaporation of an external recirculating water flow.
        - Evaporative field erected closed circuit cooling tower: a structure which rejects heat to the atmosphere though the

[1] WG to provide clarification of cross flow fan exclusion.

indirect cooling of a process fluid stream to a lower temperature by partial evaporation of an external recirculating water flow.
    - Packaged evaporative condensers: a device which rejects heat to the atmosphere though the indirect condensing of a refrigerant in an internal coil by partial evaporation of an external recirculating water flow.
    - Field erected evaporative condensers: a structure which rejects heat to the atmosphere though the indirect condensing of a refrigerant in an internal coil by partial evaporation of an external recirculating water flow.
    - Packaged air cooled (dry) coolers: a device which rejects heat to the atmosphere from a fluid, either liquid, gas or mixture thereof, flowing through an air-cooled internal coil.
    - Field erected air cooled (dry) coolers: a structure which rejects heat to the atmosphere from a fluid, either liquid, gas or mixture thereof, flowing through an air-cooled internal coil.
    - Air cooled steam condensers: a device for rejecting heat to the atmosphere through the indirect condensing of steam inside air-cooled finned tubes.
    - Hybrid (water saving) versions of all of the above listed equipment that contain both evaporative and air cooled heat exchange sections.
  - If DOE can find a regulatory mechanism within its legal requirements, fans embedded in air curtains shall be excluded from the test procedures and standards established by this term sheet.

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 08/06/2015, edited on 09/02/2015

Supply and Condenser Fans Embedded in Regulated Equipment, Where the DOE Metric Captures the Energy Use of Such Fans
Recommendation 3.

- For a supply or condenser fan that is embedded in a regulated piece of equipment listed in Appendix B (i.e., select equipment for which the DOE metric captures the energy use of the supply fans and condenser fans):
  - (1) if the fan is embedded solely in regulated equipment listed in Appendix B, the fan is exempt from the test procedure and energy conservation standards recommended as part of this Working Group.
  - (2) if the fan is also embedded in equipment not listed in Appendix B or as a standalone fan, that fan is subject to the test procedure and energy conservation standards recommended as part of this Working Group.

- The fans embedded in regulated equipment as listed in Appendix B will not be considered for any additional test procedures, certifications, standards or enforcement as part of the fans and blowers rulemaking.
- The working group recommends that the metric and test procedures for regulated equipment for which the DOE metric partially includes the energy use of supply and condenser fans, be considered for modifications during their next round of rulemaking to include the full supply and condenser fan energy use in a modified metric. DOE will initiate the test procedure rulemakings early (best efforts for at least one year). As part of each of these rulemakings, DOE will consider part-load performance and operating points.
- As part of implementing this term sheet, DOE will propose a way to distinguish fans falling under (1) (e.g. permanent marking/labeling/listing).

Vote results: Consensus (18 yes – 0 no – 5 abstention – 1 absent) on 09/03/2015

Fans Embedded in Non-Regulated Equipment, and/ or Embedded in Regulated Equipment Other Than Listed in Appendix B, and/or Any Fans That Are Not Supply and Condenser Fans in Regulated Equipment listed in Appendix B
Recommendation 4.

- If DOE can find a way to enforce this regulatory approach within its statutory framework, the approach for fans embedded in non-DOE regulated equipment, and/or embedded in regulated equipment other than listed in Appendix B, and/or fans that are not supply or condenser fans in DOE-regulated equipment listed in Appendix B will be as follows:
    - The fan will be certified over its standalone operating range;
    - The test procedure will be as discussed in Recommendation 8.;
    - The first manufacturer of a testable configuration will be responsible for certifying the standalone fan performance to DOE.
    - If a manufacturer purchases such a fan in a standalone configuration, that manufacturer must ensure that the design operating range (or design point) of the embedded fan is within the certified operating range of the standalone fan, and disclose the design operating range (or design point) of the embedded fan to the end-user.

Vote results: Consensus (19 yes - 3 no – 1 abstention – 1 absent) on 09/03/2015
Members voting no: UTC Carrier, Trane/IR, Daikin/Goodman

Scope Refinement
Recommendation 5.

- The scope of the test procedure and energy conservation standards recommended as part of this Working Group will only apply to the fan operating points with the following characteristics:
    - Fan shaft power equal or greater than 1 BHP fan shaft power; and
    - Fan airpower equal or less than 150 HP (static airpower for unducted fans, total airpower for ducted fans, see Appendix C).

Vote results: Consensus (23 yes – 0 no – 1 abstention – 1 absent) on 7/22/2015

Test Procedure and Metric

Metric
Recommendation 6.

- The metric used in the regulation will be the fan electrical input power (FEP) and the fan energy index (FEI) will be allowed for representation. The FEI will be calculated using the FEP of a fan that exactly meets the standard (FEP_{STD}) which will be fixed in time to the first level established by the regulation. Both the FEP and the FEI will be represented values determined according to the DOE test procedure and sampling plan and certified to DOE.

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 07/21/2015

Testing Standalone Fans (Non-Embedded Fans)
Recommendation 7.

- The fan test procedure should generally be based on AMCA 210 (latest version available at the time of publication) for determining bare-shaft fan performance and performance of non-embedded fans. The following installation types will be used for each fan category: (1) axial cylindrical housed (D); (2) panel (A); (3) centrifugal housed, excluding inline and radial (B); (4) centrifugal unhoused, excluding inline and radial (A); (5) Inline and mixed-flow (B); (6) Radial housed (D); and (7) Power Roof Ventilators (A).
- The testable configuration for each equipment class of non-embedded fans shall be defined in the test procedure and include, at a minimum and where appropriate, the following basic parts: an impeller, a shaft, bearings, and a structure or housing.

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 08/04/2015, edited on 09/02/2015

Recommendation 8.

Testing Embedded Fans Outside of Equipment

- Testing of embedded fans will be performed outside of the equipment in a standalone fan testable configuration. If necessary, non-impeller components of the fan that are geometrically similar to the ones used by the fan as embedded in the larger piece of equipment will be used to complete the fan testable configuration.

Vote results: Consensus (22 yes – 0 no – 0 abstention – 3 absent) on 09/02/15

Direct Measurement and Calculation-based Method for Non-Embedded Fans
Recommendation 9.

- The primary focus of the test procedure rulemaking will be to represent the fan electrical input power (FEP). The FEP shall be determined either through:
    - direct measurement of the fan electrical input power (not applicable to bare-shaft fans), and/or
    - measurement of the fan's shaft input power and the combination of default values to be incorporated in the FEP for bare-shaft fans, fans sold with regulated motors[2], fans sold with AO motors[3], and fans sold with regulated motors and dynamic continuous controls[4].

  The default values to use in the calculation of the FEP shall be included in the notice of proposed rule for the test procedure. The test procedure will also specify the calculation of the FEI.

Vote results:

Bare shaft fans: Consensus (23 yes – 1 no – 0 abstention – 1 absent) on 08/04/2015
Bare shaft fans + Motor: Consensus (23 yes – 1 no – 0 abstention – 1 absent) on 08/04/2015
Bare shaft fans + Motor + controls: Consensus (23 yes – 1 no – 0 abstention – 1 absent) on 08/04/2015
Member voting no (all three votes): Morrison Products

Horsepower of the Default Motor used in FEP Calculation for Bare-shaft Fans
Recommendation 10.

- In this rulemaking, the horsepower of the default motor will be the horsepower listed in table 5 of 10 CFR 431.25 that is equal to either: 120 percent of the fan shaft input power at a given operating point, or equal to the next highest horsepower greater than 120 percent of the fan shaft input power at a given operating point.

Vote results: Consensus (22 yes – 1 no – 0 abstention – 2 absent) on 09/02/15
Member voting no: Morrison Products

Default Values for Motor Full load Efficiency used in FEP Calculation for Bare-shaft Fans
Recommendation 11.

- In this rulemaking, at a given motor horsepower the full load efficiency of the default motor used in the FEP calculation for a bare-shaft fan will be based on the minimum of the motor full load nominal efficiency from table 5 of 10 CFR 431.25 for four pole motors and across all enclosures.

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[2] Regulated under 10 CFR 431.25
[3] Air-Over (AO) motor which otherwise meet all nine characteristics from 10 CFR 431.25(f)
[4] Variable speed controls or dynamic continuous control: any device that adjusts the speed of the fan continuously over the fan's operating speed range in response to incremental changes in the required fan output airflow during its operation

Vote results: Consensus (22 yes – 1 no – 1 abstention – 1 absent) on 09/02/2015,
Member voting no: Morrison Products

Default Values for Motor Full load Efficiency used in FEP Calculation for Fans Sold with Regulated Motors

Recommendation 12.

- In this rulemaking, the full load efficiency of the default motor used in the FEP calculation for a fan sold with a regulated motor[5] will be based on the motor full load nominal efficiency from table 5 of 10 CFR 431.25 for the motor horsepower and pole configuration identical to that of the fan's motor.

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 08/04/2015

Default Values for Motor Full load Efficiency used in FEP Calculation for Fans Sold with Air Over Motors (AO)

Recommendation 13.

- In this rulemaking, the full load efficiency of the default motor used in the FEP calculation for a fan sold with a AO motor[6] will be based on the motor full load nominal efficiency from table 5 of 10 CFR 431.25, and by using the full load efficiency corresponding to the following number of NEMA bands below the values in table 5 of 10 CFR 431.25.

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[5] Regulated under 10 CFR 431.25

[6] Air-Over (AO) motor which otherwise meet all nine characteristics from 10 CFR 431.25(f)

AO Electric Motor Full Load Efficiency (NEMA bands below Table 5 of 10 CFR 431.25 for motors)				
Motor Horsepower	Pole configurations			
	2	4	6	8
1	3	7	7	3
1.5	3	7	7	3
2	3	7	7	3
3	3	7	7	3
5	3	7	7	3
7.5	3	5	2	1
10	3	5	2	1
15	3	5	2	1
20	3	5	2	1
25	3	5	2	1
30	3	3	2	3
40	3	3	2	3
50	3	3	2	3
60	3	3	2	3
75	3	3	2	3
100	3	3	2	3
125	3	3	2	3
150	3	3	2	3
200	3	3	2	3
250	3	3	2	3

Appendix E provides the corresponding motor full load efficiencies for enclosed and open AO motors.

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 08/04/2015, edited on 09/02/2015.

Default Values for Transmission Efficiency
Recommendation 14.

- In this rulemaking, the efficiency of the default transmissions used in the FEP calculation will be based on the medium efficiency curve in AMCA 203 using the following equation: 0.96 * (1-exp(-275*BHP)^0.19); where BHP is the fan shaft input power in horsepower.
- In this rulemaking, the efficiency of the default transmissions used in the FEP calculation for direct driven fans is 1.

Vote results: Consensus (23 yes – 1 no – 0 abstention – 1 absent) on 08/04/2015
Member voting no: ebm-papst Inc.

Default Control Losses Determination
Recommendation 15.

- In this rulemaking, the default part load losses of the motor and controls at a given operating point i shall be determined by multiplying the motor default full load losses by the following polynomial equation:

$$z_i = (\mathrm{a} \times \mathrm{x_i}^2 + \mathrm{b} \times \mathrm{x_i} + \mathrm{c})$$

a,b,c = coefficients based on the horsepower of the motor with which the fan is being rated,

Motor Horsepower (hp)	Coefficients for Motor and Control Part Load Loss Factor (z_i)		
	A	**b**	**c**
≤5	-0.4658	1.4965	0.5303
>5 and ≤20	-1.3198	2.9551	0.1052
>20 and ≤50	-1.5122	3.0777	0.1847
>50	-0.8914	2.8846	0.2625

Where x_i is the load fraction for the motor at operating point i (percent), calculated as follows:

$$x_i = \frac{BHP_i}{\eta_{T,i}} \times \frac{1}{MotorHP}$$

Where:

BHP_i = shaft input power (hp) at operating point i;
MotorHP = the motor horsepower (hp) as determined in accordance with Recommendation 10., Recommendation 11., Recommendation 12., and Recommendation 13. ; and
$\eta_{T,i}$ = default transmission efficiency at rating point i (percent) as determined in accordance with Recommendation 14.

Note: this equation is valid up to a limit to be validated (e.g. x_i =1.2). Above that limit, the losses shall be capped.

Vote results: Consensus (22 yes – 0 no – 0 abstention – 3 absent) on 09/02/15

Credit for Fans with Controls
Recommendation 16.

At a given operating point P_1 (pressure) and Q_1 (flow):

- FEP_{STD} is calculated based on a default bare-shaft fan + motor + belt configuration, includes belt losses at P_1 and Q_1
- $FEP_{fan\_control}$ includes control losses at P_1 and Q_1 and a credit for use of dynamic continuous controls.

Vote results: Consensus (20 yes – 0 no – 2 abstention – 3 absent) on 09/02/2015

Note: During the September 2, 2015 meeting, the following was discussed: :

- AMCA suggested a credit that would be equal to the controls losses. Others recommended a credit greater than the control losses.
- The European example of the credit as a function of electrical input power was also mentioned.

Test Speed and Use of the fan laws
Recommendation 17.

- For a bare shaft fan with a given diameter, the fan shall be tested at:
  - A single speed if using the fan laws to determine the performance of the fan at other speeds, defined as follows:
    - for fans other than fans sold with a multispeed motor and direct driven: the fan will be tested at its average speed of operation. The average speed of operation is the average of the maximum and the minimum speed for which the fan is offered for sale,
    - for fans sold with a multispeed motor and direct driven: the fan will be tested at its average speed of operation if available for operation, or at its next lowest operating speed lower than the average speed of operation. The average speed of operation is the average of the maximum and the minimum speed for which the fan is offered for sale.
  - at each speed offered for sale if not using the fan laws to determine the performance of the fan at other speeds.

  At each tested speed, no less than a number of points to be determined by DOE that are equally spaced (flow) determinations shall be made from shut off to free delivery. Operating conditions and performance between determinations shall be based on a moving polynomial defined by DOE. Working Group members will submit a recommendation for the number of points to the docket that will be considered by DOE when determining the number of points.

Vote results: Consensus (21 yes – 0 no – 1 abstention – 3 absent) on 09/02/2015

<u>Energy Conservation Standards</u>

Calculation of the Standard Level FEP for standalone fans (FEP_{STD})

Recommendation 18.

- The maximum allowable fan electrical input power (FEP_{STD}) at each declared operating point i (Q_i, P_i) shall include: the fan shaft input power corresponding to a fan with a fan efficiency equal to $\eta_{STD,i}$; belt losses as calculated using the default belt efficiency curve ($\eta_{T,i}$); and default motor losses ($L_{M,i}$) at that operating point.

Where:

$FEP_{STD,i}$ = maximum allowable fan electrical input power kW at operating point i;
Q_i = flow (cfm) at operating point i;
P_i = total pressure for ducted fans / static pressure for unducted fans (in.wg) at operating point i;
$\eta_{STD,i}$ = minimum fan total efficiency for ducted fans / minimum fan static efficiency for unducted fans (%) at operating point i as calculated in accordance with Recommendation 19;
$\eta_{T,i}$= default transmission efficiency (percent) at operating point i as calculated in accordance with Recommendation 14;
$L_{M,i}$ =default electric motor losses (hp) at operating point i as calculated based on Recommendation 11.

Note: On 08/13/2015 the WG discussed the possibility an alternative calculation of the FEP_{STD} for fans sold with motors and controls.
On 9/2/15 the WG resolved calculation of FEP_{STD} for fans with motors and controls (See Recommendation 16.)

Vote results: Consensus (24 yes - 1 no – 0 abstention – 0 absent) on 08/05/2015
Member voting no: Morrison Products

Fan Total and Static Efficiency Equation
Recommendation 19.

- The minimum fan total efficiency for ducted fans / minimum fan static efficiency for unducted fans (%) at operating point i shall be calculated in accordance the following equation:

$$\eta_{STD,i} = \eta_{target} \frac{Q_i \times P_i}{(Q_i + Q_0)(P_i + P_0)}$$

Where:

$\eta_{STD,i}$ = minimum fan total efficiency for ducted fans / minimum fan static efficiency for unducted fans (percent) at operating point i ;
Q_i= flow (cfm) at operating point i;
P_i = total pressure for ducted fans, static pressure for unducted fans (in.wg) at operating point i;
Q_0 = flow constant, as established in Recommendation 20;
P_0 = pressure constant, as established in Recommendation 21; and
η_{target}= constant used to establish the efficiency level to be set by this rulemaking.

Note: all pressures refer to standard air density.

Vote results: Consensus (25 yes – 0 no – 0 abstention – 0 absent) on 08/05/2015

Value of Q_0
Recommendation 20.

- Q_0 shall be equal to 250

Vote results: Consensus (22 yes – 0 no – 3 abstention – 0 absent) on 08/05/2015

Value of P_0
Recommendation 21.

- P_0 shall be equal to 0.4

Vote results: Consensus (22 yes – 0 no – 3 abstention – 0 absent) on 08/05/2015Represented Values

Use of the fan laws across sizes
Recommendation 22.

- The use of the fan laws will be allowed to determine the represented values of FEP and FEI for geometrically similar fans[7] as follows:
  - If a manufacturer offers geometrically similar fans at more than three impeller diameters for sale, the manufacturer shall test at least three diameters for bare shaft performance over the range and use the fan laws to calculate operating points (and performance) for any larger diameter fans within the range offered sale based on testing of the smaller geometrically similar bare shaft fans;
  - For custom fans (those offered in selection software but not in catalogues) and for those geometrically similar fans offered for sale at three or less impeller diameter, the manufacturer shall test at least one diameter for bare-shaft fan performance over the range and use the fan laws to calculate operating points (and performance) for any larger diameter fan within the range offered sale based on testing of the smaller bare shaft fan.

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 08/04/2015

Represented Values
Recommendation 23.

- When testing is used to establish the rating of a basic model, a minimum of 1 unit shall be tested and the tested result shall be no greater than the represented value ($FEP_{STD} \geq FEP_{Rating} \geq FEP_{TEST}$). When using an AEDM to establish the rating of a basic model, the value resulting from the AEDM shall be no greater than the represented value ($FEP_{STD} \geq FEP_{Rating} \geq FEP_{AEDM}$). Conservative rating will be allowed.

[7] The definition of geometrically similar will be based on AMCA 211 Annex A which states that most design dimensions shall be proportional within +/- 1 percent (with listed exceptions).

Vote results: Consensus (22 yes – 0 no – 0 abstention – 3 absent) on 09/02/2015

AEDM minimum number of models to be tested
Recommendation 24.

- The minimum number of basic models that shall be tested to validate an AEDM shall be as follows (example):
    - (1) At least 2 compliant basic models selected for testing per equipment class for which the AEDM is to be applied.
    - (2) If an AEDM is used to rate models that simulate the wire-to-air test then the models used to validate the AEDM pursuant to bullet (1) should be tested with the full wire-to-air.

Vote results: Consensus (21 yes – 0 no – 1 abstention – 3 absent) on 09/02/2015

Validation of an AEDM
Recommendation 25.

- The predicted FEP using the AEDM may not be more than 5% less than the FEP determined from the test according to the DOE test procedure for the basic models used to validate an AEDM.

Vote results: Consensus (20 yes – 0 no – 2 abstention – 3 absent) on 09/02/2015

Certification
Recommendation 26.

- DOE will investigate whether manufacturers can be allowed to use selection software in lieu of certification to DOE. Representations would be allowed to be made for any model available in the selection software. The selection software would be available on the DOE website.
- If a manufacturer does not have selection software or DOE cannot find a viable way to administer its certification by accepting selection software, the manufacturer would have to submit the certification of the operating range for each individual model distributed in commerce to DOE. (either in tabular format or equations/curves)

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 09/03/2015

Recommendation 27.

- Manufacturers would be required to submit the general information in 429.12, including the manufacturer/model which would encompass the bare shaft fan/impeller manufacturer and model number, the motor manufacturer and model number (where applicable), and controls/driver manufacturer and model number (where applicable).

- At least the following public equipment specific information will required to be certified for each manufacturer-declared operating point: fan operating flow (CFM), fan operating pressure (in.wg. static and total for unducted and ducted fans, respectively), FEP (kW), FEI, fan operating speed (RPM) and fan shaft input power (HP) for fans using the calculation-based/default-value method and relying on the shaft input power measurement.
- At least the following public equipment specific information will required to be certified by each fan manufacturer: fan maximum operating speed.
- At least the following non-public equipment specific information will be provided: rating method (e.g. wire-to-air/direct measurement test or calculation-based/default value method, use of the fan laws or not).

Vote results: Consensus (24 yes – 0 no – 0 abstention – 1 absent) on 09/03/2015

This term sheet has been approved by the ASRAC Fans and Blowers working group by consensus vote on 09/03/2014 (18 yes – 2 no – 0 abstention - 5 absent). It can now be passed to ASRAC for consideration. It should be noted that the exact language in this term sheet may be modified when implemented by DOE as regulatory text, but the intent should remain the unchanged.

Members voting no on the Entire Term sheet: Trane/IR and AHRI

Appendix A—Members

U.S. Department of Energy—ASRAC
Fans and Blowers Working Group

Ashley Armstrong	U.S. Department of Energy
Mark Bublitz	The New York Blower Company
Larry Burdick	SPX Cooling Technologies / CTI
Duane Daddis	United Technologies/Carrier
Steve Dikeman	AcoustiFLO LLC
Gary Fernstrom	Pacific Gas & Electric Company, San Diego Gas & Electric Company, Southern California Edison, and Southern California Gas Company
Mark Fly	AAON, Inc.
Dan Hartlein	Twin City Companies, Ltd
Armin Hauer	ebm-papst Inc
Nicholas Howe	Carnes Company
Diane Jakobs	Rheem Manufacturing Company
David Johnson	Berner International Corp
Joanna Mauer	Appliance Standards Awareness Project
Paul Lin	Regal Beloit Corporation
Donald McNeil	Buffalo Air Handling Company
Laura Petrillo-Groh	Air-conditioning, Heating, and Refrigeration Institute (AHRI)
Aniruddh Roy	Daikin/Goodman
Geoff Sheard	AGS Consulting LLC
William Smiley	Smiley Engineering LLC representing Trane/IR
Wade Smith	Air Movement and Control Association International
Louis Starr	Northwest Energy Efficiency Alliance
Gregory Wagner	Morrison Products
Meg Waltner	Natural Resources Defense Council
Stephen R. Wiggins	Newcomb & Boyd
Michael Wolf	Greenheck

Appendix B—Regulated Equipment for which the DOE metric accounts for the energy use of the supply and condenser fans.

- Air-cooled commercial AC/HP, 5.5 - 63.5 tons (CUAC/CUHP)
- Water-cooled and Evaporatively-cooled AC and Water-source HP
- Commercial Single Package Vertical ACs and Single Package Vertical HPs
- PTACs and PTHPs
- Computer Room ACs
- Commercial VRFs

Appendix C—Ducted and Unducted Fans

- The following fan types are considered Ducted for the purposes of this test procedure and regulation:
    - Axial cylindrical housed
    - Centrifugal housed, excluding inline and radial
    - Inline and mixed flow
    - Radial housed
- The following fan types are considered Unducted for the purposes of this test procedure and regulation:
    - Panel
    - Centrifugal unhoused, excluding inline and radial
    - Power roof ventilators

Appendix D—Definitions

Safety Fan Definition

The definitions presented in this appendix are subject to potential edits necessary to accomplish the same intent.

Safety fan:

The current working definition is based on the European definition:

Fans designed for use in applications requiring extra safety measures, such as:

a) those designed to operate in potentially explosive atmospheres (ATEX fans);

b) those designed for emergency use only, at short-time duty, with regard to fire safety requirements (e.g. smoke extraction fans, emergency reversible tunnel fans);

c) those designed specifically to operate where the temperature of gases being moved exceed 200ºF; or

d) those designed for use in toxic, highly corrosive, or flammable environments with abrasive substances (e.g. NQ-1).

Appendix E—Air Over Motor (AO) Full Load efficiency

Motor Horsepower	Default Open AO Full Load Efficiency				Default TEAO Full Load Efficiency			
	Pole configurations				Pole configurations			
	2	4	6	8	2	4	6	8
1	72	75.5	70	70	72	75.5	72	70
1.5	80	77	77	72	80	77	78.5	74
2	81.5	77	78.5	82.5	81.5	77	80	80
3	81.5	81.5	80	84	82.5	81.5	81.5	81.5
5	82.5	81.5	81.5	85.5	85.5	81.5	81.5	82.5
7.5	85.5	86.5	88.5	88.5	86.5	87.5	89.5	85.5
10	86.5	87.5	90.2	89.5	87.5	87.5	89.5	88.5
15	87.5	89.5	90.2	89.5	88.5	88.5	90.2	88.5
20	88.5	89.5	91	90.2	88.5	89.5	90.2	89.5
25	89.5	90.2	91.7	90.2	89.5	90.2	91.7	89.5
30	89.5	92.4	92.4	89.5	89.5	91.7	91.7	89.5
40	90.2	92.4	93	89.5	90.2	92.4	93	89.5
50	91	93	93	90.2	91	93	93	90.2
60	91.7	93.6	93.6	91	91.7	93.6	93.6	90.2
75	91.7	93.6	93.6	92.4	91.7	94.1	93.6	91.7
100	91.7	94.1	94.1	92.4	92.4	94.1	94.1	91.7
125	92.4	94.1	94.1	92.4	93.6	94.1	94.1	92.4
150	92.4	94.5	94.5	92.4	93.6	94.5	95	92.4
200	93.6	94.5	94.5	92.4	94.1	95	95	93
250	93.6	94.5	95	93.6	94.5	95	95	93.6

Appendix F— Additional Recommendations Discussed For Which No Consensus Was Reached

Replacement Fans
Recommendation 28.

- The Working Group agrees to exclude a method of addressing replacement fans from the term sheet and leave for DOE to resolve. The record reflects different options and opinions on replacement fans. (09/02/2015)

Default Motor Part Load Losses Determination
Recommendation 29.

- In this rulemaking, the part load motor losses at a given operating point i shall be determined by multiplying the default full load losses by the following polynomial equation:

$$y_i = -0.4508 * x_i^3 + 1.2399 * x_i^2 - 0.4301 * x_i + 0.641$$

Where x_i is the load fraction for the motor at operating point i (percent), calculated as

follows:

$$x_i = \frac{BHP_i}{\eta_{T,i}} \times \frac{1}{MotorHP}$$

Where:

BHP_i = shaft input power (hp) at operating point i;
MotorHP = the motor horsepower (hp) as determined in accordance with Recommendation 10., Recommendation 11., Recommendation 12., and Recommendation 13. ; and
$\eta_{T,i}$ = default transmission efficiency at rating point i (percent) as determined in accordance with Recommendation 14.

Note: this equation is valid up to a limit to be validated (e.g. x_i =1.2). Above that limit, the losses shall be capped.

On 9/02/15: WG decided to leave this issue to DOE to determine whether impacts justify inclusion.

Equipment Classes
Recommendation 30.

- The regulation shall use the following equipment classes:

- Axial cylindrical housed fans
- Panel fans
- Housed centrifugal fans, excluding inline fans and radial fans
- Unhoused centrifugal fans, excluding radial fans
- Inline fans and mixed flow fans
- Housed radial, shrouded impeller fans
- Power roof ventilators

Note: Supply power roof ventilators are included in the housed centrifugal fan equipment class

Vote results: No Consensus (14 yes – 7 no – 1 abstention – 3 absent) on 09/02/15
Note: No votes believe forward-curved fans should have their own equipment class

Labeling Minimum requirements

Recommendation 31.

- The following information shall be present on the label (design point known):
  - Model number
  - Serial number or Date of manufacturing
  - Design Flow (cfm), Design Pressure (wg.)(static/total for unducted/ducted fans)
  - Associated FEI
  - Maximum RPM of the fan (as declared by manufacturer)
  - Link to DOE website (URL) to the complete performance map of the fan

Vote results: No Consensus (16 yes – 7 no – 1 abstention – 1 absent) on 09/03/2015

Recommendation 32.

- The following information shall be present on the label (design point unknown):
  - Model number
  - Serial number or Date of manufacturing
  - Max RPM of the fan (as declared by the manufacturer)
  - Link to DOE website (URL) to certified operating range of the fan

Vote results: No Consensus (16 yes – 6 no – 2 abstention – 1 absent) on 09/03/2015