STATE OF WASHINGTON

STATE BUILDING CODE COUNCIL

Washington State Energy Code Development

Standard Energy Code Proposal Form

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Code being amended: ☒ Commercial Provisions ☐ Residential Provisions

Code Section # C202, C403.8.3, C403.8.6

Brief Description: *This proposal seeks to improve the FEI level requirement based on fan type and application for non-embedded fans.*

Proposed code change text: (Copy the existing text from the Integrated Draft, linked above, and then use underline for new text and ~~strikeout~~ for text to be deleted.)

C202 General Definitions

Embedded fan. A fan that is set or fixed inside a piece of equipment whose purposes exceeds that of a fan or is different than that of a stand-alone fan. The equipment may have safety or energy efficiency requirements of its own. Examples of embedded fans include supply fans that are part of air-handling units, condenser fans in heat rejection equipment, fans as part of air curtain units, or forced draft combustion blowers in boilers or furnaces.

Air circulating fan. A fan that has no provisions for connection to ducting or separation of the fan inlet from its outlet using a pressure boundary, operates against zero external static pressure loss, and is not a jet fan or recirculating fan.

Equivalent diameter. The diameter of a circle with the same area as another geometric shape. For a rectangular cross section with width 'a' and heigh 'b', the equivalent diameter is given as $D = (4ab/\Pi)^{0.5}$

Safety fan. A fan whose operation meets one or more of the following:

(1) a reversible axial fan in cylindrical housing that is designed and marketed for use in ducted tunnel ventilation that will reverse operations under an emergency ventilation condition;

(2) a fan for use in explosive atmospheres tested and marked according to ISO 80079-36:2016, Explosive atmospheres -- Part 36: Non-electrical equipment for explosive atmospheres -- Basic method and requirements;

(3) an electric-motor-driven-Positive Pressure Ventilator as defined AMCA 240;

(4) a fan complying with ANSI/UL 705 Standard for Safety for Power Ventilators and listed as "Power Ventilators for Smoke Control Systems";

(5) a laboratory exhaust fan designed and marketed specifically for exhausting contaminated air vertically away from a building using high-velocity discharge.

Induced-flow fan. A type of laboratory exhaust fan with a nozzle and windband; the fan's outlet airflow is greater than the inlet airflow due to induced airflow. All airflow entering the inlet exits through the nozzle. Airflow exiting the windband includes the nozzle airflow as well as the induced airflow.

C403.8.3 Fan efficiency. Each fan and fan array shall have a fan energy index (FEI) of not less than ~~1.00~~ the values listed in Table C403.8.3 at the design point of operation, as determined in accordance with ~~AMCA 208~~ DOE 10 CFR 431 Appendix A to Subpart J by an approved, independent testing laboratory and labeled by the manufacturer. ~~Each fan and fan array used for a variable-air volume system shall have an FEI of not less than 0.95 at the design point of operation as determined in accordance with AMCA 208 by an approved, independent testing laboratory and labeled by the manufacturer.~~ The FEI for fan arrays shall be calculated in accordance with AMCA 208 Annex C.

Exceptions: The following fans are not required to ~~have a fan energy index~~ meet the FEI values listed in Table C403.8.3:

1. Fans that are not *embedded fans* with ~~motor nameplate horsepower~~ shaft input power of less than 1.0 hp (0.75 kW), or with ~~a nameplate~~ electrical input power of less than 0.89 kW, or with air power greater than 150 horsepower.
2. *Embedded fans* that have a motor nameplate horsepower of 5 hp (3.7 kW) or less or with a fan system electrical input power of 4.1 kW or less. *Embedded fans* greater than 5 hp (3.7kW) or with a fan system electrical input power greater than 4.1 kW shall have a FEI of not less than 1.00.
3. Multiple fans operated in series or parallel as the functional equivalent of a single fan that have a combined motor nameplate horsepower of 5 hp (3.7 kW) or less or with a fan system electrical input power of 4.1 kW or less. Multiple fans operated in series or parallel as the functional equivalent of a single fan that have a combined motor nameplate horsepower greater than 5 hp (3.7kW) or with a fan system electrical input power greater than 4.1 kW shall have a FEI of not less than 1.00.
4. Fans that are part of equipment covered under Section C403.3.2.
5. Fans included in an equipment package certified by an *approved agency* for air or energy performance.
6. *Ceiling fans*, *air circulation fans*, and air curtains.
7. Fans used for moving gases at temperatures above 482°F (250°C).
8. ~~Fans used for operation in explosive atmospheres.~~
9. ~~Reversible fans used for tunnel ventilation.~~
10. ~~Fans that are intended to operate only during emergency conditions.~~
11. *8.* Fans outside the scope of AMCA 208.

9. *Safety Fans, Induced-flow fans* and jet fans.

Table C403.8.3 – FEI Requirements by Fan Type

Fan Type	**Fan Energy Index (FEI) Without Motor Controller**	**Fan Energy Index (FEI) With Motor Controller[a]**
Axial Inline	1.10	1.05
Axial Panel	1.00	0.95
Axial Power Roof Ventilator	0.85	0.81
Centrifugal Housed	1.15	1.09
Centrifugal Unhoused	1.05	1.00
Centrifugal Inline[b]	1.07	1.02
Radial Housed	1.00	0.95
Centrifugal Power Roof Ventilator - Exhaust	1.00	0.95
Centrifugal Power Roof Ventilator - Supply	1.00	0.95

a. A 0.95 correction factor is applied to the FEI for fans equipped with motor controllers.

b. FEI levels in the table apply to Tubular Centrifugal Inline fans. Square Duct Centrifugal inline fans without a motor controller shall have an FEI of not less than 1.00, or an FEI of not less than 0.95 if equipped with a motor controller.

C403.8.6 Large-diameter ceiling fans and air circulating fans. Where provided, *large-diameter ceiling fans* and *air circulating fans* shall meet the efficiency requirements of Table C403.8.6. *Large-diameter ceiling fans* shall also meet the requirements of Section C403.8.6.1.

TABLE C403.8.6.1
CEILING AND CIRCULATING FAN EFFICIENCY REQUIREMENTS[a]

Equipment Type	Category	Minimum Efficiency[b,c]	Test Procedure
Large diameter ceiling fan	Blade span ≥ 84.5 in	CFEI ≥ 1.00 at high (maximum) speed; and CFEI ≥ 1.31 at 40% of high speed or the nearest speed that is not less than 40% of high speed	10 CFR 430, Appendix U
Air Circulating Fan	≥ 200 W input power	~~$Effcirc \geq \frac{16DD^5 + 200DD^4}{QQ^2}$~~ $Eff_{circ} \geq \frac{16D^5 + 200D^4}{Q^2}$	10 CFR 431, Appendix B to Subpart J

~~a. The minimum efficiency requirements at both high speed and 40% of maximum speed shall be met or exceeded to comply with this code.~~

~~b. Ceiling fans are regulated as consumer products by 10 CFR 430.~~

~~c. Chapter 6 contains a complete specification of the referenced test procedure, including the referenced year version of the test procedure.~~

a. Chapter 6 contains a complete specification of the referenced test procedure, including the referenced year version of the test procedure.

b. Eff_{circ} is the efficacy for *air circulating fans* (CFM/W). D is the impeller diameter for unhoused fans, and the lesser of impeller diameter and *equivalent diameter* for housed fans (inches). Q is the *air circulating fan* airflow rate, determined by the referenced test procedure at the maximum fan speed (cfm).

Purpose of code change:

A federal test procedure for FEI has been in place for several years and was brought about through industry support working with DOE. However, a rulemaking was never developed to accompany the test procedure which is referenced by all energy codes currently (IECC, Title 24, WSEC, 90.1). Additionally, as of 1/13/25, the DOE has rescinded the NOPR and does not plan to institute a rulemaking in the future.

Fans in the market can achieve the current FEI levels across all manufacturers, and the 2021 WSEC not currently push the market to adopt better fan design. The proposed levels here closely match those put forth by AMCA in their comments to the DOE with regards to their 2024 NOPR. This code change would increase the efficiency of non-embedded fans throughout the commercial market during the selection phase, leading to less energy costs for building owners and tenants.

In addition, the term "embedded fan" was shown to be a defined term in the 2021 WSEC but no definition was provided. Therefore, a definition has been proposed to clarify the distinction between embedded and non-embedded fans.

Your amendment must meet one of the following criteria. Select at least one:

- ☐ Addresses a critical life/safety need.
- ☐ The amendment clarifies the intent or application of the code.
- ☐ Addresses a specific state policy or statute. (Note that energy conservation is a state policy)
- ☒ Consistency with state or federal regulations.
- ☐ Addresses a unique character of the state.
- ☐ Corrects errors and omissions.

Check the building types that would be impacted by your code change:

- ☐ Single family/duplex/townhome
- ☒ Multi-family 1 – 3 stories
- ☒ Multi-family 4 + stories
- ☒ Commercial / Retail
- ☒ Institutional
- ☒ Industrial

Your name Nicholas O'Neil

Your organization Energy 350

Other contact name Kevin Rose, NEEA

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Economic Impact Data Sheet

Is there an economic impact: ☒ Yes ☐ No

Briefly summarize your proposal's primary economic impacts and benefits to building owners, tenants, and businesses. If you answered "No" above, explain your reasoning.

The primary economic impact of this proposal is the balance between the benefits of customers' avoided energy costs, and the upfront costs of increased equipment prices and manufacturer market adjustments. This fan efficiency proposal considers the manufacturing cost and installed cost of replacing existing or installing new stand-alone fans and blowers (i.e. non-embedded.) These impacts are derived from the US Department of Energy's (DOE's) 10 CFR Parts 429 and 431 Notice of Proposed Rulemaking (NOPR) published on January 19, 2024 (DOE NOPR). In the NOPR (which this proposal matches with several adjustments) DOE summarizes a positive impact on manufacturers, commercial and industrial customers, and the broader economy during the 2030 to 2059 compliance period. Nationwide, the DOE estimates an incremental investment of $5.7 billion would yield more than $40 billion in operating cost savings, including avoided energy costs. It further estimates more than $30 billion in combined health and climate benefits for a combined net benefit of just over $70 billion.

The DOE's January 2024 NOPR considers nine equipment classes (as per Table C403.8.3 shown above) and estimates manufacturing costs along with derived fan sales prices and installed costs. The DOE's December 2023 technical support document shares detailed methods for how these costs are estimated, including details on forecasting electricity prices and estimating avoided energy costs. Below we summarize how we interpret these DOE outputs and adjust to fit the code proposal, along with taking into account comments from AMCA on the proposed levels and their suggested adjustments. This proposal uses these same nine DOE fan equipment classes to derive estimated energy savings and incremental construction costs.

For each equipment class, the DOE reviewed respective fans from the AMCA sales database and manufacturer fan selection software to determine FEIs for baseline and max-efficiency technology fans. Typically, these were set at respective 5th and 95th percentiles of FEIs for all available fans —with some exceptions. The DOE then chose design pathways with stepped efficiency levels, including an intermediate level corresponding with FEI = 1.0 fans. Upon reviewing the application of FEI to tan selections, we believe the design pathways are not representative of realistic designs (AMCA agrees with this statement) and therefore adjustments need to be made to better reflect actual designs compared to what the DOE proposed. Some industry stakeholders—manufacturers and industry associations, including AMCA —suggested during the rulemaking comment period that many of the DOE's proposed FEI standards for each equipment class are impracticable for higher-air-power duty points. This was especially concerning for embedded fans, though AMCA also notes many standalone fans at the proposed FEI levels would make it harder to find a viable replacement. Setting a standard at a higher level that requires manufacturers to re-tool to meet proposed levels may be appropriate for a standard, however we do not suggest this as an approach for state code. Therefore, we have proposed to remove embedded fans from the scope of the proposal, and to more closely align with AMCA's proposed FEI levels by fan equipment class rather than DOE's higher FEI levels in the NOPR.

One additional deviation from the NOPR is how to treat fans with motor controllers. The DOE proposes a set of equations to credit fans with motor control, however, many stakeholder comments on the NOPR noted unnecessary complexity in its approach. Instead, as commented by manufacturers and associations including AMCA and AHRI, we propose using a 5% credit to the minimum FEI standard for each respective fan class with motor controllers. I.e., a fan with a motor controller would need to meet 95% of its respective minimum FEI. This follows how the ASHRAE 90.1-2022 standard and California's Title 24 Energy Code applies credits for fans with motor controllers. The rationale is that fans with motor

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All questions must be answered to be considered complete. Incomplete proposals will not be accepted.

controllers can save energy by pushing lower air volumes in variable flow applications, but their design is intrinsically slightly less efficient than a single speed.

Provide your best estimate of the **construction cost** (or cost savings) of your code change proposal? (See OFM Life Cycle Cost **Analysis tool** and **Instructions;** use these **Inputs**. **Webinars on the tool can be found Here and Here**)

$0.03/square foot (For residential projects, also provide $Click here to enter text./ dwelling unit)

Show calculations here, and list sources for costs/savings, or attach backup data pages:

For each of the nine NOPR fan classes, the DOE estimates manufacturing costs from a teardown of fan equipment, including raw materials, part fabrication, and assembly, into a final product. The NOPR also assessed typical markups to generate sales prices (on average, 135% of the manufactured cost). Stakeholders commented that this approach is mostly accurate, though with some critique for specific fan types. For the purposes of a high-level economic impact assessment, these costs are sufficiently accurate. We include install cost estimates, which allow us to compare installed costs across various FEI ranges. We further calculated incremental installed cost relative to FEI=1.0 fans (above and beyond the current energy code requirement). Only one category, Axial Power Roof Ventilators are shown to have a relaxed FEI compared to what is required currently however AMCA and DOE agreed this was appropriate for this fan category. As such, costs are negative compared to an FEI 1.0 compliant fan. While the FEI values from DOE do not exactly match proposed values above, they are sufficient for determining cost increases due to higher FEI values.

Class	Proposed Efficiency level	FEI	Incremental installed cost relative to FEI 1.0 ($)
Axial Inline	EL3	1.18	$592
Axial Panel	EL3	1.24	$47
Axial Power Roof Ventilator	EL4	0.85	-$1,737
Centrifugal Housed	EL3	1.15	$24
Centrifugal Unhoused	EL3	1.23	$80
Centrifugal Inline	EL4	1.07	$651
Radial Housed	EL3	1.00	$0
Centrifugal Power Roof Ventilator - Exhaust	EL4	1.00	$0
Centrifugal Power Roof Ventilator - Supply	EL4	1.00	$0

We sourced installed costs from the NOPR technical support document (EERE-2022-BT-STD-0002-0133) Table 8.5.1-17 Average LCC and PBP Results. To obtain a $/sqft we used 2019 CBSA data for average building size (determined to be a 32,100 sqft building), an assumption of 4 standalone fans per building, and an average EUI data from the 2022 Seattle Benchmarking data. From there we relied on average costs (except Axial Power Roof Ventilators) to determine an incremental $/sqft for the LCC tool.

Provide your best estimate of the **annual energy savings** (or additional energy use) for your code change proposal?

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0.07 KWH/ square foot (or) Click here to enter text.KBTU/ square foot

Show calculations here, and list sources for energy savings estimates, or attach backup data pages

For each of the nine NOPR fan classes, the DOE estimates fan energy consumption over their lifetime, considering operating hours, load profiles, and difference in fan efficiency between the "no-new-standards case" and the projected energy use under the proposed new FEI (and corresponding efficiency level). We utilized energy savings estimates for each fan class using a baseline FEI=1.0 (as specified in the current WSEC) compared to an efficiency level close to what AMCA proposed in their comments to DOE. For more information, see the NOPR technical support document (EERE-2022-BT-STD-0002-0133), Chapter 7, and Table 7.2.11 GFBs: Average Annual Energy Use by Equipment Class for each Standards Case (kWh). As noted above, for Axial Power Roof Ventilators, savings are negative compared to an FEI 1.0 compliant fan due to the proposed level being lower. While the FEI values from DOE do not exactly match proposed values above, they are sufficient for determining savings impacts due to higher FEI values.

Class	Proposed Efficiency level	FEI	Annual energy savings relative to FEI 1.0 (kWh)	AMCA Comments on FEI levels
Axial Inline	EL3	1.18	751.0	Design pathway includes guide vanes, which may not allow for direct replacement with non-guide vane applications. Potential to improve aerodynamic design further-though at higher cost to match higher savings.
Axial Panel	EL3	1.24	748.0	AMCA recommends a lower FEI to accommodate specific applications where higher FEI fans are not practical and would result in specifying alternate equipment. E.g., axial panel exhaust fan installed in wall-opening where opening can't be modified.
Axial Power Roof Ventilator	EL4	0.85	(900.0)	ACEEE, ASAP and NRDC note that the installed cost of the Axial PRV fan seems high. Especially because most of the market's fans are available at EL4/5.
Centrifugal Housed	EL3	1.15	651.0	AMCA believes centrifugal-housed fans, as widely available in the market, are already within 15% of peak efficiency. It recommends an FEI similar to EL3.
Centrifugal Unhoused	EL3	1.23	394	For stand-alone fans only. Many fans available at higher EL3 than proposed by AMCA.
Centrifugal Inline	EL4	1.07	334	Similar to US DOE proposal, assumes mixed flow fans are able to serve all centrifugal inline fan applications. Many more fans available in the EL3 to EL4 range should make this less of an issue.
Radial Housed	EL3	1.00	-	Many more radial housed fans available at EL3.
Centrifugal Power Roof Ventilator - Exhaust	EL4	1.00	-	Most centrifugal PRV exhaust fans sold are already at this FEI level. It's also aligned with FEI=1.0.
Centrifugal Power Roof Ventilator - Supply	EL4	1.00	-	Most centrifugal PRV - Supply fans fall within the EL4 level. There are few options at EL5 or above levels.

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For more details on how the energy consumption per fan unit varies by efficiency level, see Chapter 7 of the NOPR technical support document (EERE-2022-BT-STD-0002-0133).

To obtain a kWh/sqft we used 2019 CBSA data for average building size (determined to be a 32,100 sqft building), an assumption of 4 standalone fans per building, and average EUI data from the 2022 Seattle Benchmarking data. From there we relied on average costs (except Axial Power Roof Ventilators) to determine a kWh/sqft for the LCC tool.

List any **code enforcement** time for additional plan review or inspections that your proposal will require, in hours per permit application:

This code change proposal would not increase additional plan review time as current code already requires an FEI level for these fans. This would simply require a more stringent level than is currently included.

Small Business Impact. Describe economic impacts to small businesses:

None anticipated. Fans with FEI levels in excess of 1.0 are widely available and routinely specified on permit sets currently.

Housing Affordability. Describe economic impacts on housing affordability:

N/A

Other. Describe other qualitative cost and benefits to owners, to occupants, to the public, to the environment, and to other stakeholders that have not yet been discussed:

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AMCA International
Air Movement and Control Association International, Inc.
The International Authority on Air System Components Since 1917

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March 19, 2024

Mr. Jeremy Dommu
U.S. Department of Energy
Office of Energy Efficiency and Renewable Energy
1000 Independence Avenue SW
Washington, DC 20585-0121

Via email to: FansAndBlowers2022STD0002@ee.doe.gov

Docket: EERE–2022–BT–STD–0002; RIN 1904–AF40

Dear Mr. Dommu:

On behalf of Air Movement and Control Association (AMCA) International,[1] thank you for publishing the Notice of Proposed Rulemaking (NOPR) for energy-conservation standards (energy standard) for general fans and blowers (GFB) and for air-circulating fans (ACF). AMCA greatly appreciates the amount of work DOE has expended since the rulemaking began in 2011 to reach this point.

AMCA has been a willing and generous collaborator with DOE and other stakeholders since the beginning. That being said, the body of work we are providing with the attached comments is the result of subject-matter experts doing their best in the limited time they had to analyze the NOPR. DOE will note the majority of comments concerns specific sections of the NOPR document only; a few comments are made toward the technical support document, and there is no commentary on any of the six spreadsheets. Many of the "issues on which DOE seeks comment" have no AMCA response because of a lack of sufficient time and resources to respond.

AMCA's comments are provided in four sections. Section 1 provides comments on topics germain to GFB. Section 2 provides comments germain to ACF. Section 3 provides responses to the issues on which DOE seeks comment as enumerated in the NOPR. Section 4 consists of the appendices called out in sections 1 and 2.

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[1]AMCA International is a not-for-profit association of manufacturers of fans, dampers, louvers, air curtains, and other air-system components for commercial HVAC, industrial-process, and power-generation applications. With programs such as certified ratings, laboratory accreditation, verification of compliance, and international-standards development, its mission is to advance the knowledge, growth, and integrity of the air-movement-and-control industry.

To summarize what AMCA believes are its most important positions and findings:

1. In 2022, AMCA contracted a third party to develop a database of 2021 fan-shipment data. The database contains detailed sales data voluntarily submitted by AMCA member companies. After reviewing the NOPR, AMCA concluded it needed to provide updated data to improve the accuracy of the analyses driving DOE's proposed fan-energy-index (FEI) limits. AMCA provided the database to Lawrence Berkeley National Laboratory and Guidehouse under non-disclosure agreements. It is not known at this time if DOE will publish a notice of data availability (NODA) that communicates what DOE contractors will do with the data. Under normal circumstances, with DOE receiving such a voluminous set of updated market data—substantially greater than the 10-year-old dataset on which it had previously relied—stakeholders would be given the opportunity through publication of a NODA to digest and react to data before DOE presented regulatory recommendations.

   AMCA requests DOE issue such a NODA and allow the normal time for parties to examine it.

2. AMCA is certain the FEI levels arrived at for GFB are too high to be feasible. AMCA believes the analytical approach undertaken to set and justify the levels is inconsistent with the nature of the FEI metric and, in fact, works against the metric's purpose to improve fan selections for given duty points. In short, the levels proposed by DOE would make many fan types unavailable for typical applications at high airflows and pressures, despite them being very efficient in moving air. The high levels proposed by DOE that have been used to calculate energy savings from the rulemaking are illusory. The mathematics used to calculate the savings are inconsistent with a properly applied FEI metric. We can see both the equations and philosophy of pre-FEI measures of fan efficiency creeping into the most recent analysis and leading to erroneous conclusions.

   AMCA recommends DOE consider AMCA's alternative analytical approach around a conceptual "FEI_{max}" defined in AMC's GFB comments as using the same 10,000 operating points defined in Section 1 DOE uses in its analyses and then a table of proposed FEI levels by product class. This leads to lower total energy savings than DOE's proposed rule; however, the savings calculation in the NOPR was fundamentally flawed and those savings never were available.

3. AMCA believes the "B" adjustment is too complex and results in an unnecessary and confusing doubling of the product classes from nine to 18. Given the broad and deep changes the fan industry and industries that rely on fans will have to make to respond to the energy standard, AMCA believes that simplicity is warranted where it can be found and that DOE can revisit certain areas in subsequent rulemakings.

   AMCA recommends DOE establish a simple adjustment of 5 percent to align with model and state energy codes that have used that adjustment since 2019. By the

time the DOE energy standard takes effect, the adjustment will have been in place for approximately 10 years.

4. AMCAs comments describe in great detail how DOE's proposed FEI limits for radial fans could result in rapid failures of shrouded fans in applications where unshrouded fans are used.

   AMCA recommends the establishment of a new product class for unshrouded radial fans, which already have a size exemption in the test procedure and NOPR for fans having a diameter less than 30 in. or a width less than 3 in. Note that this would not eliminate the requirement that unshrouded radial fans be tested to the DOE procedure if in its scope.

5. AMCA provides substantial commentary around the NOPR's content regarding ACF. Primary among the comments is that DOE's proposed tabular approach to creating subclasses of ACF based on diameter could easily lead to gaming and penalize high-performing ACF while "giving a pass" to poor-performing ACF. Thus, an equation-based approach is proposed, as are efficacy levels. Additionally, AMCA provides commentary around the impact guards have on ACF performance, which we could not find any prior documentation about in open-source literature. With the limited time available, one manufacturer was able to provide measurements taken in an AMCA-accredited laboratory to provide insights and recommendations for considering the effects of guards on an ACF regulation.

   AMCA recommends DOE adopt AMCA's proposals for ACF.

6. AMCA commends DOE for proposing to amend the GFB test procedure with a calculation method for regulated motors and drives. Two methods proposed by DOE, which can be summarily referred to as the IEC Standard 61800-9-2:2023 approach and the modified ANSI/AMCA Standard 214-21 approach, would be alternatives to wire-to-air testing and alternative efficiency-determination methods (AEDM) as a nod toward relieving manufacturers of substantial testing burden. Without this proposed amendment, manufacturers would be forced to use the AEDM approach because wire-to-air testing of the magnitude needed is simply unachievable. And this is not without problems, as there is insufficient subject-matter expertise and resources, especially among smaller fan manufacturers, to develop AEDM and perform validation testing for products they do not produce.

   AMCA recommends that DOE allow use of the modified ANSI/AMCA Standard 214-21 approach for regulated motors and VFD. AMCA has provided new coefficients needed to make this calculation method sufficiently conservative. Additionally, AMCA recommends IEC Standard 61800-9-2:2023 be recognized for certain unregulated motors and VFD, but only when their full performance has been tested.

7. In relation to Item 5 above, manufacturers are laboring intensively to meet the deadline for California's Title 20 regulation for fans and blowers, while also endeavoring to comply with the DOE test procedure in time to meet the extended deadline of April 29, 2024, all while also having had to respond to the energy-standard NOPR. While the calculation methods proposed by DOE promise to relieve testing (and AEDM engineering) burden, unless extensions or delays in enforcement, along with an expedited final rule for test-procedure changes, are granted, manufacturers will be under legal peril with both DOE and the California Energy Commission, as they will be unsure which method, if not both, DOE would allow.

   AMCA recommends DOE publish a final rule on the calculation method(s) it is considering for the test procedure for GFB. This will provide earliest-possible clarity on calculating ratings in general and toward the April 29 deadline for the Title 20 regulation specifically.

AMCA thanks the Department for considering the comments and supporting information below.

Sincerely,

Michael Ivanovich
Senior Director, Global Affairs
AMCA International
mivanovich@amca.org

AMCA International Comments to:
Notice of Proposed Rulemaking for Energy Standard for Fans and Blowers

Docket: EERE–2022–BT–STD–0002; RIN 1904–AF40

March 19, 2024

Point of Contact:

Michael Ivanovich
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SECTION 1: COMMENTS ON GENERAL FANS AND BLOWERS (GFB)

GFB 1: AMCA Provided a 2021 Fan-Shipment Database to Replace the 2012 Database

AMCA's comments about this NOPR should be understood in the context of its commitment and the resources it has expended throughout the rulemaking to ensure DOE and its consultants have the most current and relevant marketplace data on which to base decisions. The most recent example of this is the provision of a database of 2021 shipment data. In developing the database, AMCA imposed strict anonymity requirements such that each product class has at least five manufacturers, with no single manufacturer comprising more than 50 percent of the data in the category. With this limitation, data for unhoused centrifugal fans was omitted from the final edition of the database. Data underwent rigorous scrutiny for quality, with the contractor contacting data providers to resolve questionable data. The final database has more than 178,000 fan shipments that are within the scope of the proposed DOE energy standard, with sufficient participation from manufacturers to populate eight of the nine DOE product classes.

GFB 2: DOE's Proposed FEI Values for GFB Are Too High, and Their Foundational Analyses are Inconsistent with the Nature of the FEI Metric

DOE's proposed fan-energy-index (FEI) values, which measure the ratio of the energy of a reference fan to that of an actual fan at a given duty point, are higher than expected for most equipment classes. For some classes, the proposed FEI levels cannot be attained without eliminating product availability for consumers, as the products are at their current maximum technological capabilities in the market. While DOE defined max-tech solely in terms of FEI without considering duty points, AMCA urges DOE to focus instead on the maximum achievable fan efficiency for each equipment class, which is readily available within life-cycle-cost (LCC) sample selections.

It is crucial to understand that FEI is a duty-point metric, and comparing FEI values at different duty points can be misleading, especially when comparing high-air-power fans to low-air-power fans. There is a distinction between "fan efficiency" as defined in ANSI/AMCA Standard 210/ASHRAE Standard 51, *Laboratory Methods of Testing Fans for Certified Aerodynamic Performance Rating*, and the FEI "fan-efficiency metric" that is not evident in the NOPR. Accompanied by an explanation and analysis, AMCA proposes alternate FEI values based on maximum achievable fan efficiency. These values allow highly efficient, stable selections across operating ranges while removing inefficient options and saving energy.

Why FEI Levels Vary Between Duty Points

The reference-fan shaft-power equation in ANSI/AMCA Standard 214-21, *Test Procedure for Calculating Fan Energy Index (FEI) for Commercial and Industrial Fans and Blowers*, includes coefficients that increase required fan efficiency at high airflow and pressure (high fan air power). The result is that a low-air-power fan having the same efficiency as a high-air-power fan yields a higher FEI value. Assuming that an FEI value provides the same efficiency at all duty points leads to incorrect conclusions as to market capability and availability.

Looking at the Equation

The FEI equation FEP_{ref}/FEP_{act} can be simplified when motor and transmission efficiencies are negated. DOE uses the term “bare shaft FEI” in the NOPR. The bare-shaft FEI is $H_{i,ref}/H_{i,act}$. For a total-pressure application, $H_{i,ref}$ is calculated using Equation 1, which is taken from ANSI/AMCA Standard 214-21:

Equation 1:

$$H_{i,ref} = \frac{(Q + Q_0)\left(P_t + P_0 \times \frac{\rho}{\rho_{std}}\right)}{6343.3 \times \eta_o}$$

Where:

$H_{i,ref}$	is reference fan shaft power in kW (SI) or hp (I-P)
Q	is fan airflow rate in m^3/s (SI) or cfm (I-P)
P_t	is fan total pressure at density ρ in Pa (SI) or in. wg (I-P)
ρ	is fan air density at duty point in kg/m^3 (SI) or lbm/ft^3 (I-P)
ρ_{std}	is standard air density, 1.2 kg/m^3 (0.075 lbm/ft^3)
Q_0	= 0.118 m^3/s (SI) or 250 cfm (I-P)
P_0	= 100 Pa (SI) or 0.40 in. wg (I-P)
η_0	= 66% (0.66)

$H_{i,ref}$ incorporates two coefficients, Q_o and P_o, which increase $H_{i,ref}$ value and subsequent FEI value. At low volume and pressure duty points, these coefficients can play a significant role by increasing the relative value of $H_{i,ref}/H_{i,act}$, or FEI.

Graphically, this can be seen on a representative FEI bubble diagram (Figure 1).

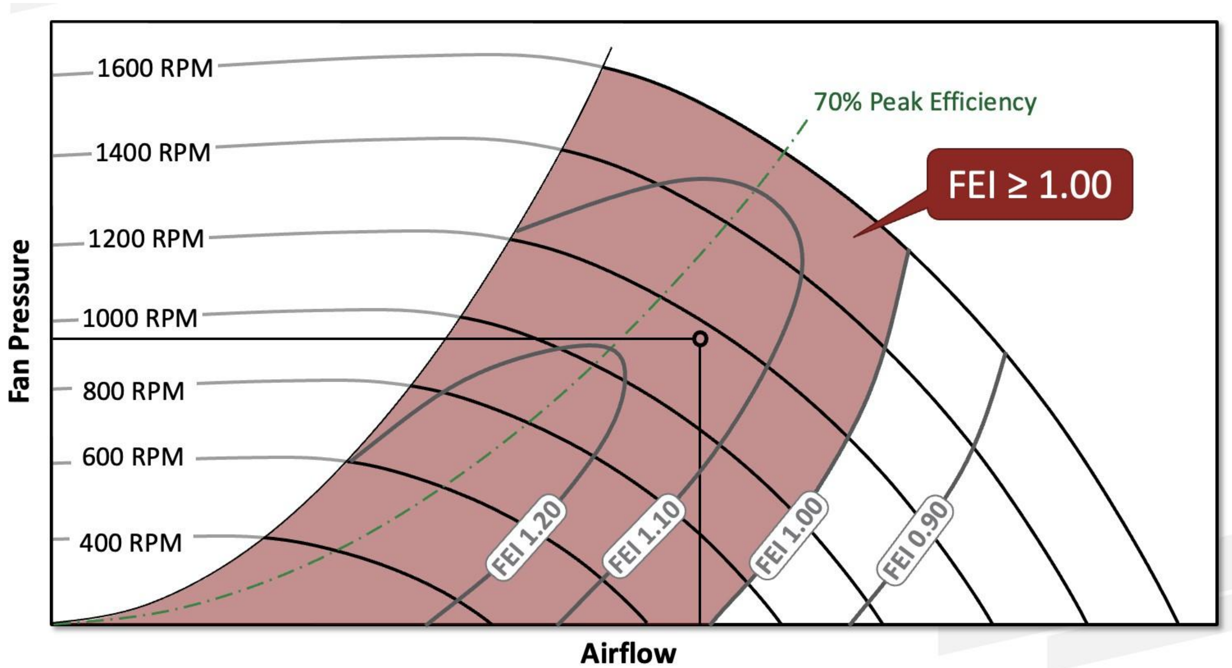


FIGURE 1. "Bubble chart" showing FEI bands at multiple speeds.

Moving upward along the peak-efficiency curve, the width of the FEI bubble narrows as air power increases and FEI values decrease. At higher air powers, the bubble narrows to a singular point, even on the peak-efficiency curve.

Reviewing some real-world selections helps to show the effect the coefficients have on FEI values. Figure 2 shows a duty point of 5,000 cfm and 5 in. w.g. has an FEI of 1.28. If you follow the system curve downward (same fan efficiency), however, the fan at lower air power has an FEI of 1.40.

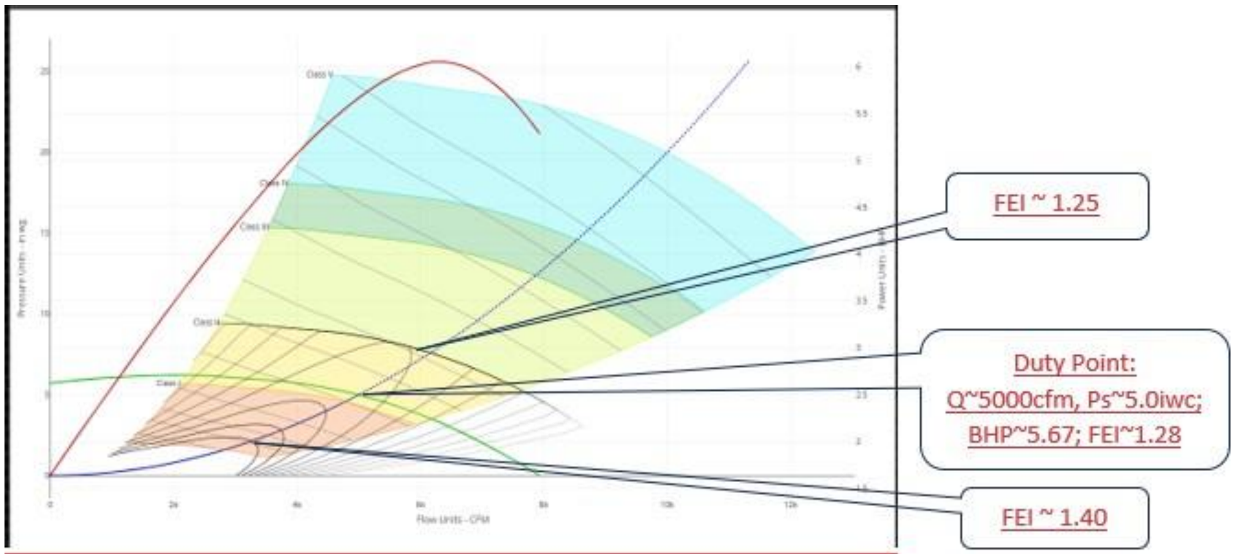


FIGURE 2. Fan performance curve (airflow vs. pressure) showing how FEI varies with duty point along a system curve.

In Figure 3, a fan with high air power of 40,000 cfm at 20 in. w.g. has an FEI of 1.26. To generate higher FEI values, one would have to descend significantly down the system curve (same efficiency). Note the large zone of negligible FEI change at high air power. This occurs when Q_o and P_o are small with respect to airflow and pressure.

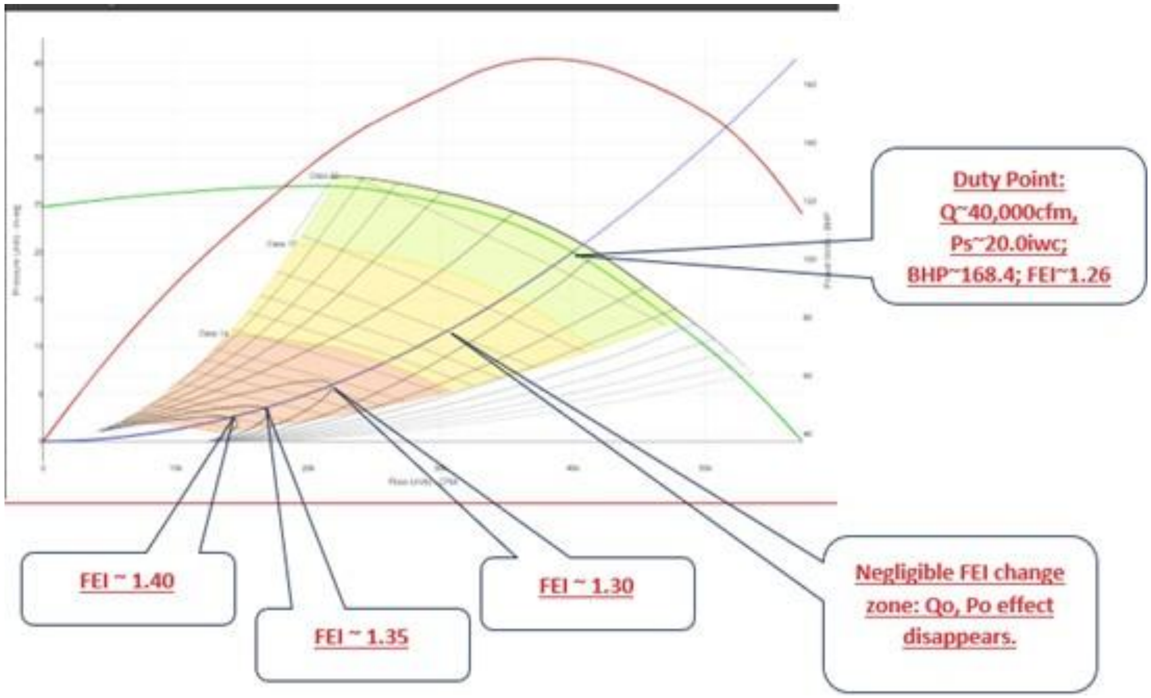


FIGURE 3. Fan performance curve for high air power (airflow vs. pressure) showing how FEI varies with duty point along a system curve.

The sample data points provided in the LCC analysis are a good representation of the range of duty points needed by consumers (Figure 4). Using these data points and looking only at FEI values without considering duty point or corresponding air power, one would mistakenly conclude the entire performance range would be capable of the higher FEI values attained at low air power.

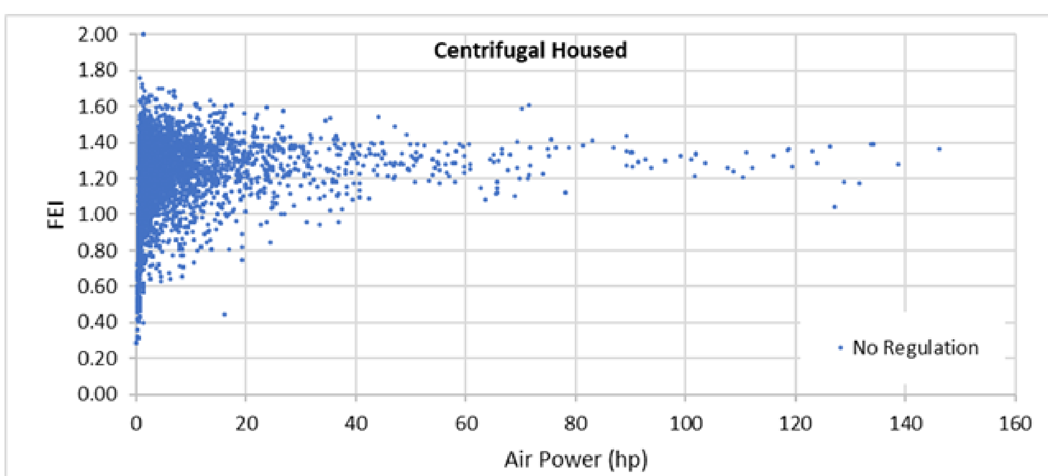


FIGURE 4. Centrifugal-housed LCC data points, air power vs. FEI.

Another way to illustrate the impact of Q_0 and P_0 is to duplicate the previous plot with the two coefficients set to zero. This is shown in Figure 5, which clearly indicates a consistent maximum FEI value for both low- and high-air-power selections.

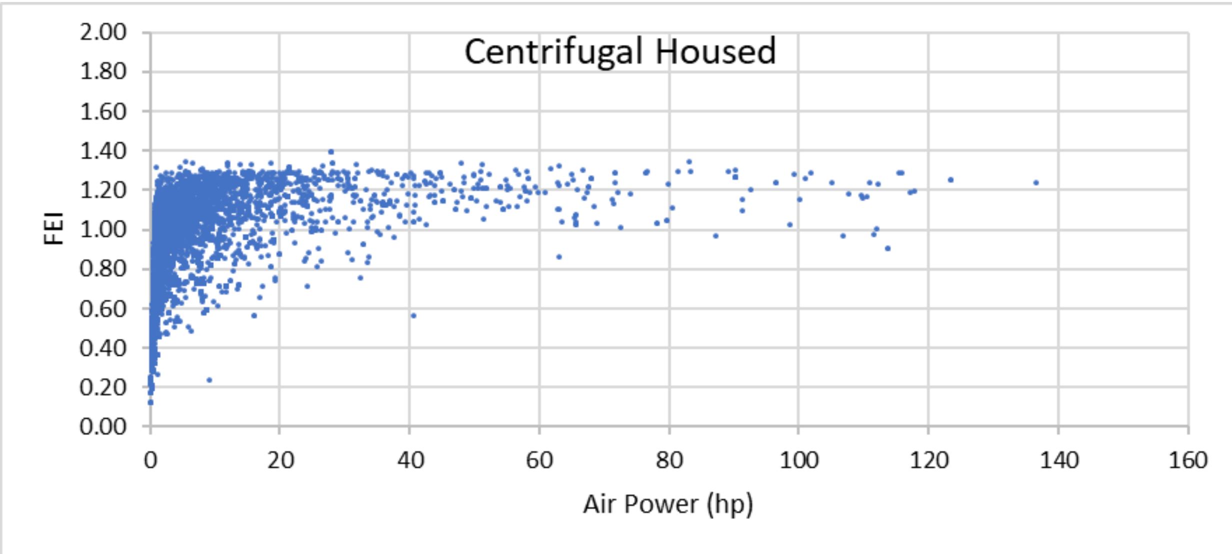


FIGURE 5. Centrifugal-housed data points, air power vs. FEI with Q_o and P_o set to zero.

All points along the top edge of this data are selected at peak fan efficiency. In fact, peak fan efficiency can be determined directly from this data. If the top edge is at a value of 1.30, the peak fan efficiency is 1.30 times 66 percent (n_o), or about 86 percent.

Given these considerations, AMCA now will describe a methodology for determining “max-tech FEI” (FEI_{max}) based on peak efficiency for the entire operating range of each of the nine equipment classes. The FEI value will provide stable operation for variable-air-volume (VAV) operation and allow selections near peak efficiency for available sizes.

The centrifugal-housed equipment class will be used as an exemplar product class. Charts for all equipment classes are provided in Appendix A.

FEI_{max} Methodology

In the LCC analysis, DOE provided 10,000 duty points for each equipment class. The volume/pressure duty points are plotted in Figure 7. AMCA believes these points are representative of the duty points currently required by consumers for stand-alone fans. Some classes may have underrepresented fans embedded in OEM equipment.

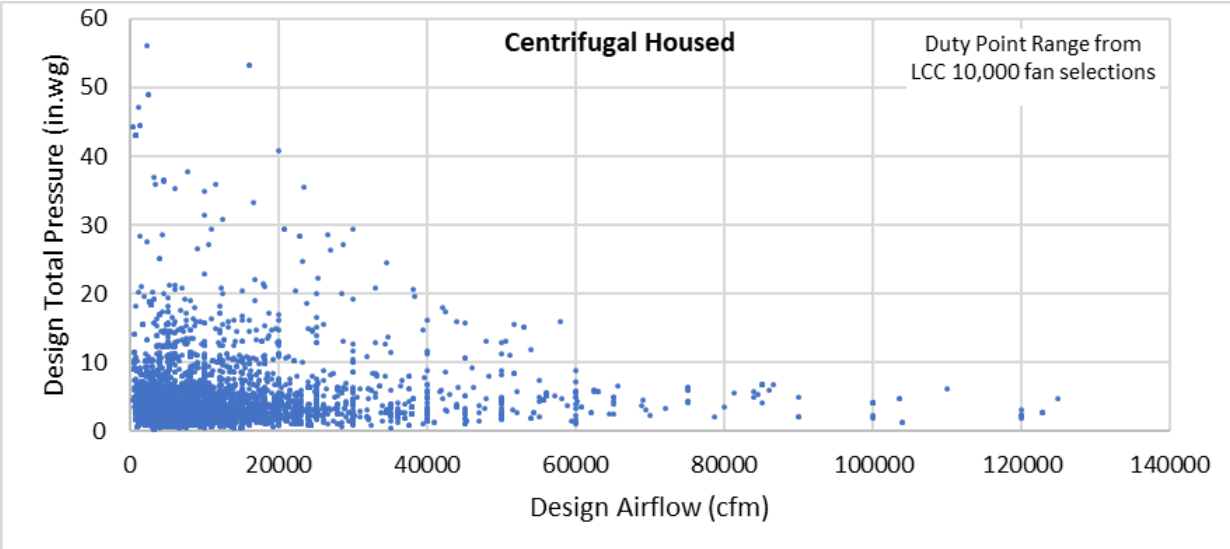
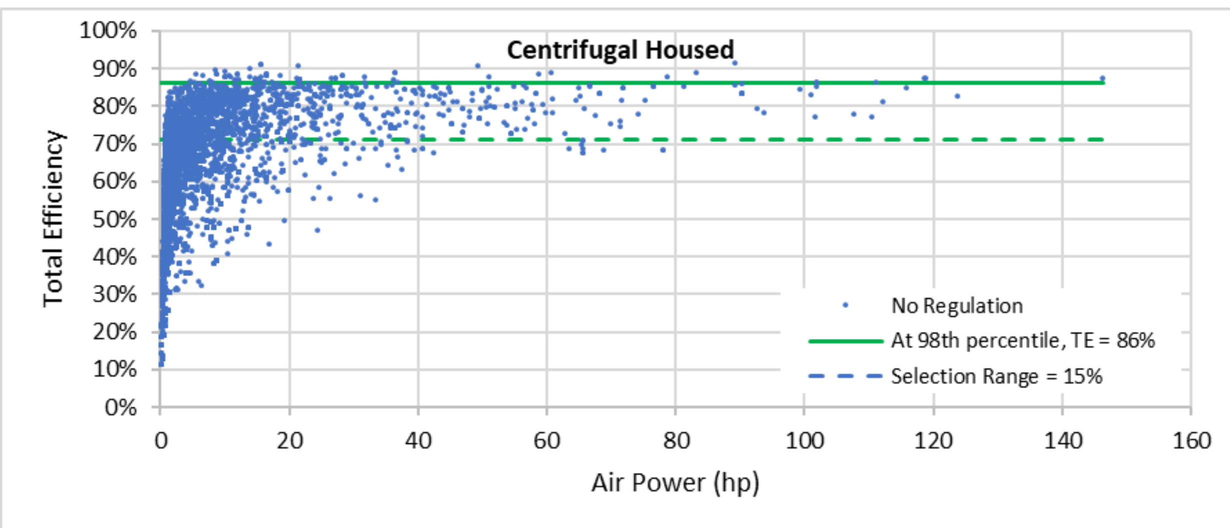


FIGURE 7. LCC data points for centrifugal housed fans, airflow vs. pressure.

Using these same duty points and calculating total efficiency at the baseline power, we plot the values against air power (Figure 8). The equipment class shows a consistent peak efficiency level for all air-power selections. The 98th percentile yields a total efficiency of 86 percent. AMCA is considering this max-tech fan efficiency for the equipment class.



Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

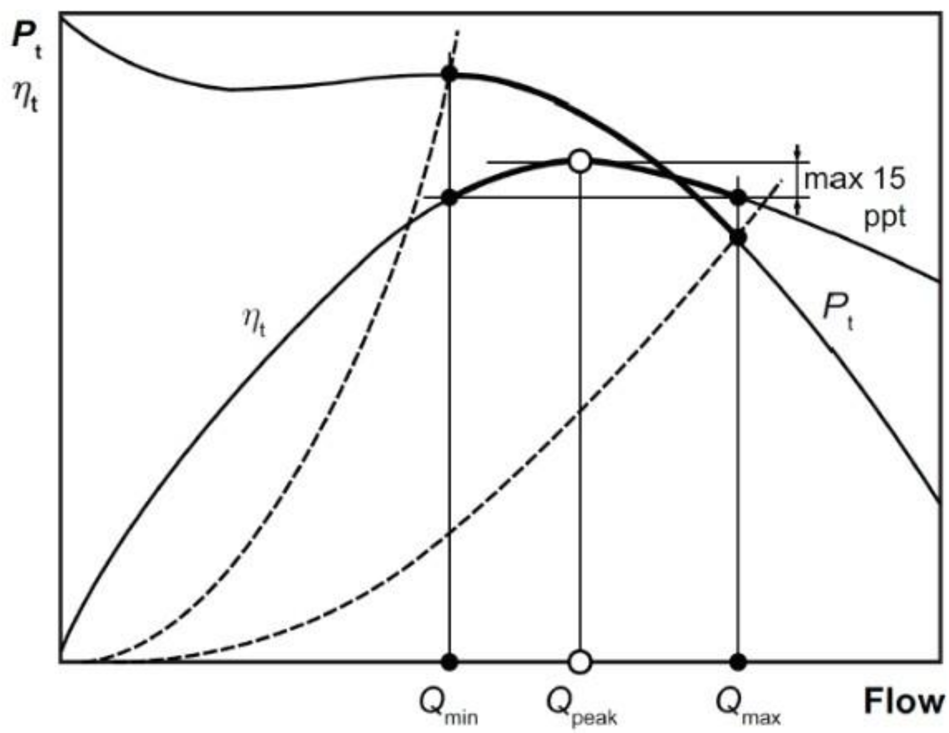
FIGURE 8. LCC data point for centrifugal housed fans, air power vs. total efficiency.

Selections at higher air powers (right side of chart) tend to cluster near the peak total-efficiency level and within a 15-percent band of peak efficiency.

A 15-percent band is being used to allow selections whereby peak efficiency falls between available sizes. Additionally, stability issues with VAV systems and peak-efficiency selections near surge create a need to allow duty points to vary from peak efficiency and still have stable operation for turndown operation. The 15 percentage points align with the previous metric of fan-efficiency-grade- (FEG-) allowable selections (Figure 9).

According to Section 7 of ANSI/AMCA Standard 205-19, *Energy Efficiency Classification for Fans*, any code or specification that requires an FEG shall also require that fan efficiency at all intended operating points be within 15 or fewer percentage points of fan peak efficiency. The restriction imposed by this limitation is explained in Annex B of ANSI/AMCA Standard 205-19.

According to Annex B of ANSI/AMCA Standard 205-19, energy-efficient operation requires that a system fan be selected close to peak fan efficiency. Fan operating efficiency at all intended operating points shall be less than 15 percentage points below fan peak efficiency (see the heavier portions of the fan curves in Figure 9).



P_t is the fan total pressure
η_t is the fan total energy efficiency
Q is the flow
Q_{min}, Q_{max} represent the flow range for the allowable fan efficiency range

Figure B.1 — Fan Efficiency and Flow Range Allowance for an Application with an Outlet Duct

FIGURE 9. Figure B.1 from ANSI/AMCA Standard 205-19.

The following example of a forward-curved, double-wide blower shows how an increased FEI requirement reduces the selectable range and places selections closer to the stall region. If the equipment requires a turndown, it can easily go into the stall region, causing damage and poor performance. The 15-percent selection band away from peak efficiency allows users to make efficient selections while allowing safe turndown, saving even more energy (Figure 10).

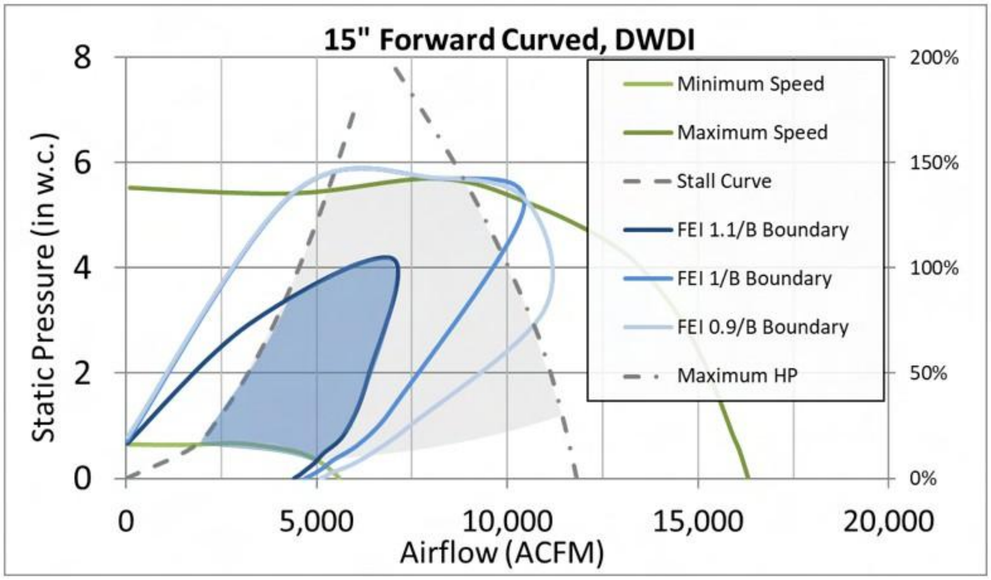


FIGURE 10. Higher FEI points closer to stall curve.

Once max-tech peak fan efficiency (AMCA used 98th-percentile fan efficiency) and the max-tech selection efficiency band (AMCA used a 15-ppt range) are determined, max-tech in terms of FEI (*FEI_{max}*) can be calculated.

FEI_{max} is the maximum FEI value using LCC sample selections that allows stable operation and efficient fan selections without limiting the operating range of the equipment class.

FEI_{max} is numerically calculated using the following equations.

The calculation is based on “bare-shaft FEI,” as documented in the NOPR:

$$FEI_{bare\ shaft} = \frac{Reference\ Fan\ Shaft\ Power}{Actual\ Fan\ Shaft\ Power} = \frac{H_{i,ref}}{H_{i,act}}$$

The calculation for $H_{i,ref}$ is from ANSI/AMCA Standard 214-21, Section 5.1.1, while the calculation for $H_{i,act}$ is from ANSI/AMCA Standard 210/ASHRAE Standard 51:

$$FEI_{bare\ shaft} = \frac{\left[\frac{(Q+Q_0)\left(P_t+P_0\times\frac{\rho}{\rho_{std}}\right)}{6343.3\times\eta_0}\right]}{\left[\frac{(Q\times P_t)}{6343.3\times\eta}\right]}$$

Substituting constants and constraining to standard density:

$$FEI_{max} = \frac{\left[\frac{(Q_{max}+250)(P_{t,max}+0.4\times1.0)}{6343.3\times66\%}\right]}{\left[\frac{(Q_{max}\times P_{t,max})}{6343.3\times\eta_{max}}\right]}$$

Rearranging:

$$FEI_{max} = \left(\frac{\eta_{max}}{66\%}\right)\left(\frac{Q_{max}+250}{Q_{max}}\right)\left(\frac{P_{t,max}+0.4}{P_{t,max}}\right)$$

Where:
Q_{max} is the maximum airflow range (available maximum range for the equipment class)
$P_{t,max}$ or $P_{s,max}$ is the maximum pressure range, total or static (available maximum range for the equipment class)
η_{max} is the max-tech efficiency - 15 percentage points

This calculation is performed for each equipment class. The results for FEI_{max} are shown in Table 1.

Max-Tech FEI Calculations										
	DOE Proposed Levels			**Static Pressure Basis**			**Total Pressure Basis**			
Equipment Class	**Fan**	**Efficiency**	**Max Airflow**	**Max Ps (in.wg)**	**Max Tech SE (%)**	**Max Stable**	**Max Pt**	**Max Tech TE (%)**	**Max Stable TE (%)**	**FEImax**
Axial Inline	1.18	EL3	100000				10	85%	70%	**1.10**
Axial Panel	1.48	EL4	60000	1.25	53%	38%				**0.84**
Axial PRV	0.85	EL4	60000	2	48%	33%				**0.66**
Centrifugal Housed	1.31	EL4	100000				20	86%	71%	**1.10**
Centrifugal Unhoused	1.35	EL4	60000	12	76%	61%				**1.05**
Centrifugal Inline	1.28	EL5	80000				10	76%	61%	**0.96**
Radial Housed	1.17	EL4	40000				60	80%	65%	**1.00**
Centrifugal PRV - Exhaust	1.00	EL4	30000	4	61%	46%				**0.85**
Centrifugal PRV - Supply	1.19	EL5	30000				2.5	58%	43%	**0.76**

TABLE 1. *FEI_{max}* calculation table.

AMCA makes the following recommendations for FEI (Table 2).

Recommended FEI Levels				
	DOE Proposed Levels			
Equipment Class	Fan Energy Index (FEI)	Efficiency Level	FEImax	Recommended Level (FEI)
Axial Inline	1.18	EL3	**1.10**	**1.10**
Axial Panel	1.48	EL4	**0.84**	**1.00**
Axial PRV	0.85	EL4	**0.66**	**0.85**
Centrifugal Housed	1.31	EL4	**1.10**	**1.10**
Centrifugal Unhoused	1.35	EL4	**1.05**	**-***
Centrifugal Inline	1.28	EL5	**0.96**	**1.00**
Radial Housed	1.17	EL4	**1.00**	**1.00**
Centrifugal PRV - Exhaust	1.00	EL4	**0.85**	**1.00**
Centrifugal PRV - Supply	1.19	EL5	**0.76**	**1.00**
*Recommended values pertain to stand alone fans only. Fans used in equipment should be analyzed based upon that specific equipment's utility and performance range required.				
AMCA does not have sufficient data in its 2021 database to make a recommendation for the FEI level for Centrifugal Unhoused				

TABLE 2. AMCA-recommended FEI values.

AMCA's analysis shows FEI_{max} values below the FEI values proposed by DOE. AMCA evaluated each equipment class to determine if it was possible to round up to achieve greater energy savings and have closer alignment with energy codes and rulemaking such as ANSI/ASHRAE/IES 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, and California titles 20 and 24 without excessive loss of utility or product range.

The lowest FEI_{max} was calculated for the axial power-roof-ventilator (PRV) equipment class and rounded up to the DOE-proposed 0.85. The FEI_{max} values of the remaining equipment classes, which were below 1.00, were rounded up to 1.00. This is indicated in the column in Table 2 labeled "Recommended Level (FEI)." AMCA recognizes that the equipment classes that have been rounded up likely will see a reduced operation range, impacts on utility for certain topologies, and possible system redesign. Individual manufacturers may comment on the loss of utility for their respective products with this "rounding up" process.

Continuing with the example of centrifugal housed fans and the proposed FEI level of 1.10, Figure 11 shows red dots where fan selections no longer would be made and FEI levels would be increased to at least the green line. The result is that 41 percent of the current product selections used in the LCC analysis would be eliminated, resulting in a 5.6-percent energy savings.

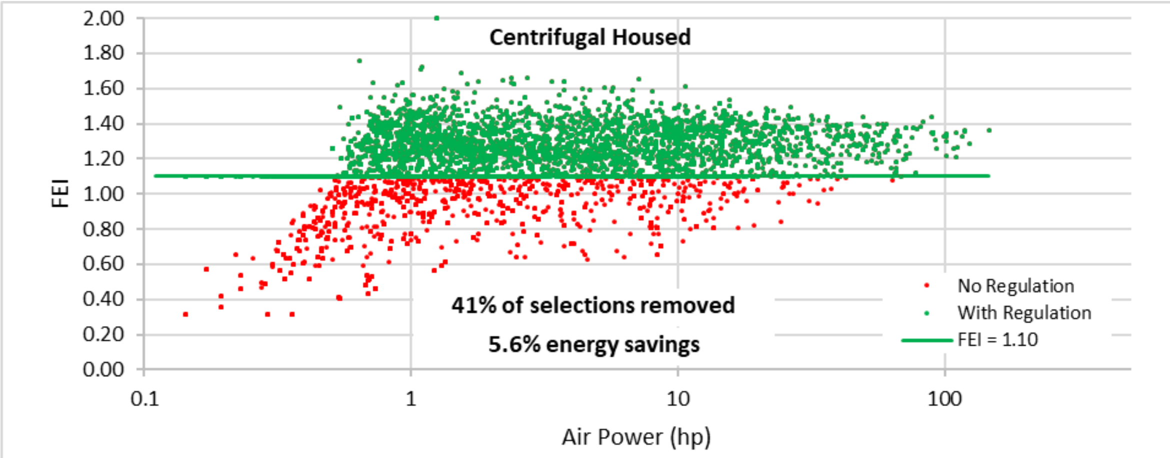


FIGURE 11. Centrifugal-housed air power vs. FEI indicating recommended FEI level.

Moving the green line up or requiring a higher FEI value would have an even greater impact on the equipment class and significantly limit the equipment class's performance range and utility. A change from the recommended 1.10 to 1.20 would remove another 16 percent of selections for a total of 57 percent and affect more than half of the data points in the LCC analysis.

The previous graph of FEI vs. air power (Figure 4 above) used a linear air-power axis to clearly show the difference between low- and high-air-power selections. In Figure 11, the air power was deliberately changed to a log scale to illustrate the high number of lower-power selections affected by the proposed rule.

Analyses and charts are provided for each equipment class in Appendix A.

AMCA believes the proposed analysis methodology and FEI values allow product selections throughout the entire operating range of each equipment class. They require good fan selections near peak efficiency and save energy without eliminating utility.

Substitution Considerations

Use of the AMCA-proposed FEI levels would avoid the unintended consequences of equipment-class switching attributed to significant differences in DOE-proposed FEI requirements across equipment classes.

An unintended consequence of the differences in proposed FEI values across fan and blower equipment classes is that customers may switch to a class with lower FEI requirements if unable to meet the level for their initial class. This penalizes fans with higher FEI values and results in lower overall energy savings.

Market volume may shift to fans with lower FEI values that save less energy. Substituting classes also may increase overall carbon footprint. For example, as shown in Figure 12, customers currently using a housed centrifugal fan with ducting for rooftop exhaust may be forced to switch to a centrifugal PRV exhaust with a lower FEI requirement to meet the same duty point.

Selections at 7000 cfm @ 1.25"

Fan	Min FEI	Act FEI	Compliant
24” Housed Centrifugal	1.31	1.30	No
24” Centrifugal PRV	1.00	1.10	Yes

FIGURE 12. How a housed centrifugal fan could be substituted with a rooftop exhaust fan to the detriment of energy savings.

AMCA cautions that the proposed FEI levels create a regulatory incentive for customers to switch to lower efficiency classes, undermining energy-savings goals. The alternate FEI values proposed by AMCA account for maximum achievable efficiencies across duty points while avoiding unintended shifting between classes.

In another example, because of a high FEI requirement and an inability to increase the size of a building opening, a customer that typically would use a sidewall exhaust panel fan for ventilation is forced to look at other options (Figure 13). Because the panel fan has a specific utility, the substitutions are less desirable from the standpoints of installation cost, product cost, and overall carbon footprint.

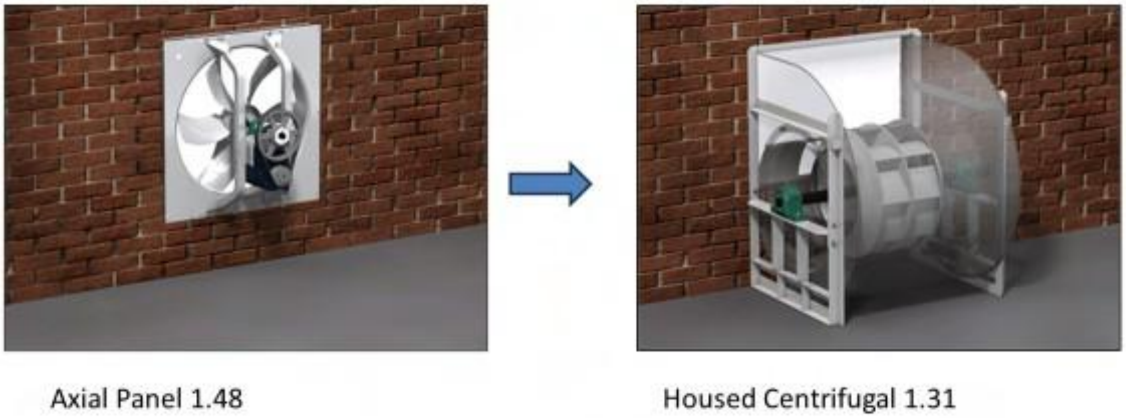


FIGURE 13. Example of an axial panel fan being replaced with a housed centrifugal fan because of the customer's inability to facilitate a larger opening for a higher-FEI axial panel fan.

The substitutions for a panel fan are limited. Each option has a higher cost and increased weight, as shown in Table 3. Weight and geometry impact shipping costs, site-modification costs, embodied energy, and other sustainability considerations.

Model Comparisons

Reference Duty Point 6000 cfm @ 0.5"

	Axial Panel	Cent. PRV x 2	Housed Cent.
Req. FEI	1.48	1.00	1.31
FEI actual	1.14	1.11	1.39
Size	24"	16"	24"
FEP	1.2	1.38	1.08
# of fans	1	2	1
Weight	156	224	358
Cost Ratio	1	2.7	3.0

TABLE 3. Comparison of an axial panel fan, two centrifugal PRV, and a housed centrifugal fan being used to replace an axial panel fan with limited options for site modifications.

To summarize, with the AMCA-proposed FEI values, significant differences between equipment classes would be avoided and customers could base fan selections on utility.

GFB 3: Acceleration of Final Rule for NOPR Sections Amending GFB Test Procedure

AMCA is concerned about DOE modifying the test procedure after the effective date of Oct. 30, 2024, and even when the 180-day extension granted to 34 manufacturers expires on April 29, 2024. April 29, 2024, is the day manufacturers are required to begin mandatory compliance filing for California's Title 20 fan regulation. California's Title 20 fan regulation is in flux because of a post-final rulemaking adopting the DOE test procedure and making other changes.

In a letter to DOE dated March 1, 2024, AMCA requests that DOE accelerate publication of a final rule on the portions of the energy-standard NOPR that amend the test procedure and that it reset the compliance deadline for the test procedure to 180 days after the final rule. Additionally, AMCA asks for a delay in enforcement of the test procedure until the final rule is published. The rationale for these requests are explained in the letter. Since the letter was transmitted, CEC has released a 15-day rulemaking that proposes filing-data changes (Table X of Title 20) that would accommodate either or both of the test-procedure changes proposed in the NOPR.

GFB 4: Calculation-Only Methods for GFB Test Procedure

Section 2.6 of the NOPR proposes two calculation-only methods (no validation testing required) for determining FEI for fans tested without a speed controller. One is based on IEC Standard 61800-9-2:2023, *Adjustable speed electrical power drive systems (PDS) - Part 9-2: Ecodesign for motor systems - Energy efficiency determination and classification*, and supplemental materials. DOE states it also would consider ANSI/AMCA Standard 214-21 Section 6.4.2.4.2 but with modified coefficients based on IE2-level variable-frequency drives (VFD) that would make FEI calculations more conservative. Alternatively, per the test procedure, manufacturers can test everything wire-to-air or use approved alternative efficiency-determination methods (AEDM) that would need validation testing. In providing the two calculation methods for review/comment, DOE stated it is seeking to reduce manufacturer testing burden.

AMCA supports having a calculation-only method available, but there are several issues that need to be addressed. The first is timing, as discussed in GFB 3.

The second issue is that IEC Standard 61800-9-2:2023 is highly complex, and fan manufacturers had been unaware of it until the NOPR brought it to their attention. More time is needed for industry to learn how the method works and how it can be integrated into their selection software.

The third issue is the modified coefficients for ANSI/AMCA Standard 214-21 were not provided by DOE in the NOPR; they needed to be developed by AMCA and in time to meet the deadline for the review comments. Manufacturers have not had sufficient time to widely learn about and test the coefficients. More time is needed for industry to learn how the method works and how it can be integrated into their selection software.

Lastly, if DOE were to accept both methods, AMCA believes it is necessary that manufacturers be able to learn about, test, and consider how to apply each so they can have the confidence to select one or the other. If that is not an option, AMCA's position is that the IEC Standard 61800-9-2:2023 method *must* be used for replacing Section 6.5 of ANSI/AMCA Standard 214-21 and the modified ANSI/AMCA Standard 214-21 method would be much easier for manufacturers to apply for replacing ANSI/AMCA Standard 214-21 Section 6.4.2.4.2.

In the energy standard NOPR, DOE is considering amending the test procedure a method of calculating the part-load efficiency of regulated 3-phase induction motors paired with commercially available VFD. This calculation method is critical for fan manufacturers to be able to use shaft-to-air fan test data. Induction motors are the workhorse of the U.S. fan market. They long have been regulated and are commoditized to the point their performance is well-known. They often are paired with VFD (by third parties) for variable-volume applications.

While DOE recognized the constant-speed motor-efficiency calculations of ANSI/AMCA Standard 214-21 Section 6.4.2.3, it did not recognize the motor and VFD calculations of Section 6.4.2.4. DOE is now proposing two alternatives for this calculation method. The first is the interpolation method of IEC Standard 61800-9-2:2023 using loss coefficients corrected to reflect U.S. motor regulations and VFD losses reflecting baseline IE2 efficiencies. The second uses the ANSI/AMCA Standard 214-21 method of Section 6.4.2.4 but with coefficients modified to align with the first alternative.

If having to choose between these two options, AMCA recommends the modified ANSI/AMCA Standard 214-21 method. AMCA's position is that the IEC Standard 61800-9-2:2023 interpolation method is overly complex for this purpose. These calculations must be made instantaneously in fan-selection software. When the user enters the airflow and pressure needed to select a fan, the software sifts through thousands of possible fan selections before returning only those meeting the minimum FEI requirement. Speed of data retrieval and calculation is paramount.

In support of this recommendation, AMCA developed a new set of coefficients and procedures to use in place of Section 6.4.2.4 of ANSI/AMCA Standard 214-21, which can be found in Appendix B. These coefficients were developed to match (1) the typical loss coefficients of IEC Standard 61800-9-2:2023 Table E.4 corrected to reflect U.S. motor regulations and (2) VFD losses reflecting reference CDM losses of IEC Standard 61800-9-2:2023 Table A.1 corrected to baseline IE2 levels.

AMCA asks DOE to publish a final rule on the calculation methods it is considering for the test procedure for GFB. This will provide earliest-possible clarity on the calculation of ratings in general, specifically toward the April 29 deadline of the CEC Title 20 regulation.

Comparisons

Comparisons of combined motor and VFD efficiency for the two methods are shown in figures 14 through 19. The "AMCA 214 Modified" method follows the procedures in Appendix B. The "IEC 61800-9-2" method follows the proposed method from the energy-standard NOPR. The load ratio, or fraction of full-load motor power, is shown as the abscissa in these plots. The motor is assumed to be unloaded from full nameplate speed and power along a normal variable-torque curve, where torque varies with speed squared. The match between the "AMCA 214 Modified" and "IEC 61800-9-2" curves is consistent for all motor sizes and pole counts.

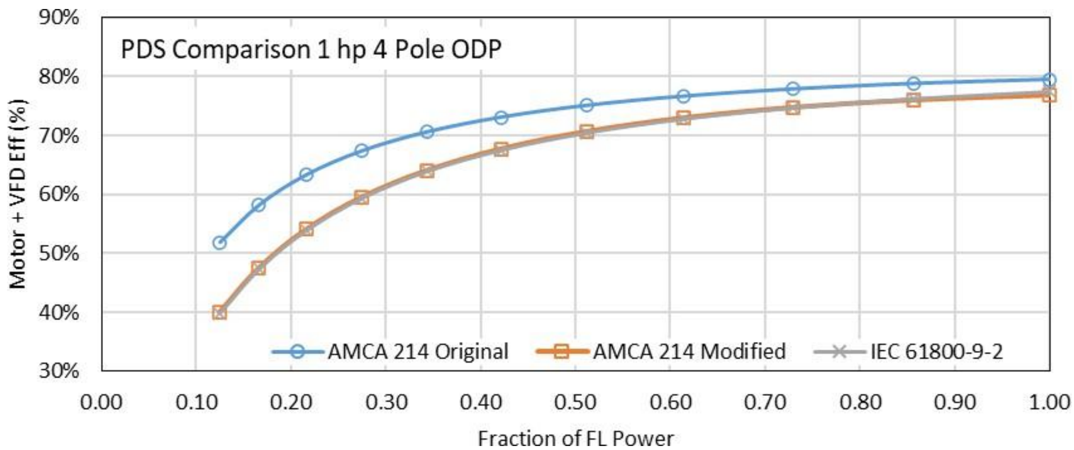


FIGURE 14. Power-drive-system (PDS) efficiency, comparison of proposed methods, 1-hp, 4-pole open-drip-proof (ODP) motor.

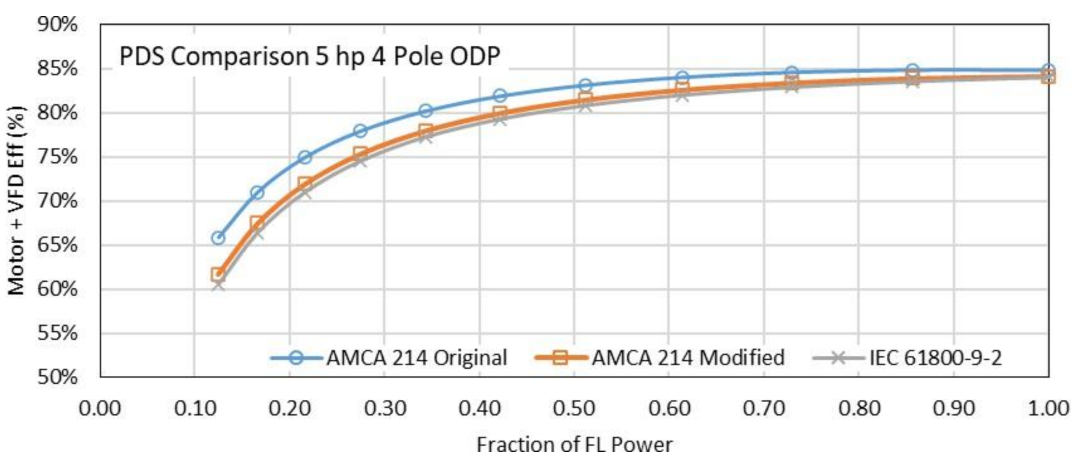


FIGURE 15. Power-drive-system (PDS) efficiency, comparison of proposed methods, 5-hp, 4-pole open-drip-proof (ODP) motor.

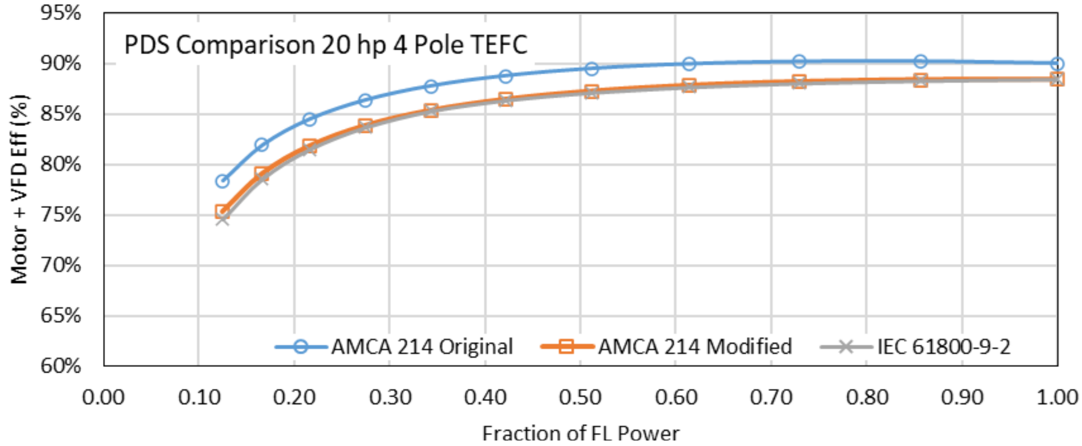


FIGURE 16. Power-drive-system (PDS) efficiency, comparison of proposed methods, 20-hp, 4-pole totally closed, fan-cooled (TEFC) motor.

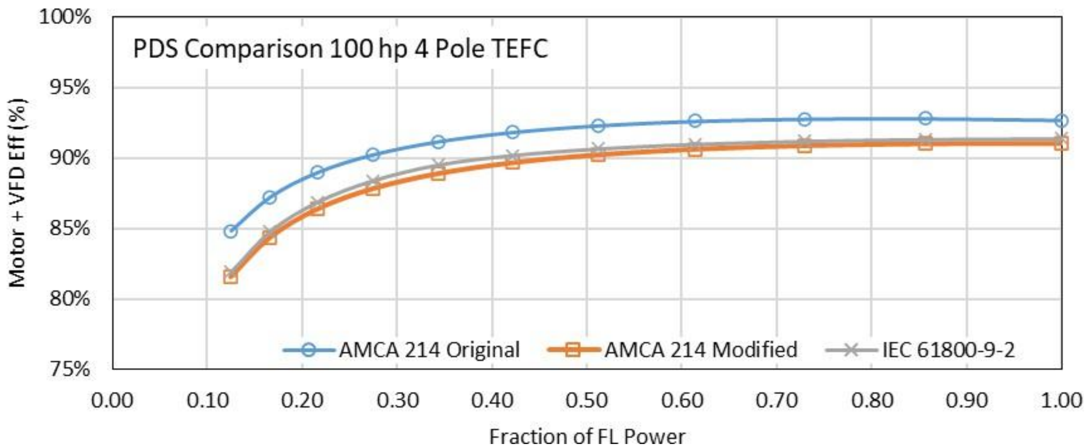


FIGURE 17. Power-drive-system (PDS) efficiency, comparison of proposed methods,– 100-hp, 4-pole totally enclosed, fan-cooled (TEFC) motor.

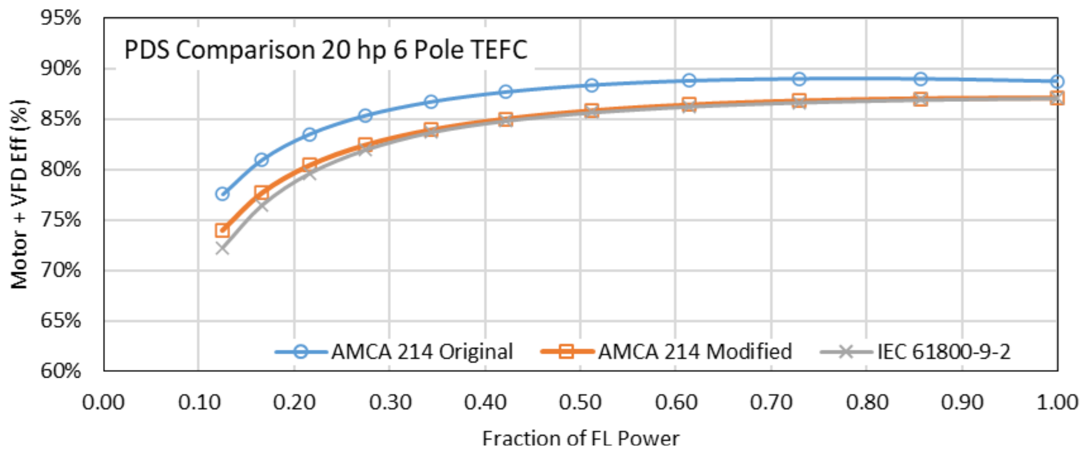


FIGURE 18. Power-drive-system (PDS) efficiency, comparison of proposed methods, 20-hp, 6-pole totally enclosed, fan-cooled (TEFC) motor.

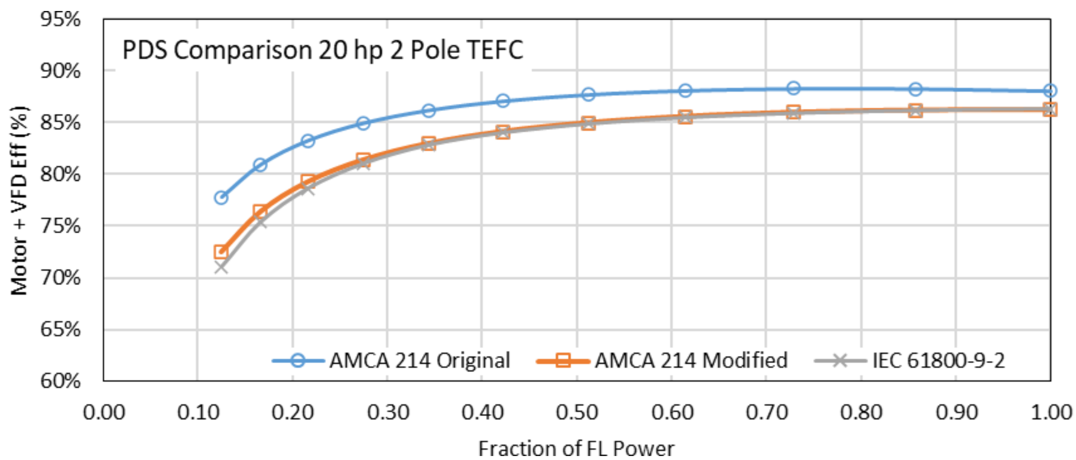


FIGURE 19. Power-drive-system (PDS) efficiency, comparison of proposed methods, 20-hp, 2-pole totally enclosed, fan-cooled (TEFC) motor.

ANSI/AMCA Standard 214-21 Section 6.5

Because DOE is considering amending the test procedure to include the calculation method for regulated motors paired with VFD, AMCA requests that DOE also recognize ANSI/AMCA Standard 214-21 Section 6.5 and a revised form of Annex F. This method is important for fan manufacturers to be able to utilize newer, higher-efficiency products that have been tested but not regulated. Its use will lead to increased overall energy efficiency.

ANSI/AMCA Standard 214-21 Section 6.5 is a procedure for combining efficiencies of components. However, the testing and interpolation requirements of Annex F were admittedly vague. At the time the standard was written, there was no test standard for motors and motor controllers that covered the range needed in the fan industry. Since then, IEC Standard

61800-9-2:2023, which includes extrapolation to frequencies (speeds) beyond 60 Hz, has been published.

The main purpose of IEC Standard 61800-9-2 is to detail test requirements for variable-speed motors, test requirements for motor controllers, and the method of interpolation/extrapolation from these test data to any other condition. A secondary purpose is to provide typical values of part-load losses for these components when tests are not available; this was the proposed use by DOE in the energy-standard NOPR. AMCA recommends IEC Standard 61800-9-1:2023 be used in place of the vague interpolation requirements of ANSI/AMCA Standard 214-21 Annex F, specifically Section F.4.

IEC Standard 61800-9-2:2023 defines exact test requirements, including defined test points. Motor and VFD manufacturers recognize these test requirements and prefer to publish results in accordance with this standard.

DOE recognition of ANSI/AMCA Standard 214-21 Section 6.5 and Annex F, both modified to include IEC Standard 61800-9-2:2023, would encourage fan manufacturers to apply higher-technology motors and controllers and clarify communication between fan manufacturers and motor and controller manufacturers.

GFB 5: Editorial Consideration for Section 2.6

In subsections of Section 2.6 of the NOPR, DOE adopts by reference sections of ANSI/AMCA Standard 214-21. Some of these sections were not adopted by DOE in the NOPR, so this circumstance, plus the “back and forth” nature of going from one document to another to assemble a test procedure, can lead to confusion and noncompliance with the test procedure after it is amended. AMCA recommends replacing references to ANSI/AMCA Standard 214 with the content being referenced. This would provide clarity and ease implementation. For example:

2.6. Calculation based on Shaft-to-air testing for Fans with Motors and Motor Controllers.
The provisions of section 6.4 of AMCA 214–21 apply except that the instructions in section 6.4.2.4.1 of AMCA 214–21 are replaced by section 2.6.1 of this appendix, and the instructions in section 6.4.2.4.2. of AMCA 214–21 are replaced by section 2.6.2 of this appendix.

2.6.1 Motor efficiency if used in combination with a VFD.
This section replaces section 6.4.2.4.1 of AMCA 214–21 and provides methods to calculate the efficiency of the motor if it is combined with a VFD.

2.6.1.1 Motor efficiency Calculation, if used in combination with a VFD.
The efficiency of the motor if it is combined with a VFD is calculated as follows:

$$\eta_{mtr',act} = \frac{L_m}{(L_m + p'_L)}$$

Where:

$\eta_{mtr',act}$ is the actual motor efficiency if used in combination with a VFD.
L_m is the is motor load ratio calculated ~~per section 6.4.2.4.1.3 of AMCA 214–21~~ using the equations below:

$$L_m = \frac{H_{m,act}}{H_{mo}}$$

Where:
L_m is motor load ratio
$H_{m,act}$ is motor output power (kW or hp)
H_{mo} is motor nameplate output power (kW or hp)

p'_L are the relative losses of a motor if used in combination with a VFD that that exactly meets the applicable standards at § 431.25 per section 2.6.1.2. of this appendix.

GFB 6: Equipment Classes for GFB Sold with and without Motor Controllers

In the AMCA 2021 Fan-Shipment Database, 4.1 percent of 148,123 sales of stand-alone fans or fans of unknown installation type included a VFD. This supports AMCA's assertion that very few fan sales include a VFD; VFD typically are provided by a third party in other segments of the fan-buying chain.

While AMCA appreciates the efforts of DOE to create an aligned calculation methodology for motor efficiency and allow for motor-speed-controller losses, the additional complexity of both an equation-based approach and the doubling of the number of fan categories diminishes the value.

In particular, Coefficient *B* (as proposed) is a multi-step equation based on FEP_{act} (figures 20 and 21), which requires selection of a fan and determination of its fan-electrical-power (FEP) value and then calculation of a “credit.” This credit is used to calculate the *B* coefficient and FEP_{act}. Finally, the *B* coefficient is used to modify the proposed minimum FEI value to determine a new compliant FEI value that can be compared to the original-selection FEI value to determine compliance.

Table 5.5.6 FEI Levels for GFBs with Motor Controller

Fans with Motor Controller with:	FEI level for Fans with Motor Controller*
$FEP_{act} < 20$ kW (26.8 hp)	$FEI = FEI_{EL\_no\_mc} \times B$; where $B = \frac{FEP_{act} - Credit}{FEP_{act}}$; where $Credit = 0.03 \times FEP_{act} + 0.08$ **[SI]** $Credit = 0.03 \times FEP_{act} + 0.08 \times 1.341$ **[IP]**
$FEP_{act} \geq 20$ kW (26.8 hp)	$FEI_{EL\_no\_mc} \times 0.966$

*Rounded to the hundredth

FIGURE 20. Equation for valuation of *B* coefficient from technical support document.

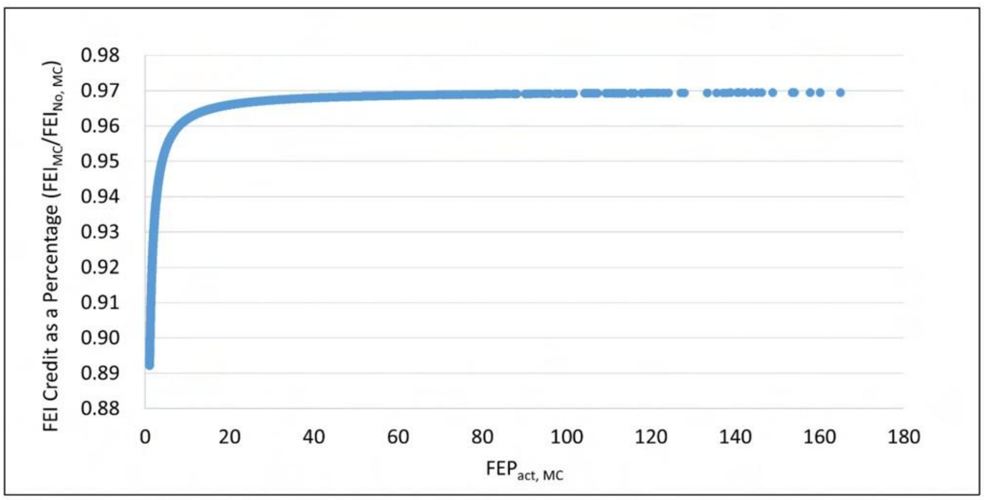


Figure 5.5.8 FEI Credit as a Percentage versus FEP_{act} for Fans Sold with a Motor Controller

FIGURE 21. Figure 5.5.8 from the technical support document.

While AMCA supports the industry's move to fans with speed controls, the complexity of the DOE's process would be inhibitive for smaller businesses and require significant selection-software computational updates for others. Additionally, the proposed process would cause confusion, as a FEI limit remains unknown until a selection is made and could lead to the perception that fans sold with controllers are less efficient than fans sold without them, which is counterproductive to saving energy. For these reasons, AMCA proposes to add the credit to the FEI value as described below, rather than lowering the FEI limit.

To simplify the process while still encouraging the use of fans with speed controls, AMCA recommends following the FEI-based fan-efficiency provisions of ANSI/ASHRAE/IES 90.1, the International Energy Conservation Code, and California Title 24, which provide a 5-percent credit for VAV applications (speed-modulated). Even in constant-speed applications, controllers enable optimization of operation and can allow adjustments from balancing and other system changes to improve fan efficiency. Such adjustments are common because static pressure in air-distribution systems tends to be higher than shown in design documents because duct-system contractors often have to accommodate unplanned obstacles and elevation changes.

The 5-percent adjustment can be achieved by applying the credit directly to the FEI value for fans sold with a controller, regardless of the methodology used to calculate FEI (i.e., wire-to-air, AEDM, or a method proposed in Section 2.6 of the energy-standard NOPR). By applying the credit directly to the FEI value instead of the FEI limit, the need for doubling the number of fan categories from nine to 18 is eliminated. This is easily accommodated in manufacturer selection software because a selected fan always will be compliant with the FEI limit after a VFD is selected. Furthermore, this approach is consistent with the widespread industry practice whereby VFD are provided by third parties instead of fan manufacturers.

In summation, AMCA recommends adding a credit to to the FEI value, not the FEI limit, consistent with the method and value used in model energy and state energy codes, for fans sold with controllers or representations whereby a controller is required to offset the controller efficiency loss factored into FEP and FEI calculations. This would reduce the proposed number of fan classes from 18 to nine, be consistent with industry practice and energy codes that have been in place since 2019 (about 10 years before the energy standard is estimated to take effect), and be more easily accommodated in manufacturer selection software.

GFB 7: FEI Levels and Radial Fans

Radial fans require high air-stream velocities and rugged impellers, which degrade efficiency capability. When FEI analysis was applied, unshrouded impellers became the lowest common denominator in the radial-fan class, compared to shrouded radial impellers. Unshrouded radial fans have great difficulty achieving an FEI level of 1.00; from a design and history perspective, achieving an FEI rating of 1.17 appears impossible.

AMCA proposes further distinguishing impeller types, which are not delineated in the LCC data or historical sales data submitted for analysis, to separate industrial-process dust-and-material-handling fans from light-duty dust-handling fans that could more closely be included in the general radial-equipment class. An exemption for unshrouded radial fans less than 30 in. in diameter is proposed. Extending this exemption to all sizes of unshrouded radial fans and considering the RIM-style (defined below) impeller as unshrouded would clearly separate radial fans with the harshest industrial-process requirements from general radial products.

To summarize, AMCA has defined a physical difference between shrouded and unshrouded radial fans based on impeller design characteristics:

- This physical difference is manifested in fan data showing clear capability separation between shrouded and unshrouded radial fans.
- There is an exemption in the DOE test-procedure final rule's scope for unshrouded fans with a diameter under 30 in. or a blade width less than 3 in. Considering AMCA has analyzed radial-fan data in this manner, now seems a reasonable time to ask that the size limitation be removed in light of the next bullet point. Because the test procedure is final, the next best option is to recommend that DOE not set an energy standard for unshrouded radial fans when the final rule is published. As such, DOE would establish that shrouded radial fans are a product class with requirements in the energy standard and define two classes of unshrouded radial fans, neither of which would have requirements in the energy standard.
- Extending the exemption for unshrouded radial fans would avoid the utility argument and allow FEI to be raised for the entire equipment class.
- The issue of regulating unshrouded radial fans could be addressed in a future regulatory review cycle.

Given these considerations, AMCA proposes the following for exempting unshrouded radial fans from the energy standard:

Unchanged from GFB test-procedure final rule:

> *Radial-housed fan* means a fan with a radial impeller in which airflow exits into a housing that is generally scroll-shaped to direct the air through a single fan outlet. Inlets and outlets can optionally be ducted.

Where suitable in the energy-standard final-rule narrative for CFR language:

> Note: These definitions are modified from the *Federal Register* test-procedure final rule, Page 27320, Footnote 26[1]:
>
> *Radial-housed-unshrouded fan* means a radial fan with impeller blades attached to a backplate and hub (i.e., open radial blade) or a hub only (i.e., open paddle wheel) and with an open front at the impeller's inlet.
>
> *Rimmed radial-unshrouded fan* means a radial-housed unshrouded fan having a vertical rim on both sides of the impeller.
>
> *Radial shrouded fan* means a radial housed fan with impeller blades attached to a backplate and a "shroud'' at the impeller's inlet. The shroud can be canted or vertical.
>
> <u>DOE does not propose energy standards for radial housed unshrouded fans or rimmed radial unshrouded fans.</u>
>
> Or, if new definitions cannot be codified without the test procedure being adjusted:

DOE does not propose energy standards for radial housed unshrouded fans, which are radial fans with impeller blades attached to a backplate and hub (i.e., open radial blade) or a hub only (i.e., open paddle wheel) and with an open front at the impeller's inlet, or rimmed radial unshrouded fans, which are radial housed fans with impeller blades attached to a backplate and to a "shroud'' at the impeller's inlet. The shroud can be canted or vertical.

---

[1] Footnote 26 of *Federal Register* Final Rule for Fans and Blowers Test Procedure: Specifically, radial housed unshrouded fans, which means a radial housed fan for which the impeller blades are attached to a backplate and hub (*i.e.,* open radial blade), or to a hub only (*i.e.,* open paddle wheel), and with an open front at the impeller's inlet. These are different from radial shrouded fans, for which the impeller blades are attached to a backplate and to a ring or ''shroud'' at the impeller's inlet.

GFB 8: Availability of Products to Meet DOE-Proposed FEI Levels

In the NOPR (Page 20.a, Section I.D), DOE states that, with regard to technological feasibility, products achieving proposed standard levels already are commercially available for all equipment classes covered by the proposal and the benefits of the proposal far exceed the burdens.

AMCA disagrees with the statement that products achieving the proposed FEI levels across all duty points required by the market already are commercially available for each equipment class.

As discussed previously, it appears DOE utilized a percentile approach, which is weighted significantly toward lower-air-power duty points, to set the proposed FEI level for each equipment class. The reference-fan shaft-power-equation coefficients (*Po* and *Qo*) supplement higher FEI values for lower-air-power fans, as shown above. However, DOE's definition of "available" does not seem to fully encompass the entire required performance range for each equipment class or characterize the complete market requirements.

AMCA maintains the proposed FEI levels would improperly restrict product availability by not accounting for the full operating duty points mandated by the market for certain equipment classes. The alternate FEI values proposed by AMCA aim to allow efficient selections while ensuring product availability across required duty-point ranges. Below are examples of how the proposed FEI levels restrict the market availability and utility of equipment classes.

Using the LCC data provided by DOE for axial panel fans, the proposed level of 1.48 (Figure 22) provides no currently available duty points above ~4 hp. The 1.48 significantly reduces the utility of this equipment class and likely creates unintended consequences, such as equipment-class substitution.

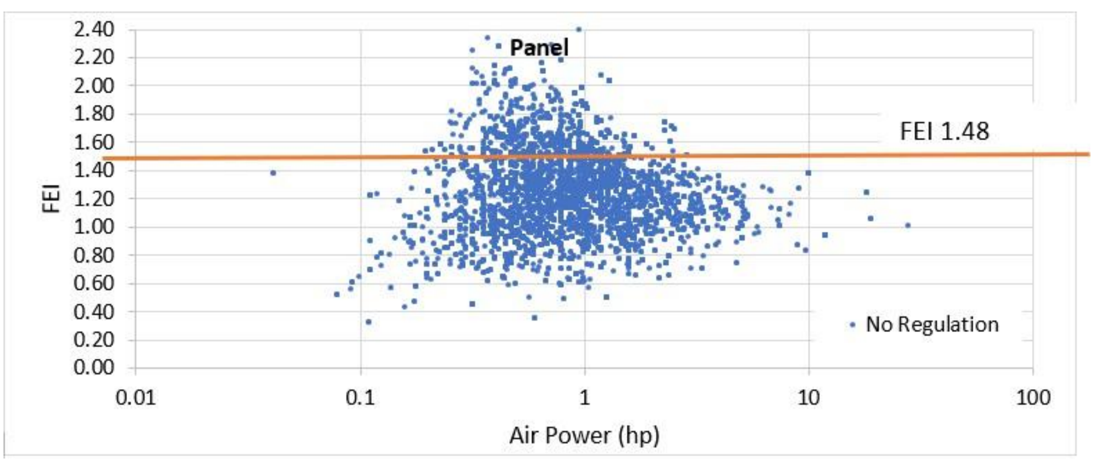


FIGURE 22. Axial-panel-fan air power vs. FEI.

The centrifugal-housed equipment class using similar LCC data shows minimal duty points above 50 hp (Figure 23).

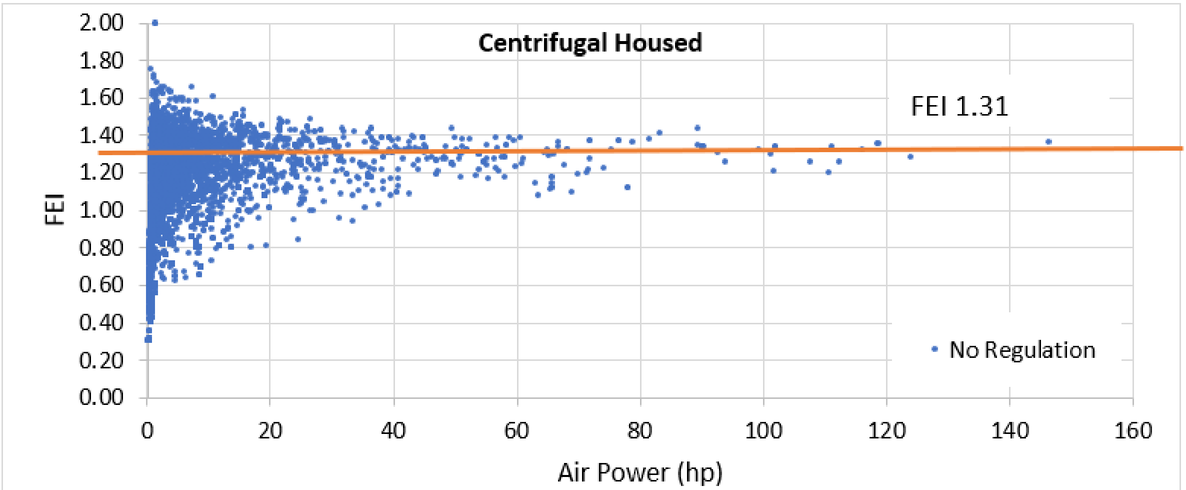


FIGURE 23. Centrifugal-housed air power vs. FEI.

As discussed previously, the centrifugal-housed selections already are within 15 percentage points of peak efficiency for the equipment class, indicating good FEI selections for the applicable duty point (Figure 24).

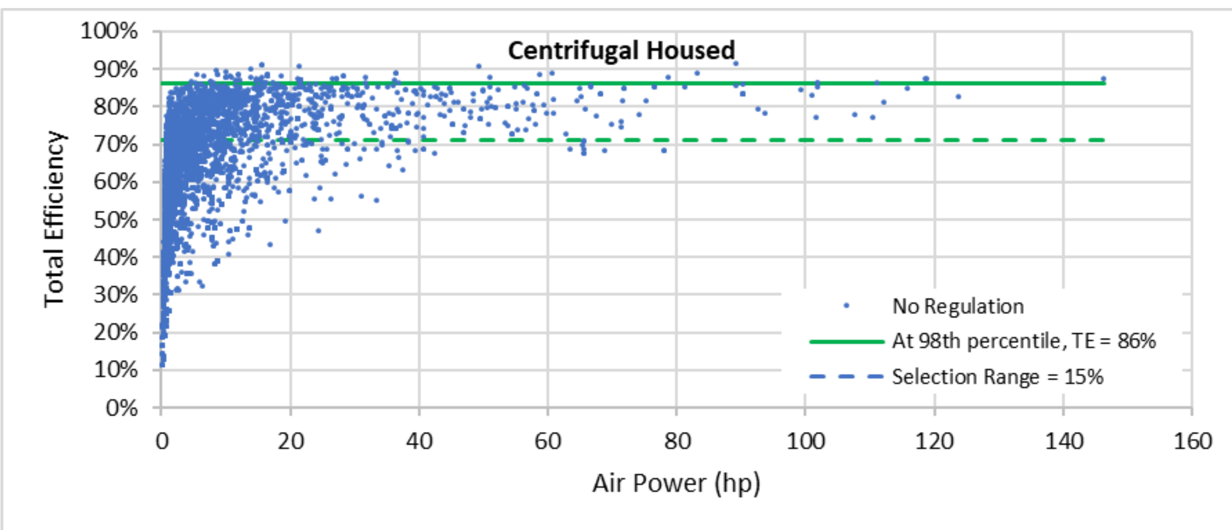


FIGURE 24. Centrifugal-housed air power vs. total efficiency with selection range.

Consistent with AMCA's position and rationale in GFB 2, AMCA recommends DOE re-evaluate the statement, "Products achieving these standard levels are already commercially available for all equipment classes covered by this proposal," and consider the entire performance range of the equipment class when proposing FEI levels. If this is not done, equipment classes will see loss of utility within the current performance range.

GFB 9: Accommodation of Variances in Airflow and Pressure Measurements During Certified-Performance Check Tests

In the NOPR (Page 147.c, Section V.E.2.a), DOE states:

“When testing a single fan at multiple duty points, DOE proposes to first determine either bhp or FEP, dependent on the test method specified by the manufacturer, for the range of certified airflow, pressure, and speed (duty points) according to appendix A of subpart J to 10 CFR part 431. DOE acknowledges that it may not be feasible to exactly replicate the measurements at the certified duty points, or within the certified range of duty points; therefore, DOE will verify that, at a given speed, the airflow at which the test is being conducted is within 5-percent of the certified airflow and the pressure is within between $P \times (1 - 0.05)^2$ and where P is the certified static or total Pressure.”

AMCA supports the inclusion of these allowable variances in airflow and pressure measurement during a certified-performance check test. However, it believes the text should be corrected to read as follows:

“… DOE will verify that, at a given speed, the airflow at which the test is being conducted is within 5-percent of the certified airflow and the pressure, i.e. P is within $P \times (1 - 0.05)^2$ and $P \times (1 + 0.05)^2$ where P is the certified static or total pressure.”

For reference, this is an implementation of the allowable airflow and pressure variances in AMCA Publication 211-22 (Rev. 01-23), *Certified Ratings Program Product Rating Manual for Fan Air Performance*. This variation is allowed to accommodate variances in flow and pressure at a specified speed; however, because flow and pressure are corrected to the rated values along with the measured power value, AMCA is hesitant to associate these variances with power or efficiency tolerances.

The corrections are shown mathematically and graphically in figures 25 and 26.

For periodic check tests, the airflow tolerance is 5%.

$T_Q = 5\%$

For precertification check tests where the same fan was tested at two different labs, the airflow tolerance is reduced to 3%.

$T_Q = 3\%$

The low limit for airflow is:

$$Q_{test} \geq Q \cdot (1 - T_Q) \quad \text{Eq. 10.1}$$

The low limit for pressure is:

$$P_{s,test} \geq P_s \cdot (1 - T_Q)^2 \quad \text{or} \quad P_{t,test} \geq P_t \cdot (1 - T_Q)^2 \quad \text{Eq. 10.2}$$

The high limit for airflow is:

$$Q_{test} \leq Q \cdot (1 + T_Q) \quad \text{Eq. 10.3}$$

The high limit for pressure is:

$$P_{s,test} \leq P_s \cdot (1 + T_Q)^2 \text{or } P_{t,test} \leq P_t \cdot (1 + T_Q)^2 \quad \text{Eq. 10.4}$$

Where:

T_Q is the airflow tolerance
Q is rated catalog fan airflow rate, or originally tested airflow for a precertification test
P_s is rated catalog fan static pressure, or originally tested fan static pressure for a precertification test
P_t is rated catalog fan total pressure, or originally tested fan total pressure for a precertification test
Q_{test} is the check test airflow
$P_{s,test}$ is the check test fan static pressure
$P_{t,test}$ is the check test fan total pressure

The check test performance shall not be outside the low and high tolerance limits (curve or points) over the performance range cataloged (see Figure 10.1).

FIGURE 25. Excerpt on airflow and pressure tolerances during check tests from AMCA Publication 211-22 (Rev. 01-23).

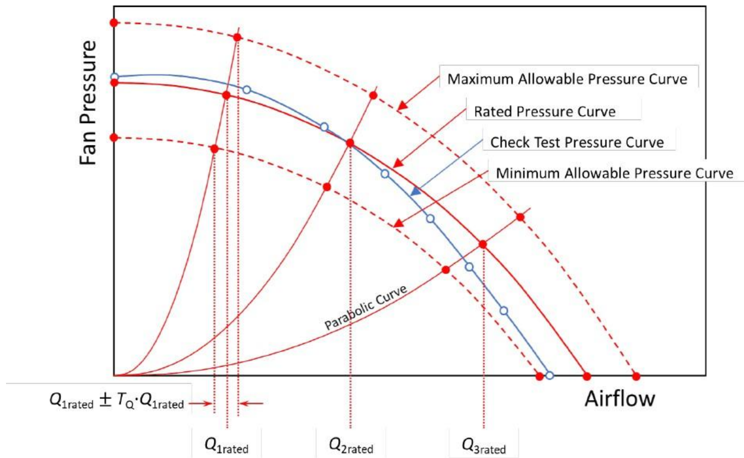


Figure 10.1 — Application of Airflow Tolerance

FIGURE 26. Excerpt on airflow and pressure variation during check tests from AMCA Publication 211-22 (Rev. 01-23).

GFB 10: Verification of Certification Duty Points

In the NOPR (Page 147.c, Section V.E.2.a), DOE states:

“If DOE is unable to verify some or all certified duty points (i.e., the fan is unable to perform at airflows and pressures at a given speed that are within the prescribed margin of the certified airflows and pressures), the certified rating cannot be used to determine compliance. DOE will consider the certified rating to be invalid and DOE will rely on the measured duty point (i.e., measured flow and pressure at the given speed) to determine compliance.”

AMCA recommends that, for all conditions, DOE correct the test point by using the fan laws to intersect with the rating curve to calculate power, FEP, and FEI values. In the process, the 5-percent allowable variance likely would not be required by DOE for calculation of compliance. AMCA likely would continue to use the 5-percent allowable variance in its Certified Ratings Program to ensure customers receive allowable pressure and flow performance in addition to claimed power consumption for fan energy output.

In the situation DOE describes, at the speed specified, the test values do not lie within the allowable variance. An example of such a condition is shown in Figure 27.

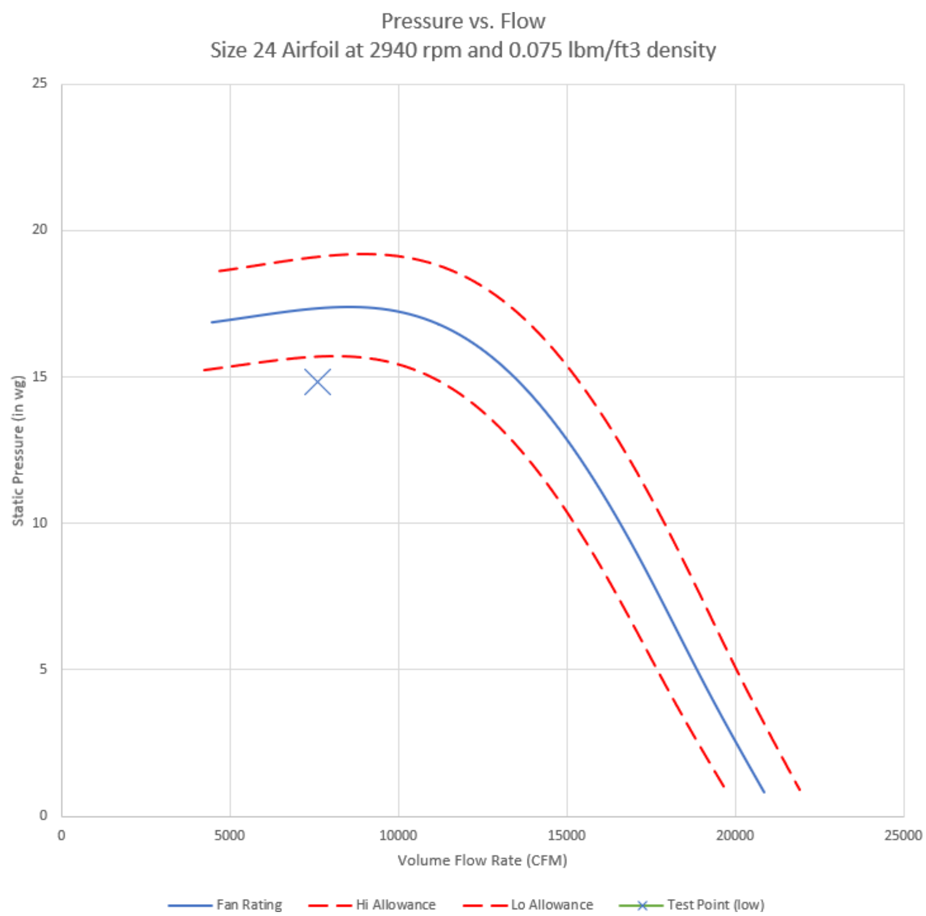


FIGURE 27. A duty point falling outside 5 percent on flow and 5 percent (squared) on pressure.

As written, the NOPR could be used to calculate FEI at the test condition. Two challenges, however, exist. First, FEI values should be compared only at similar duty points. Thus, the requirement to make the fan perform at the specified duty point is simulated by, in the condition shown, speeding up the fan to drive the measured point onto the rating curve. This process happens in reverse for duty points above the rating curve. In this case, the fan would be simulated as being slowed down to drive the duty point onto the rating curve.

In both cases—the fan performing below the rated flow-pressure curve or the fan performing above the rated flow-pressure curve—the fan laws would be employed to calculate the resulting power at the duty point on the fan rating curve.

The second challenge is this: Calculating FEP and FEI at the test point would not represent the manufacturer’s claim of operation of the fan at the specified speed. There is no indication that another supposedly identical fan operating at the same speed would not operate closer to the rated fan curve.

Consequently, AMCA recommends that, for all conditions, the test point be corrected using the fan laws to intersect with the rating curve for calculations of power, FEP, and FEI. This is illustrated in Figure 28.

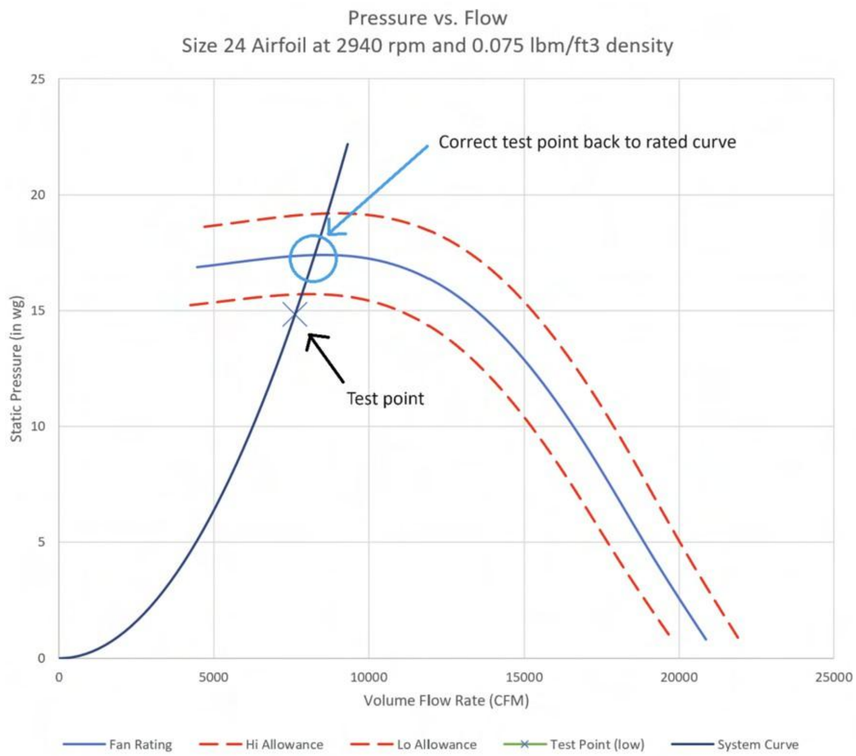


FIGURE 28. Same duty point as Figure 27, corrected to the rating speed for surveillance.

SECTION 2: COMMENTS ON AIR-CIRCULATING FANS (ACF)

ACF 1: DOE's Estimated Manufacturer's Production Cost (MPC)

AMCA does not believe DOE's estimated MPC for each cost level is accurate and underestimates or overestimates the cost delta for each efficiency level (EL).

In general, AMCA recommends DOE reconvene with NEMA motor manufacturers to update the MPC financial analysis, which appears to be outdated. For example, AMCA believes the actual pricing difference is roughly 300 percent higher for the 24-in.-diameter ACF EL0 motor and 80 percent higher for the 54-in.-diameter ACF motor.

In addition to outdated cost data, there is no cost change for eliminating belt drives (EL0 to EL1). It is AMCA's belief that EL1 is flawed. A belt drive is used to transform the synchronous speed of a motor to the operating speed of fan blades. If a belt drive is eliminated in moving from EL0 to EL1, then a higher-pole-count motor likely will be associated with EL1. Higher-pole-count motors generally are more expensive. AMCA recommends DOE work with NEMA to evaluate the costs associated with moving from a 4-pole (~1,800 rpm) motor to a 6-pole (~1,200 rpm) motor or an 8-pole (~750 rpm) motor in EL1.

More generally, the proposed values appear to underestimate the cost for each motor technology. In addition to the costs being lower than would be expected, it should be noted that industry sells into both the commercial and industrial fans-and-blowers markets. While commercial fans and blowers are significantly more cost-sensitive and cost-constrained, industrial fans and blowers, which often are more robust, generally are more expensive.

ACF 2: ACF Performance Data and Analyses

DOE used a combination of Bioenvironmental and Structural Systems Laboratory (BESS Lab) ACF test data and manufacturers' catalog data for its analysis. AMCA does not believe the BESS Lab data is characteristic of the current ACF marketplace (skewed to premium efficiency) and that catalog data can accurately be converted into ANSI/AMCA Standard 230-23, *Laboratory Methods of Testing Air Circulating Fans for Rating and Certification*/DOE test-procedure data. Additionally, AMCA believes DOE has ignored critical characteristics of ACF within the dataset.

Per the BESS Lab website: "The Bioenvironmental and Structural System (BESS) Laboratory is a research, product-testing and educational laboratory. The lab provides unbiased engineering data to aid in the selection and design of agricultural buildings and assists equipment manufacturers in developing better products." As noted by DOE, certified test data from either BESS Lab or AMCA is required for ACF efficiency rebates from many U.S. utility companies. So, while the diameter and airflow data contained in the BESS Lab online database may be considered representative for the ACF industry, the efficiency levels are skewed very high. DOE acknowledges this in the energy-standard NOPR: “DOE acknowledges that the BESS Labs combined database likely contains higher efficiency fans than the overall ACF market, since many agricultural incentive programs require that fans be tested at BESS Labs and meet certain performance requirements” (Page 3758, *Federal Register*, Vol. 89, No. 13, Jan. 19, 2024). Despite acknowledging the skewed nature of the BESS Lab data, DOE’s proposed EL4 fails over 40 percent of the most efficient ACF on the market.

Figure 29 shows the distribution of high-speed airflow for the 500-plus fans in the BESS Lab database. AMCA notes significant variation by diameter.

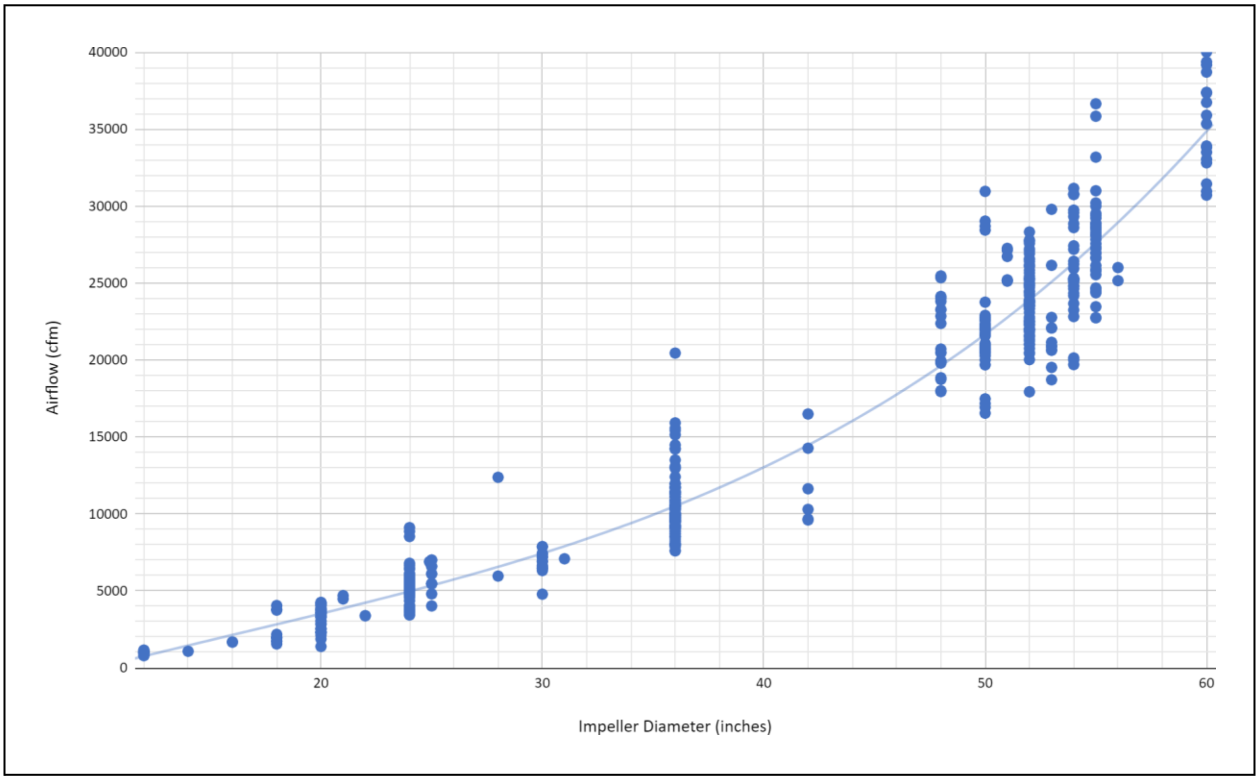


FIGURE 29. Distribution of maximum airflow vs. impeller diameter. Source: BESS Lab

Because of the nonlinear relationship between airflow and power, DOE’s proposed binned cfm/W minimum efficacies (Table 4) create a highly inequitable and gameable scenario. Manufacturers can lower maximum speed or decrease impeller pitch to achieve the current binned cfm/W efficacy levels. This will sacrifice consumer utility but create a compliant fan without requiring the use of more expensive and efficient components. Additionally, the binning of efficiencies encourages manufacturers to shift to the largest diameter in the lowest efficiency bins. For example, current fans with 36-in.-diameter impellers can be made compliant more easily by changing the impeller diameter to 35.4 in., which dramatically reduces the compliance hurdle and would count towards quads saved in DOE’s analysis without actually saving significant real-world energy.

Impeller-Diameter Range	DOE EL 4 Minimum Efficacy	Diameter Selected for Baseline Fan
12” to <36”	12.2 cfm/W	24”
36” to <48”	17.3 cfm/W	36”
≥48”	21.5 cfm/W	52”

TABLE 4. DOE’s binned minimum efficacy at EL4 and baseline fan diameters.

With regard to inequity, Table 5 shows that a fixed minimum cfm/W for a given diameter provides inequitable wire-to-air efficiencies when fans of the same diameter move different volumes of air.

Impeller Diameter	DOE EL4 Minimum Efficacy	DOE EL4 Airflow and Wire-to-Air Efficiency	BESS Lab Minimum Airflow and Wire-to-Air Efficiency	BESS Lab Maximum Airflow and Wire-to-Air Efficiency
24”	12.2 cfm/W	3,792 cfm (13.0%)	3,450 cfm (10.8%)	9,100 cfm (74.9%)
36”	17.3 cfm/W	8,421 cfm (18.0%)	7,600 cfm (14.3%)	15,900 cfm (64.0%)
52”	21.5 cfm/W	20,684 cfm (30.1%)	17,950 cfm (23.3%)	28,300 cfm (57.9%)

TABLE 5. Required minimum wire-to-air efficiency to achieve EL4 cfm/W at BESS Lab minimum and maximum airflow.

For comparison, a high-airflow, 36-in. ACF would require a wire-to-air efficiency of 64 percent at the cfm/W specified in EL4. This is more than double the 31.2-percent efficiency of the 36-in. EL6 baseline ACF shown in Table 6. As written, the current binned minimum efficacies will eliminate products with a high airflow for their relative size.

Efficiency Level	Summary of EL Logic	W-T-A Efficiency
EL0	Baseline: low-efficiency PSC motor + belt drive	5.30%
EL1	Direct drive	6.03%
EL2	High-efficiency PSC motor	6.63%
EL3	Aero Redesign 1: EL2 + (EL4-EL2)*0.33	10.35%
EL4	Aero Redesign 2: Ag rebate efficiency levels	17.96%
EL5	Aero Redesign 3: Max. efficiency in ACF database	26.27%
EL6	EC motor (20% better than IE4)	31.20%

TABLE 6. DOE efficiency levels and resulting wire-to-air efficiency for baseline 36-in. ACF.

The BESS Lab online database includes information on whether an ACF is guarded or unguarded. Fan guards are a critical, mandatory safety feature of fans for many applications. Figure 30 shows the discrepancy in wire-to-air efficiency for guarded vs. unguarded ACF. The average unguarded ACF has a wire-to-air efficiency of 42.6 percent, while the average guarded ACF has an efficiency of 21.8 percent. Because the current regulation does not differentiate between unguarded and guarded ACF, DOE is providing a free pass to most unguarded fans, while penalizing guarded fans for having a safety feature mandated by the Occupational Safety and Health Administration (OSHA) and other product-safety standards.

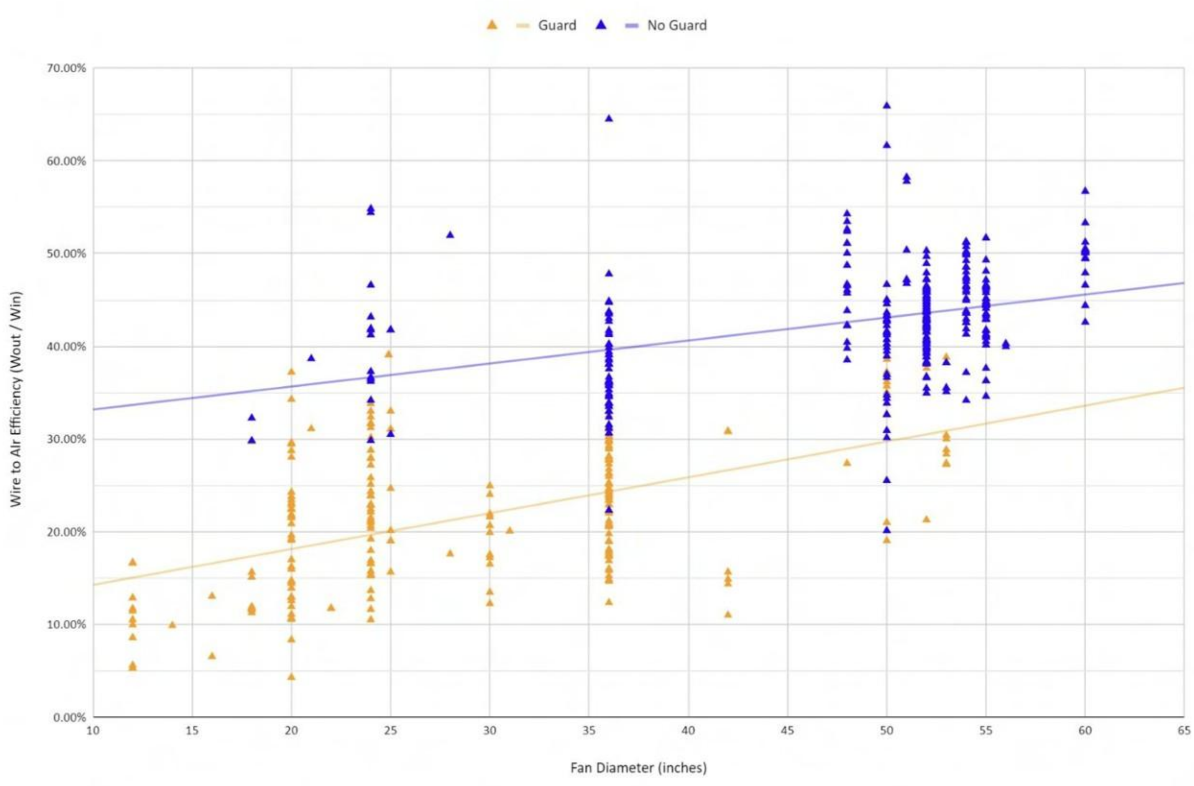


FIGURE 30. Wire-to-air efficiency vs. diameter, guarded (blue) and unguarded (orange) ACF.

To create a more equitable minimum efficiency, AMCA proposes DOE eliminate the efficiency bins and utilize a minimum efficiency level that changes with impeller diameter. Additionally, AMCA proposes that DOE account for the amount of work (output air power) a fan performs by factoring in wire-to-air efficiency to determine a fan's minimum efficacy (cfm/W) and account for the impact of a fan guard on fan performance. For additional details, see the next comment.

ACF 3: DOE's Proposed Efficiency Bins and Levels

In the NOPR, DOE proposes three bins of minimum efficacy levels (Figure 31).

Equipment Class*	Efficacy at Maximum Speed (CFM/W)
Axial ACFs; 12 inches ≤ D < 36 inches	12.2
Axial ACFs; 36 inches ≤ D < 48 inches	17.3
Axial ACFs; 48 inches ≤ D	21.5
Housed Centrifugal ACFs	N/A

*D: Diameter in inches
N/A: Not applicable; DOE is not proposing to set a standard for this equipment class.

FIGURE 31. DOE-proposed bins of minimum efficacy.

AMCA believes the proposed minimum-efficiency strategy will encourage gaming via speed reduction, impeller de-pitching, and changes in impeller diameter. Additionally, there is no correlation between the minimum levels proposed by DOE and the wire-to-air efficiency of fans in the real world. AMCA does not believe the airflows analyzed by DOE accurately characterize the range of airflows available on the market today.

AMCA believes the purpose of an ACF is to generate elevated air speed across (a) target object(s). This air speed increases the rate of heat transfer from the object to the surrounding air. The utility of an ACF is its ability to create this elevated air speed. For human thermal comfort, air speeds of 100 fpm to 400 fpm are common design practice. Agricultural requirements depend on the animal; in poultry applications, air speeds of 600 fpm to more than 700 fpm are recommended, while, in dairy applications, air speeds of 400 fpm to more than 600 fpm are common. To be economical, these air speeds need to be delivered over a significant area, not just at a single point. ACF can provide cooling via increased heat loss at an energy density of a fraction of a Watt per square foot (~0.2 W/sq ft), as compared to spot cooling with air conditioning, which is dramatically more energy-intensive (~3 W/ sq ft).

For comparative purposes, average fan exit air speed can be approximated by dividing airflow by fan outlet area or fan discharge area, as appropriate. BESS Lab also directly measures "5D centerline velocity," which: "refers to the air velocity at a single point along the center axis of the fan blade, downstream from the fan five times the propeller diameter. For example, velocities are measured 15 feet downstream for a 36" (3ft) diameter fan" (http://bess.illinois.edu/selcritc.html).

In Figure 32, the wire-to-air efficiency of more than 500 ACF is plotted against the 5D centerline velocity. The fans are grouped into two categories: The green fans meet or exceed DOE's proposed (EL4) minimum efficiency, and the red fans fail to meet DOE's proposed minimum efficiency. The chart clearly shows the proposed energy-conservation standard does not eliminate the least efficient fans on the market. Additionally, the ACF that provide the highest utility have a disproportionately high failure rate. ACF with 5D centerline velocity above 1,250 fpm have a failure rate of nearly 70 percent.

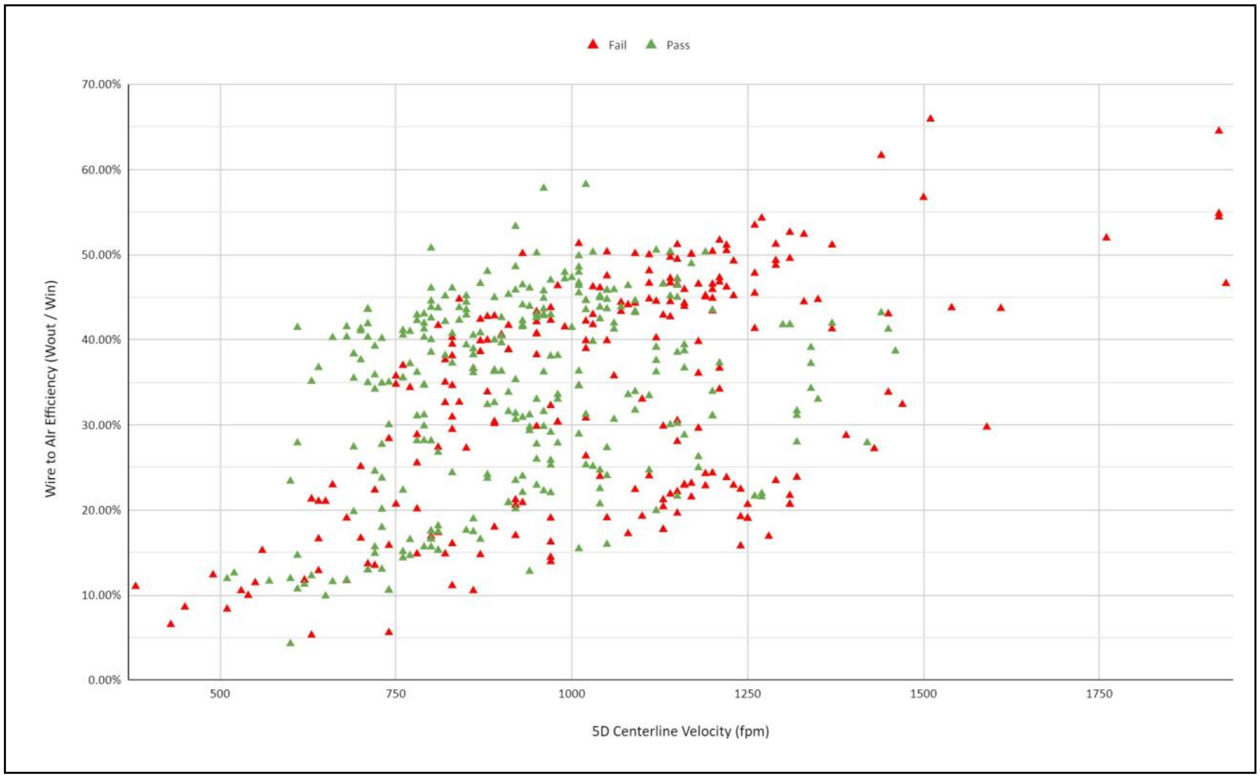


FIGURE 32. Wire-to-air efficiency vs. 5D centerline velocity, DOE EC minimums.

The 44.9-percent failure rate of fans with wire-to-air efficiencies greater than 40 percent is virtually the same as the pass rate (47.2 percent) of fans with efficiencies less than 20 percent. Again, this indicates the pass/fail rate of the proposed regulation has little relationship to ACF efficiency and is unlikely to save energy in the real world.

A similar random distribution of pass/fail for DOE's EL4 minimum cfm/W is seen when comparing wire-to-air efficiency to fan-impeller diameter using the BESS Lab database. In Figure 33, the trend lines for ACF that fail actually is slightly higher than the trend line for fans that pass, based on the proposed efficiency bins.

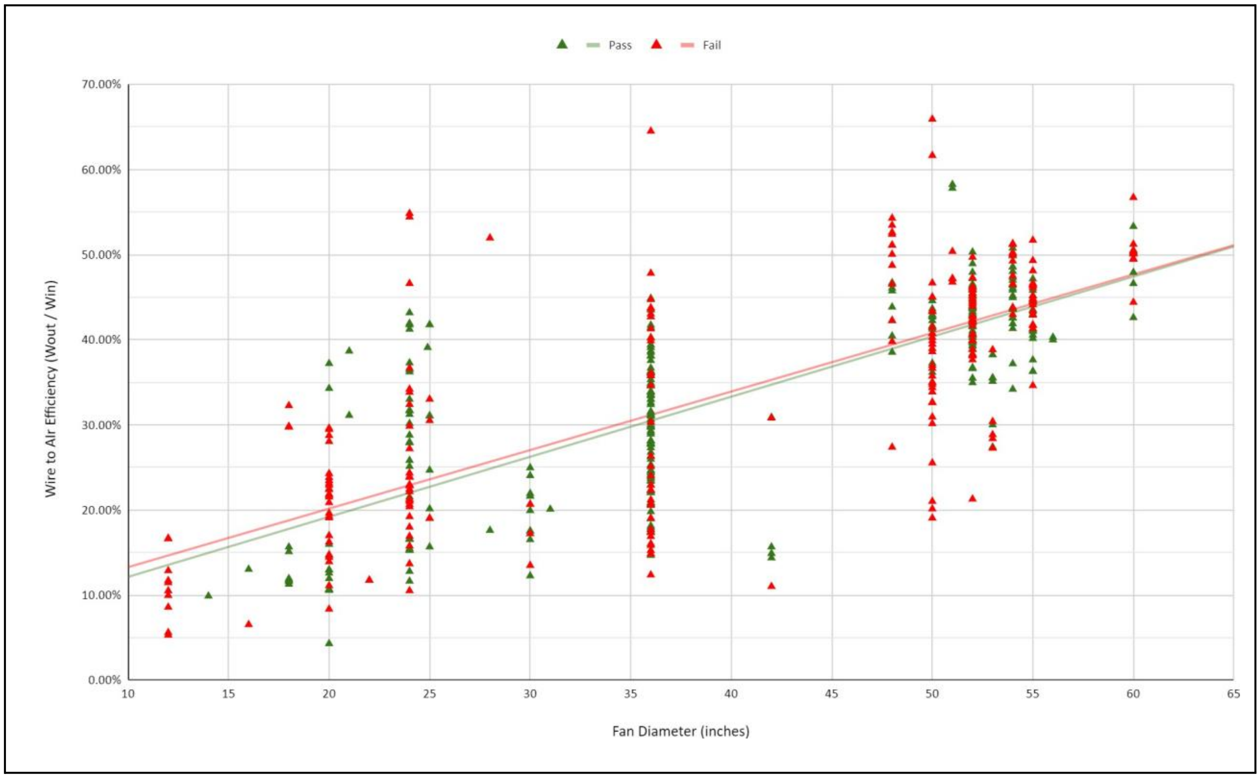


FIGURE 33. Wire-to-air efficiency vs. fan diameter, DOE EC minimums.

To increase the efficiency of ACF and save energy, AMCA proposes replacing the binned cfm/W method with Equation 1. Equation 1 was derived from circulating-fan-efficacy equation EQ 8.16 in ANSI/AMCA Standard 230-23 and equations for overall fan efficiency, output air power, fan total pressure, and fan swept area. The derivation for Equation 1 is contained in Appendix B.

$$Eff_{circ} \approx \frac{4066 * D^4 * \eta}{Q_0^2} \qquad \text{Equation 1}$$

Where:
Eff_{circ} = minimum ACF efficacy (cfm/W)
D = fan (impeller) diameter* (inches)
η = fan wire-to-air efficiency (air power / input power) (percent)
η = X * D + Y
Q = fan airflow (cfm)

*Diameter is impeller diameter for unhoused fans or the lesser value of impeller diameter and equivalent diameter for housed fans. Per ANSI/AMCA Standard 99, *Standards Handbook*, equivalent diameter means the diameter of a circle having the same area as another geometric shape. For a rectangular cross section having width *a* and height *b*, equivalent diameter is given by D_e = (4*ab*/π)^0.5.

Applying to Equation 1 a minimum wire-to-air efficiency that increases based on fan diameter yields a much more logical distribution of fans that comply vs. fans that fail, as shown in Figure 34. Note that the pass/failure in Figure 34 is a proof of concept, not based on a suggested minimum efficiency level.

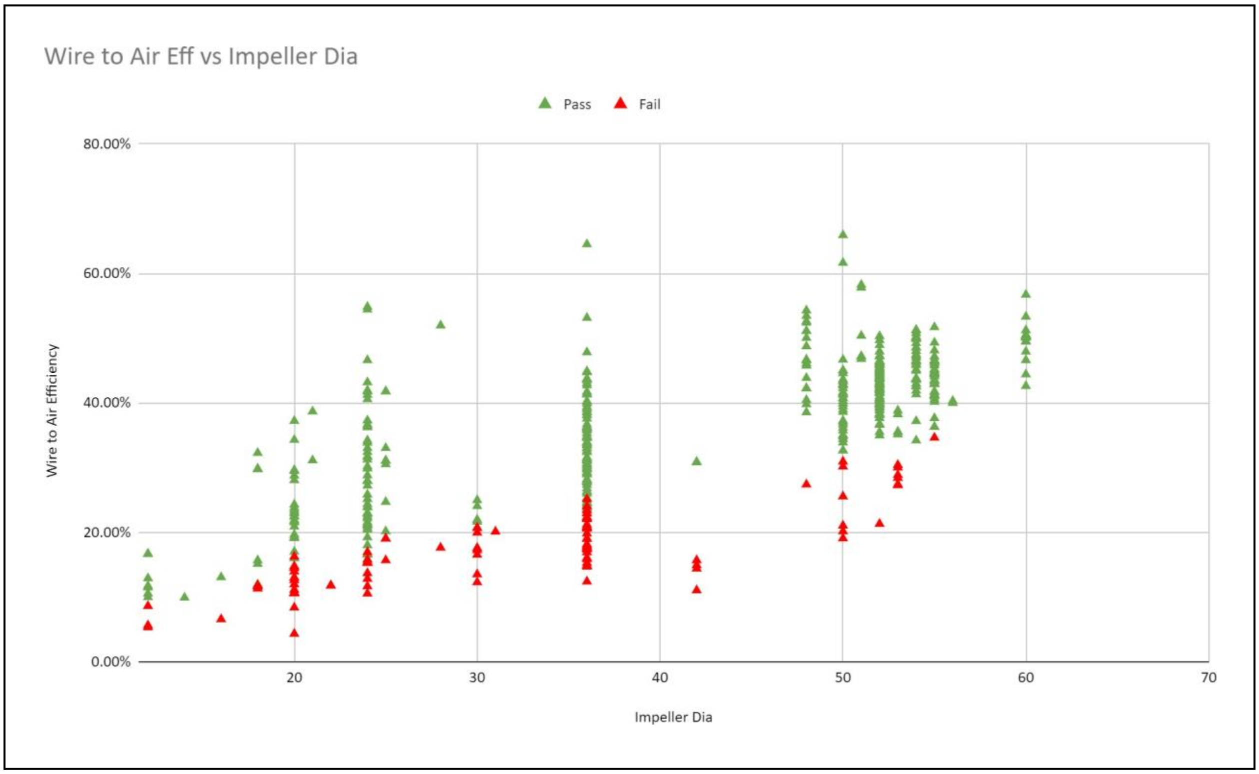


FIGURE 34. Wire-to-air efficiency vs. fan diameter, Equation 1.

In Figure 35, wire-to-air efficiency is again plotted against the 5D centerline velocity, but Equation 1 is substituted for the DOE efficiency bins. Figure 35 clearly shows the elimination of the least efficient fans on the market. Additionally, the fans at all 5D centerline velocities, including the highest velocities, remain, provided the fan is sufficiently efficient. Note that the pass/failure in Figure 35 is a proof of concept, not based on a suggested minimum efficiency level.

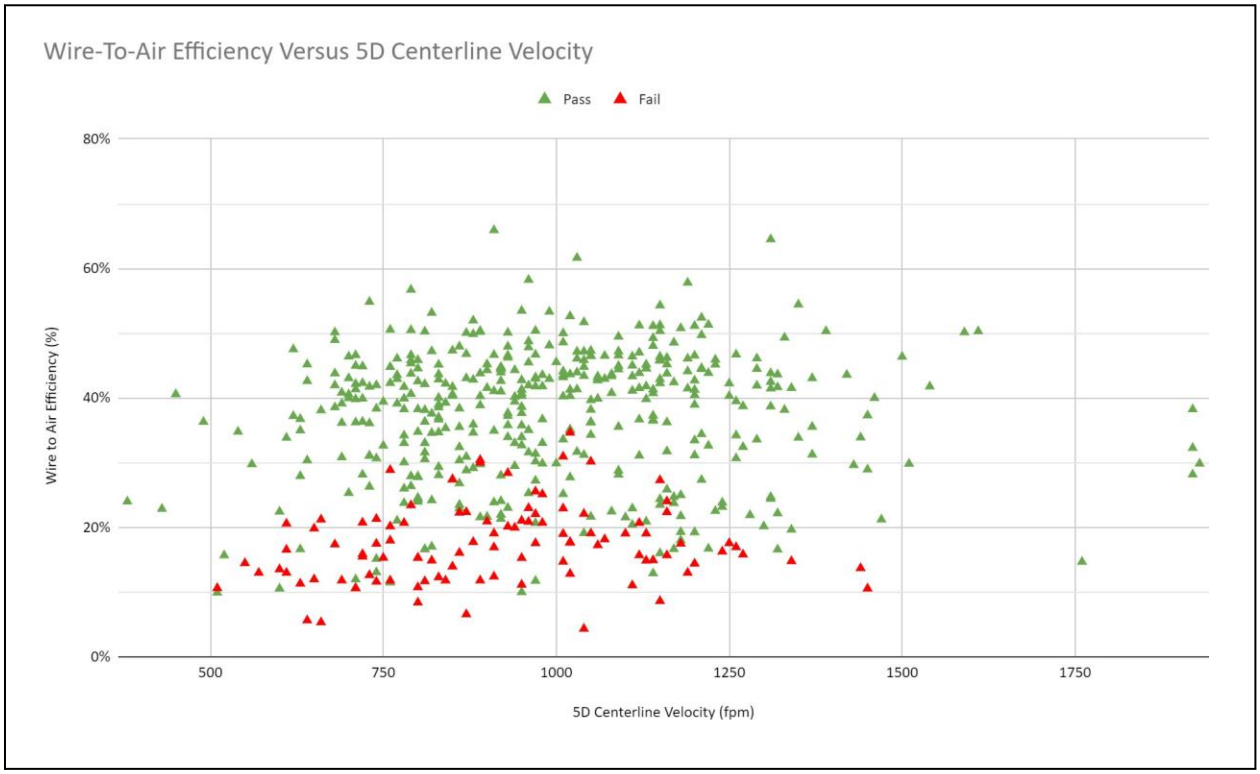


FIGURE 35. Wire-to-air efficiency vs. 5D centerline velocity, Equation 1.

Figures 34 and 35 are conceptual because DOE's proposed regulation does not address the reality that some ACF applications require guards and some do not.

Fan guards are used to protect occupants (man or beast) from the moving parts of a fan. There are several types of guards commonly used on ACF. For example, guards are used on fans in commercial and industrial buildings to protect workers' fingers from contact with fan blades. Fans in agricultural settings typically are designed with guards that provide some protection to occupants but often have wider openings to prevent clogging by objects such as feathers. Table 7 summarizes typical ACF guards by application.

Application	**Maximum Guard Spacing**	**Reference**
Workplace	0.5 in.	OSHA 1910.212(a)(5)
Residential/consumer	~1.0 in.	UL 507 Section 9.2
Residential/consumer	~1.25 in.	UL 507 Section 9.3
General agriculture	~1.5 to 2.0 in.	Typical Agricultural

TABLE 7. Summary of common spacing sizes and referenced standard for guard spacing.

Figure 36 provides a visual representation of the difference in level of obstruction provided by three common fan-guard spacings.

FIGURE 36. Left to right: 0.5 in., 1.0 in., and 2-in. guard spacing, ½-in. OSHA guard (http://bess.illinois.edu/pdf/c21003c.pdf), 1-in. UL 507 guard (http://bess.illinois.edu/pdf/c21178.pdf), and 2-in. agricultural fan guard (http://bess.illinois.edu/pdf/c21009.pdf).

Impact of Guards on ACF Performance

To evaluate the impact of fan guards on fan efficiency, 24-in. and 36-in. housed ACF heads were tested under four scenarios: (1) no guard, (2) guard with 2.0-in. (ag) spacing, (3) guard with 1.0-in. (UL 507) spacing, and (4) guard with 0.5-in. (OSHA) spacing. Table 8 summarizes the eight performance tests.

Impeller Diameter (in.)	Guard Spacing (in.)	Airflow (cfm)	Input Power (W)	Wire-to-Air Efficiency (%)	Change vs. No Guard
24	No guard	5,400	311	37.5%	-
24	2.0	5,139	312	32.3%	-5.3%
24	1.0	5,073	317	30.5%	-7.0%
24	0.5	4,823	317	26.2%	-11.3%
36	No guard	11,542	598	37.7%	-
36	2.0	10,843	600	31.1%	-6.6%
36	1.0	10,652	597	29.7%	-8.0%
36	0.5	9,866	596	23.6%	-14.1%

TABLE 8. Impact of various fan-guard spacings on wire-to-air efficiency.

The impact on efficiency was significant for all three guards tested. To contextualize the impact of the guard on the wire-to-air efficiency of the 36-in. air-circulator fan, the absolute value of the change in efficiency from adding an OSHA-compliant guard (14.1 percent) is greater than the efficiency delta between EL0 and EL4 (12.7 percent).

The BESS Lab online database provides further insight. Only two guarded ACF in the BESS Lab database have wire-to-air efficiencies above 40 percent, and both fans have problematic data. Test 19342, when imported from the BESS Lab online database to a spreadsheet, has a wire-to-air efficiency of approximately 55 percent. Test 19342 imports with an incorrect thrust value (14.7 lbf vs. 8.25 lbf, per the individual test report). Test C20136 has a wire-to-air efficiency of 41 percent. Test C20136 lists a 24-in. impeller when the actual impeller is 24.9 in., per the individual test report. When these tests are corrected, zero guarded ACF in the BESS Lab online database have a wire-to-air efficiency greater than 40 percent. In contrast, the average unguarded ACF 36 in. or larger has an efficiency of 40 percent or greater.

To see if air speed changes the impact of a guard on wire-to-air efficiency, the 36-in. fan from Table 8 was tested at various operating speeds without a guard and with the 0.5-in. guard. Table 9 summarizes the results of that testing, which show a fairly small variation from 100-percent to 60-percent speed. At very low (40 percent of maximum) airflow, the impact of the guard on efficiency began to tail off.

Impeller Diameter (in.) and Operating Speed (%)	Airflow, No Guard (cfm)	Airflow, 0.5-in. Guard (cfm)	Efficiency, No Guard (%)	Efficiency, 0.5-in. Guard (%)	Efficiency Change vs. No Guard (%)
36 in., 100%	11,542	9,866	37.7%	23.6%	-14.1%
36 in., 80%	9,206	7,825	37.4%	23.2%	-14.3%
36 in., 60%	6,793	5,831	35.2%	22.5%	-12.7%
36 in., 40%	4,452	3,937	28.7%	19.3%	-9.4%

TABLE 9. Impact of various fan-guard spacings on wire-to-air efficiency.

Use of ACF and Retention of Utility

As previously stated, while human cooling with air circulation typically is done at lower air speeds (~100 to ~400 fpm across the human body), achieving higher air speeds is critical for many agricultural applications. For poultry applications, air speeds across animals of 600 fpm to more than 700 fpm are common design requirements. Air speeds of 400 fpm to more than 600 fpm are common for dairy applications. Some recent agricultural design recommendations are nearing 1,000 fpm. AMCA recommends that DOE contact either the Biosystems & Agricultural Engineering department at the University of Kentucky or the Agricultural and Biological Engineering department at the University of Illinois for additional details on air movement requirements in agricultural applications

Regardless of the application, to be practical from both a first-cost and an operational-cost standpoint, circulating fans generating required air speeds need to cover as large of a floor area as possible. Because of the wide variety of applications, the current range of air speeds offered by ACF must be maintained or substitution to higher energy intensity technologies is likely to occur.

Regulatory Options

As fan guards significantly impact the performance of ACF and, in certain applications (OSHA, etc.), are required, an equitable regulation needs to account for fan guards. For fan guards to be included in the energy-conservation standard, a definition is needed to limit potential gaming. A proposed definition of fan guard is shown below. AMCA also suggests DOE refer to OSHA's definition of a guard in 29 CFR 1910.212(a)(5) for further refinement of the guard definition.

> *Fan guard* means a physical barrier that envelops fan blades, with openings that allow air to pass through while preventing contact with moving parts of the fan. For the purpose of this definition, energy impacts shall only be considered for fan guards of ½ in., as required by OSHA; 1 in., as required by UL 507; or 2 in. There shall be no considerations for fan-guard spacings (as shown below) larger than 2 in.

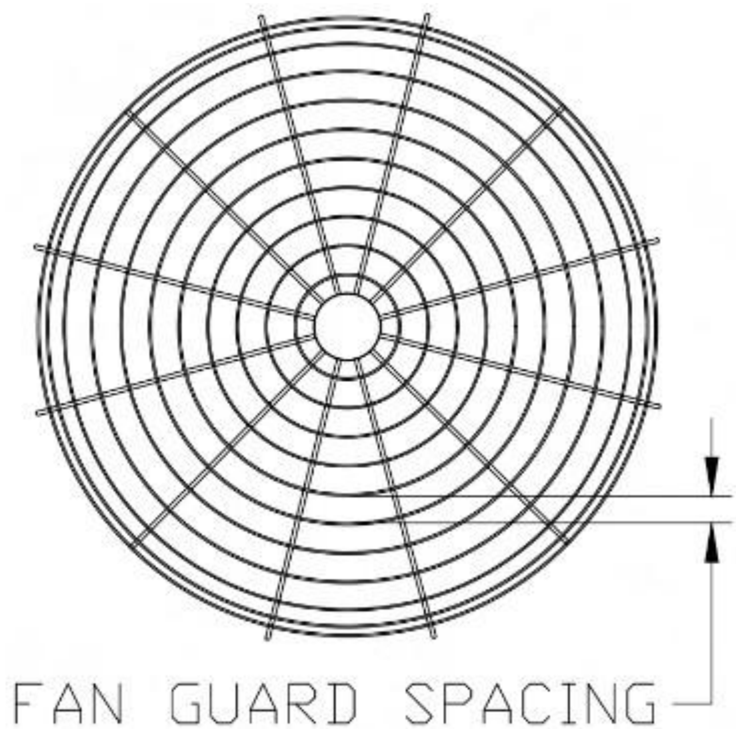


FIGURE 36-1. Illustration of fan-guard spacing.

With “fan guard” defined, the next question is how to incorporate the impact of fan guards into the proposed energy standard.

Option 1: Exemption
Exempt guarded fans in a manner similar to safety fans. Because of the limited time available to fully analyze the impact on fan performance for a critical safety feature, this option would ensure that ACF that meet current safety requirements are available to the market. All fans with OSHA-compliant and UL 507-compliant cages could be exempted from the energy-conservation standard.

Option 2: Credit
A single minimum efficiency can be specified for all ACF and credit can be given for ACF with certain guards. For example, if the equation below were used as the basis for ACF, whatever value or equation for η was utilized could be adjusted to account for the impact of a guard, similar to the motor-controller credit for GFB.

$$Eff_{circ} \approx \frac{4066 * D^4 * \eta}{Q_0^2}$$

Where:
Eff_{circ} = ACF efficiency, cfm/W
Q_0 = fan airflow rate, cfm
η = wire-to-air efficiency, unitless
D = fan-impeller diameter or equivalent diameter, inches

Table 10 shows potential credits allowed for common guard types identified in Table 7.

Maximum Guard Opening	**η Credit**
0.5 in.	12.7%
1.0 in.	7.5%
1.5 in.	6.7%
2.0 in.	5.9%

TABLE 10. Efficiency allowance for fan guards.

Mathematically, Table 10 could be approximated with an equation and used with Equation 1 above or Equation 3 below to create a more equitable regulatory limit. AMCA does not prefer this method due to the complexity and the lack of robust data on the impact of various guard designs on ACF efficiency.

Option 3: Separate Classes

A simpler path may be to provide two minimum-efficiency numbers/equations, one for guarded fans and one for unguarded fans. Figure 36 shows the significant difference between guarded and unguarded fans in the BESS Lab database.

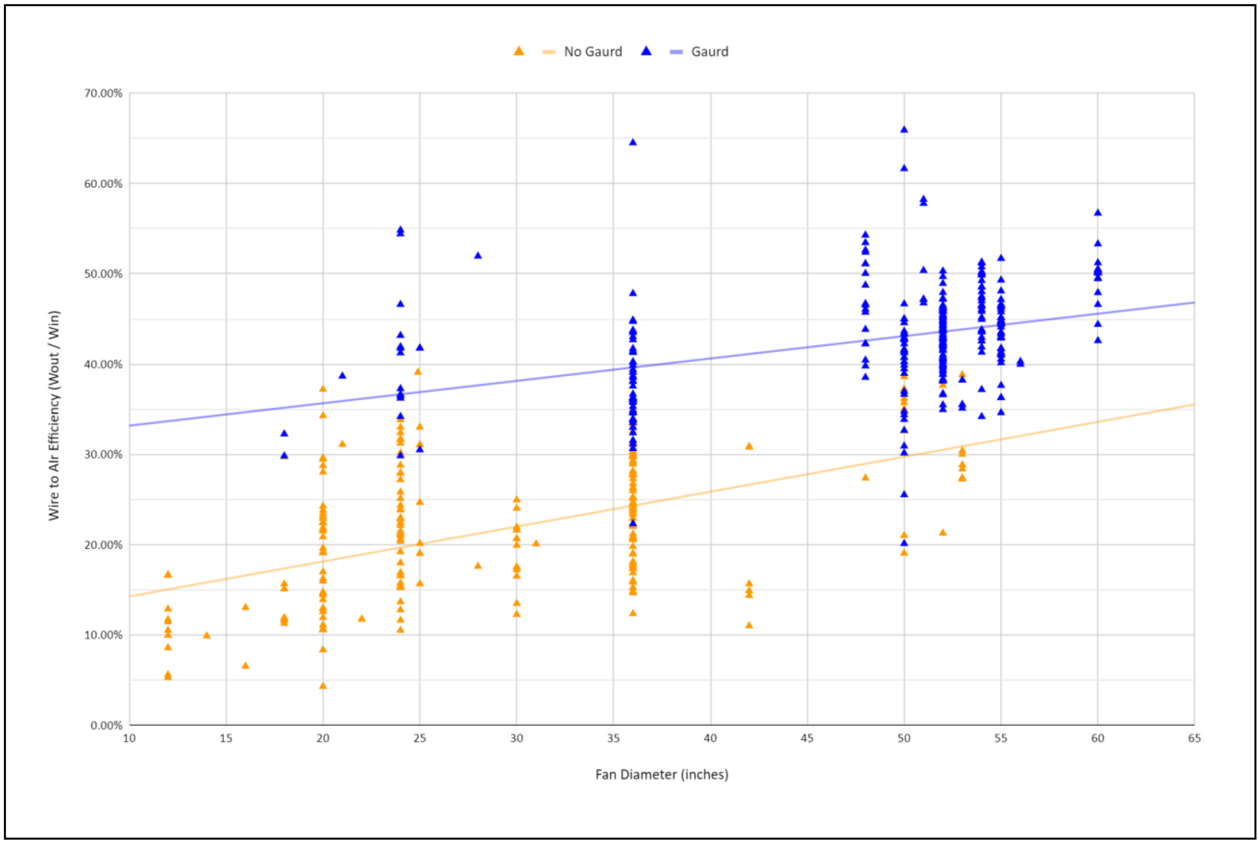


FIGURE 36. Wire-to-air efficiency vs. impeller diameter for the BESS Lab dataset.

With clearly distinct efficiency ranges, separate classes would allow for a simpler regulation. Unguarded fans would need to meet a higher minimum efficiency than guarded ones. With Equation 1 as the basis, equations 2 and 3 show how the two regulatory limits could be set:

$\eta_{guarded} = 0.004*D+0.06$ Equation 2

$\eta_{unguarded} = 0.003*D+0.25$ Equation 3

Figure 37 shows the pass/fail rate using equations 2 and 3 and can be contrasted with Figure 32, which is based on the DOE binned efficacies. Unlike the binned efficacies, equations 2 and 3 retain ACF with a wide range of 5D centerline velocities, while the fans that fail have the lowest wire-to-air efficiencies.

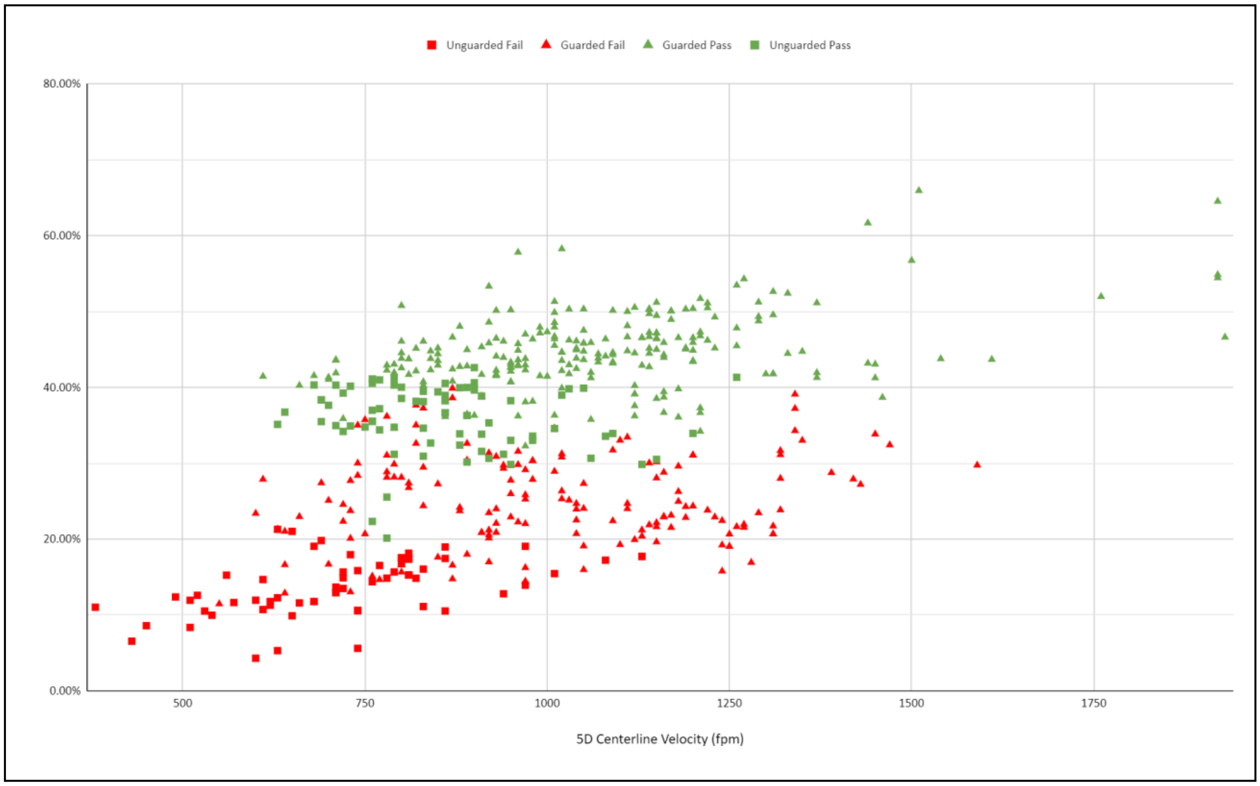


FIGURE 37. Pass/fail analysis of BESS Lab database based on equations 1 and 2.

Using equations 1 and 2, the overall failure rate for the BESS Lab database of fans is approximately 27 percent. As a reminder, this 27-percent failure rate is for a dataset DOE has stated "likely contains higher efficiency fans than the overall ACF market." AMCA believes that, for a first regulation, this is an aggressive but reasonable proposal because it is highly likely that the industry-wide failure rate would be much higher. DOE should consider that the impacts of this market will affect agricultural businesses, which already are under stress. In September 2023, the U.S. Senate Committee on Agriculture, Nutrition, and Forestry reported U.S. Department of Agriculture data indicating that weakening prices and increasing costs would lead in 2023 to a 23-percent ($42 billion) decline in profitability from 2022 for U.S. farms. "If realized, the $42B decline in U.S. net farm income will be the largest on record in nominal terms and the third largest of all time when adjusted for inflation," the agency said. With these stressors likely unmitigated and possibly exacerbated by inflation and these products being regulated for the first time, AMCA believes its recommended levels would achieve DOE goals for energy savings while also being fair and equitable for the industries that would be impacted, the agricultural community in particular.

SECTION 3: ISSUES ON WHICH DOE SEEKS COMMENT

In the NOPR, DOE enumerates 28 issues on which it seeks comment. AMCA responds below to as many issues as time in the review period allowed. Some responses refer to comments in sections 1 and 2.

Issue 1: *DOE requests comment on its proposed clarification for fans that create a vacuum. Specifically, DOE requests comment on whether fans that are manufactured and marketed exclusively to create a vacuum of 30 inches water gauge or greater could also be used in positive pressure applications. Additionally, DOE requests information on the applications in which a fan not manufactured or marketed exclusively for creating a vacuum would be used to create a vacuum of 30 inches water gauge or greater.*

AMCA Response:
A fan that is not manufactured or marketed exclusively for creating a vacuum would not be reliable for this application without substantial structural enhancement and geometric-characteristic change to withstand the aerodynamic surge that happens in this region of the performance curve.

The structural design and cost of the fan will change based on the pressure requirements and the material of construction.

What AMCA was seeking to exempt from the regulation was side-channel blowers and vacuum pumps (regenerative blowers) because it did not think these types of fans are testable to ANSI/AMCA Standard 210/ASHRAE Standard 51 (Figure 38).

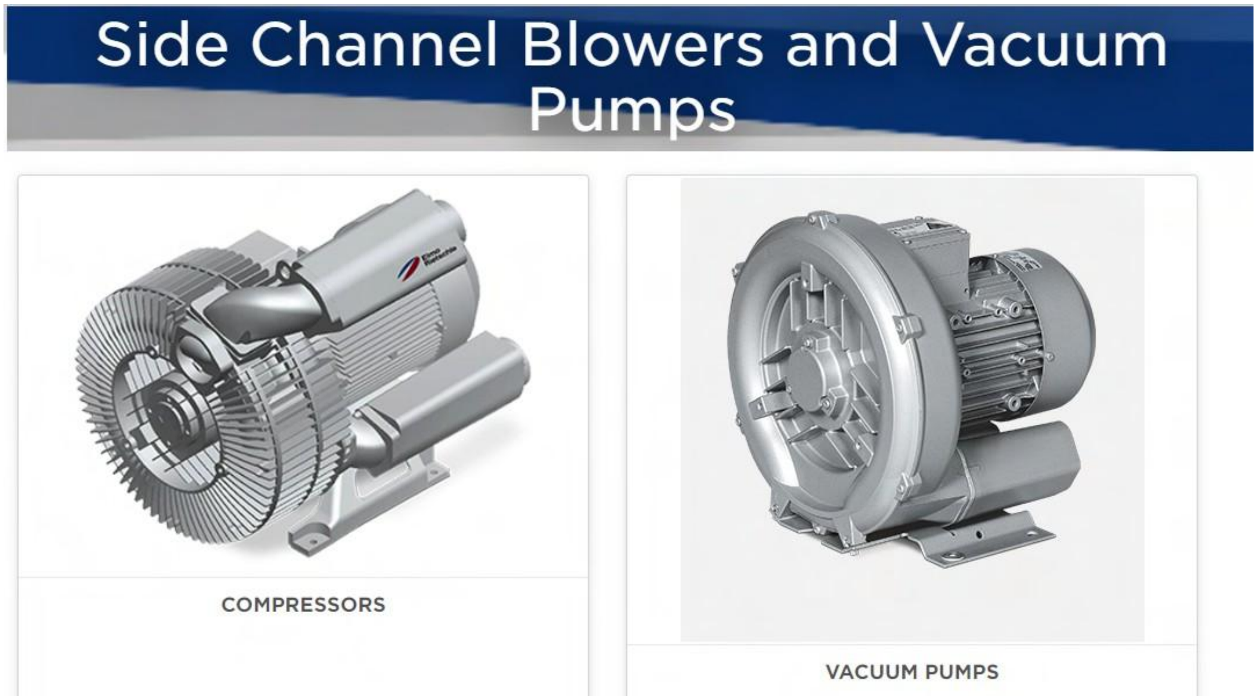


FIGURE 38. Side-channel blowers and vacuum pumps.

Issue 2: *DOE requests comments and feedback on the proposed methodology and calculation of motor and motor controller losses as well as potentially using an alternative calculation based on adjusted ANSI/AMCA Standard 214-21 equations.*

AMCA Response:
The comments and feedback DOE is seeking is contained in GFB 4 in Section1.

Issue 3: *DOE requests comment on whether there are specific fans that meet the axial ACF definition that provide utility substantially different from the utility provided from other axial ACFs and that would impact energy use. If so, DOE requests information on how the utility of these fans differs from other axial ACFs and requests data showing the differences in energy use due to differences in utility between these fans and other axial ACFs.*

AMCA Response:
Axial ACF are sold with or without a guard for the inlet and outlet of the impeller. The addition of accouterments such as guards, housings, and shutters affects performance.

For example, consider the performance of (1) a hanging air-circulating axial panel fan without guard tested per ANSI/AMCA Standard 230-23 and that of (2) an exhaust agricultural ventilation fan with housing (slant wall box), inlet shutter, and outlet cone with guard tested per ANSI/AMCA Standard 210/ASHRAE Standard 51 (Table 11, figures 39 and 40).

Acme 48" & 54" Air Circulating Axial Panel fan without guard									
BESS Number	D Impeller ø (in.)	Thrust (lbf)	rpm	Volts	Amps	kW	Efficiency Ratio (lbf/kW)	Airflow (Thrust cfm)	(thrust cfm/W)
c10004	48	20.38	355	229.5	4.35	0.792	25.7	20200	25.5
c10001	53.9	21.16	257	229.9	4.73	0.924	22.9	23000	24.9

Acme 48" & 54" Agricultural Ventilation Fan fan w/ housing, inlet shutter and outlet cone with guard									
BESS Number	D Impeller ø (in.)	SP (in.wc)	rpm	Volts	Amps	W	Airflow (cfm)	Efficacy (cfm/W)	
278	48	0.1	390	230	-	1047	19700	18.8	
300	54	0.1	354	230	-	1136	24200	21.3	

TABLE 11. Examples of ACF with guards.

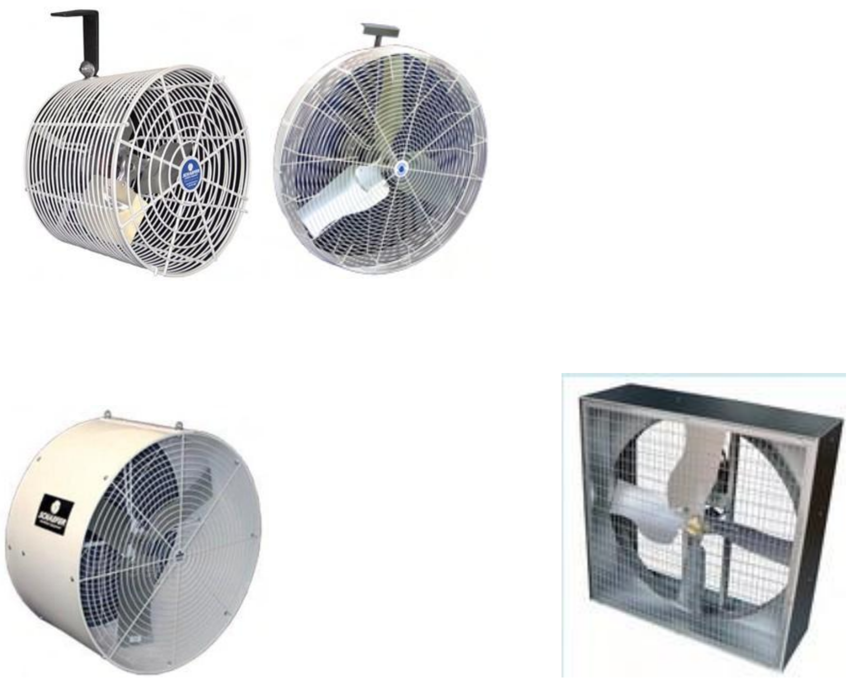


FIGURE 39. Unhoused ACF.

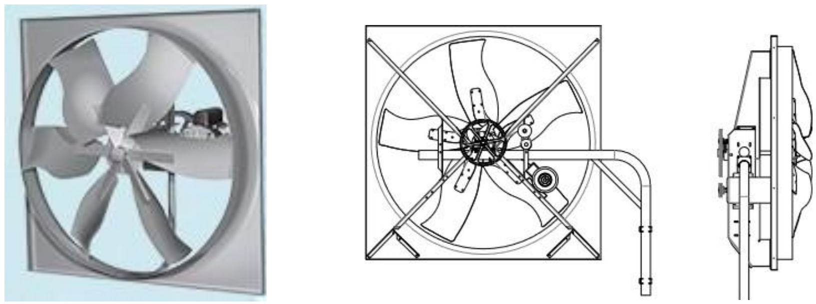


FIGURE 40. ACF without guards.

Summary

Both OSHA and UL 507, *Electric Fans*, require a fan to have a guard. OSHA requires that guards not have openings wider than one-half inch. OSHA requires guards for fans mounted less than 7 ft above the floor or working level.

Guards on basket circulating fans typically are made with concentric metal wire rings 0.5 in. apart on center.

Air-circulating axial panel fans hung a minimum of 7 ft from the floor typically do not have guards.

Issue 4: *DOE requests comment on its understanding that the diameter increase design option could be applied to non-embedded, non-space-constrained equipment classes.*

AMCA Response:
Utilizing a larger fan of similar type is a viable design option that can be applied to achieve higher FEI levels in non-embedded, non-space-constrained applications. The larger fan will run at a lower speed and have lower exhaust velocities. In many applications, this is an acceptable adjustment.

Non-embedded, non-space-constrained equipment classes are being interpreted as PRV for the purpose of this response. Increasing the diameter of PRV for a given application is not always a viable option. Many factors contribute to the choice of fan size in the design of a building ventilation system. There are national, state, and local building codes that factor into the overall system design, along with building-specific requirements (building footprint, operational considerations, architectural requirements, etc.). Increasing the diameter of a rooftop fan or fans can have a dramatic impact on overall HVAC-system design and building-structure requirements.

Retrofit fans face additional challenges. For example, an existing roof structure designed to support certain loads may need to be modified to support the additional weight of (a) larger-diameter fan(s). Changing duct and roof-curb size to match a larger fan typically is not practical in retrofit applications because of a variety of factors, including overall expense, permitting, and loss of operation time. This leads to the use of a larger fan with a curb adapter to sit on a smaller roof curb than the fan is designed for based on curb-base size. Using fans in this manner typically results in reduced performance of the fan because of the restricted inlet area of the roof curb vs. fan-curb base, which increases resistance pressure and decreases outlet airflow. The larger-diameter fan, then, may have to perform at higher static pressure, which may offset any performance benefit from switching to a larger fan.

Lastly, many axial PRV fans are equipped with weather protection that seals the outlet opening when not in operation. The protection mechanism often requires a minimum volume and velocity to open when turned on. Increasing fan diameter in low-volume circumstances will negate functionality and result in an inoperable condition. Specifically, for axial PRV fans, an increase in diameter may negate the functionality of the equipment and lead to a hazardous situation.

Issue 5: *DOE requests comment on whether the FEI increases associated with an impeller diameter increase for centrifugal PRVs and for axial PRVs are realistic. Specifically, DOE requests comment on whether it is realistic for axial PRVs to have a FEI increase that is 3 times greater than that for centrifugal PRVs when starting at the same initial diameter. Additionally, DOE requests comment on the factors that may impact how much an impeller diameter increase impacts a FEI increase.*

AMCA response:
The NOPR (Page 3765, Column B) states:

> *When analyzing its data sources, DOE found that this 18 percent diameter increase when maintaining the operating point could result in a range of FEI increases, from as low as 4-percent to as high as 30- percent, corresponding to a FEI increase of approximately 0.03 to 0.30. For this NOPR analysis, DOE assumed that a diameter increase for centrifugal PRV exhaust and supply fans would result in a 0.03 increase in FEI and a diameter increase for axial PRV fans would result in a 0.09 – 0.10 increase in FEI. DOE recognizes that initial diameter size, operating airflow, and operating pressure may impact how effective an impeller diameter increase is for increasing FEI. Specifically, the duty points that DOE chose to evaluate may be duty points where a diameter increase is very effective at increasing fan efficiency or may be duty points where a diameter increase has minimal impact on fan efficiency. DOE could adjust the efficiency gains from an impeller diameter increase in its analysis so that there is a larger FEI gain for all PRVs, and where PRVs could reach higher FEI values for a lower cost. Alternately, DOE could decrease the FEI gain for axial PRVs from an impeller diameter increase, allowing axial PRVs to reach higher FEI values for a higher cost since the impeller diameter increase would no longer provide such a large increase in FEI.*

DOE then provided representative duty points it used to analyze centrifugal PRV and axial PRV (TSD, Page 83, Table 5.5.1) (Table 12).

Table 5.5.1 Representative Diameters and Duty Points for GFBs

Equipment Class	Diameter (in.)	Pressure (in. wg)	Airflow (CFM)
Axial Inline	24	1.25	7,000
		3.50	11,000
	30	1.00	10,000
		3.00	10,000
	36	2.00	30,000
		2.25	20,000
Panel	24	0.50	6,000
	36	0.25	10,000
	48	0.25	25,000
Axial PRV	36	0.25	10,000
	48	0.50	30,000
	60	0.50	50,000
Centrifugal PRV – Exhaust	15	1.00	3,000
	24	0.25	6,000
	30	0.50	10,000
Centrifugal PRV - Supply	10	0.50	3,000
	12	0.25	3,000
	15	0.75	7,000
Centrifugal Housed	12	2.00	2,000
		5.25	2,000
	15	1.50	3,000
		5.25	3,000
	22	1.25	7,000
		4.25	13,000
	33	1.75	15,000
		6.25	22,000
	40	1.75	20,000
		7.50	30,000
Centrifugal Unhoused	18	3.00	4,000
	24	3.50	11,000
	27	4.00	15,000
Centrifugal Inline	18	1.00	5,000
	30	1.25	12,000
	36	2.00	14,000
Radial Housed	14	9.75	1,000
	22	30.0	500
	33	17.0	8,000

TABLE 12. Table 5.5.1 from TSD.

Inspection of one manufacturer's selection software for centrifugal-PRV and axial-PRV fans shows a wide variety of percent-diameter increases from one size to another. Diameter increases range from 9 percent to 25 percent, depending on model and place in the size range. DOE’s use of an 18-percent diameter increase often necessitates an increase by two fan sizes. Specifically, comparing FEI from one given size and utilizing an 18-percent increase in diameter rarely yields a fan size from the same manufacturer that could be used to generate a comparison FEI value.

Using selection software from two manufacturers, an attempt was made to duplicate DOE's findings (tables 13 and 14). Selections were made for two centrifugal-PRV- (CPRV-) exhaust models at duty points and sizes provided in Table 5.5.1 of the TSD (or Fig. 12 above) to determine FEI and the delta in FEI with an 18-percent increase in diameter.

Fan Type	DOE Model Size	Available Model Size	Airflow (CFM)	Static Pressure (in. w.g)	FEI	18% increase	Closest Available Model	Increased Diameter FEI	FEI Differential
CPRV 1	15	16	3000	1.00	1.16	17.70	18	1.42	0.26
CPRV 1	24	24	6000	0.25	1.04	28.32	30	below 1 hp	NA
CPRV 1	30	30	10000	0.50	0.98	34.40	36	1.37	0.39
CPRV 2	15	16	3000	1.00	1.2	17.7	18	1.34	0.14
CPRV 2	24	24	6000	0.25	1.15	28.32	30	below 1 hp	NA
CPRV 2	30	30	10000	0.50	1.07	34.40	36	size not avail.	NA

TABLE 13. Manufacturer 1 CPRV-exhaust diameter-increase selections.

Fan Type	DOE Model Size	Available Model Size	Airflow (CFM)	Static Pressure (in.w.g.)	FEI	18% Diameter Increase	Closest Available Model	Increased Diameter FEI	FEI Differential
CRPV-1	15	14	3000	1.00	1.03	17.70	16	1.24	0.21
CRPV-1	24	24	6000	0.25	1.04	28.32	30	1.62	0.58
CRPV-1	30	30	10000	0.50	0.97	35.40	36	1.47	0.50
CRPV-2	15	14	3000	1.00	1.16	17.70	16	1.29	0.13
CRPV-2	24	24	6000	0.25	1.21	28.32	30	1.74	0.53
CRPV-2	30	30	10000	0.50	1.07	35.40	36	1.57	0.50

TABLE 14. Manufacturer 2 CPRV-exhaust diameter-increase selections.

The same exercise was attempted for axial-PRV selections. The user needs to be cautious to ensure that a geometrically similar impeller is selected when comparing sizes. Fan models with adjustable pitches, hub sizes, or blade count can yield very different results when performance, FEI ratings, or efficiency are being compared. Additionally, many axial-PRV fans are equipped with weather hoods or butterfly dampers that seal the fan-outlet opening when the fan is not in operation. The protection mechanism often requires a minimum volume and velocity to open when turned on. Upsizing in low-volume circumstances will negate its functionality and result in an inoperable condition. Because of these issues and limited larger sizes, data availability is limited.

Both manufacturers found that the DOE representative operating points used for axial-PRV fans generally were not possible for the baseline fan sizes. Recommended operating points whereby all operating points are viable selections for the baseline fan size of at least one manufacturer are shown in tables 15-18. The name of the manufacturer(s) have been anonymized. The axial-PRV examples are at the recommended operating points, not the baseline points used by DOE in the NOPR.

Fan Type	DOE Model Size	Available Model Size	Airflow (CFM)	Static Pressure (in.w.g.)	Operable
ARPV-1	36	36	10000	0.25	Yes
ARPV-1	48	48	30000	0.50	No
ARPV-1	60	60	50000	0.50	No
ARPV-2	36	36	10000	0.25	Yes
ARPV-2	48	48	30000	0.50	No
ARPV-2	60	60	50000	0.50	No

Note: For this and tables 16-18, “model size” denotes nominal model size; it does not correlate 100 percent to impeller diameter.

TABLE 15. Manufacturer 2 APRV-exhaust operating-point viability.

Fan Type	DOE Model Size	Available Model Size	Airflow (CFM)	Static Pressure (in.w.g.)	Operable
ARPV-1	36	36	10000	0.25	Yes
ARPV-1	48	48	20000	0.50	Yes
ARPV-1	60	60	30000	0.50	Yes
ARPV-2	36	36	10000	0.25	Yes
ARPV-2	48	48	20000	0.50	Yes
ARPV-2	60	60	30000	0.50	Yes

Note: Yellow-highlighted cells indicate changed airflow values to create possible selections.

TABLE 16. Manufacturer APRV-exhaust operating-point recommendations.

Fan Type	DOE Model Size	Available Model Size	Airflow (CFM)	Static Pressure (in. w.g)	FEI	18% increase	Closest Available Model	Increased Diameter FEI	FEI Differential
APRV 1	36	36	10000	0.25	1.54	42.48	42	NA	NA
APRV 1	48	48	20000	0.50	1.26	56.64	54	1.34	0.08
APRV 1	60	60	30000	0.50	1.22	70.82	NA	NA	NA
APRV 2	36	36	10000	0.25	1.71	42.48	42	NA	NA
APRV 2	48	48	20000	0.50	1.16	56.64	54	1.26	0.1
APRV 2	60	60	30000	0.50	1.26	70.82	NA	NA	NA

TABLE 17. Manufacturer 1 APRV-exhaust diameter-increase selections.

Fan Type	DOE Model Size	Available Model Size	Airflow (CFM)	Static Pressure (in.w.g.)	FEI	18% Diameter Increase	Closest Available Model	Increased Diameter FEI	FEI Differential
ARPV-1	36	36	10000	0.25	1.27	42.48	42	1.38	0.11
ARPV-1	48	48	20000	0.50	1.02	56.64	54	1.4	0.38
ARPV-1	60	60	30000	0.50	0.95	70.80	72	1.19	0.24
ARPV-2	36	36	10000	0.25	1.62	42.48	42	1.63	0.01
ARPV-2	48	48	20000	0.50	1.28	56.64	54	1.54	0.26
ARPV-2	60	60	30000	0.50	1.23	70.80	72	0.98	-0.25

TABLE 18. Manufacturer 2 APRV-exhaust diameter-increase selections.

AMCA is unable to confirm DOE's conclusion that a diameter increase of 18 percent provides a three-times greater increase in FEI for an axial PRV than for a centrifugal PRV. AMCA examples show that FEI changes associated with increases in diameter can fluctuate depending on the duty point selected and the model being compared. The variability in FEI change based on diameter appears to be greater with axial-PRV fans than with centrifugal-PRV fans likely because of the various additional variables—pitch, hub size, blade count—present in axial fans. Specifically for axial-PRV fans, an increase in diameter may negate the functionality of the equipment as intended and lead to a hazardous situation. Impeller configuration is likely to change for axial fans to better "tune" performance for the desired duty point. Comparing FEI values for different impeller configurations may be misleading.

Issue 6: *DOE requests comment on the ordering and implementation of design options for centrifugal PRV exhaust and supply fans and axial PRV fans*.

AMCA Response:
In the NOPR (Page 3765, Column C), DOE applies the impeller changes and aerodynamic redesigns for PRV to the baseline fan so that PRV can reach higher efficiency levels while maintaining the baseline impeller diameter. While manufacturers would have the option of

achieving higher efficiencies by increasing fan diameter, DOE assumed that, if manufacturers were to change the impeller or redesign a PRV, they would apply these design changes to their entire diameter range, enabling the baseline-diameter fan to reach the higher efficiency levels.

The design path for all PRV is shown in Table 19. For the PRV equipment classes, the impeller change(s) and diameter increase(s) are ordered by FEI increase, with the design option with the smallest FEI increase ordered first. DOE could consider an analysis with a different ordering of design options based on manufacturer sales price (MSP) increase or cost-effectiveness. Alternatively, DOE could consider an analysis that does not include increased fan diameter as a design option. In this alternative analysis, DOE could consider an additional impeller change as a design option to increase FEI. However, based on its analysis, DOE expects that removing increased fan diameter as a design option in its analysis would increase the cost of achieving higher efficiency.

Table IV-11 Summary of Efficiency Levels for All GFB Equipment Classes

		EL0	EL1	EL2	EL3	EL4	EL5†	EL6†	EL7†
Axial Inline	Design Option	Baseline: tube axial	Impeller change	Switch to vane axial	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	-	-
	FEI	0.84	0.87	1.00	1.18	1.36	1.55	-	-
Panel	Design Option	Baseline	Impeller change	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	4th Aero redesign	-	-
	FEI	0.80	0.86	1.00	1.24	1.48	1.73	-	-
Axial PRV	Design Option	Baseline	Impeller change 1	Impeller change 2	Diameter Increase*	Diameter Increase*	1st Aero redesign	2nd Aero redesign	3rd Aero redesign
	FEI	0.66	0.69	0.72	0.75	0.85	1.00	1.25	1.49
Centrifugal PRV Exhaust	Design Option	Baseline	Diameter Increase	Impeller change*	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	4th Aero redesign	-
	FEI	0.67	0.7	0.72	0.86	1.00	1.20	1.39	-
Centrifugal PRV Supply	Design Option	Baseline	Diameter Increase	Impeller change*	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	4th Aero redesign	-
	FEI	0.69	0.72	0.76	0.88	1.00	1.19	1.37	-
Centrifugal Housed Main Path	Design Option	Baseline	Impeller change	Airfoil Impeller	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	-	-
	FEI	0.63	0.93	1.00	1.15	1.31	1.46	-	-
Centrifugal Housed FC Path**	Design Option	Baseline	Impeller change	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	-	-	-
	FEI	0.63	0.93	1.00	1.15	1.31	-	-	-
Centrifugal Unhoused	Design Option	Baseline	Impeller change 1	Impeller change 2	1st Aero redesign	2nd Aero redesign	3rd Aero redesign	-	-
	FEI	0.94	1.00	1.10	1.23	1.35	1.49	-	-

TABLE 19. Table IV-11 of the NOPR assumes manufacturers would generally apply a propeller change across an entire model line.

Manufacturers generally offer several models for each equipment class. Models differ by features, drive type, performance, technology, cost, etc. Based on the duty point selected and the customer's requirements, one model may be more desired than another because of these model differences. Depending on the duty point needed and resulting FEI value of each model and size, a customer may choose to upsize the desired fan model or convert to another model to meet necessary FEI requirements and design needs.

Generally, all models can meet FEI values by having their performance range limited (being slowed down) or their diameter increased. Once their range becomes limited or their diameter increased to the point they no longer are competitive in the market, the models likely are obsoleted or redesigned using more efficient technologies. The decision to increase diameter or implement more energy-efficient technology is not cut-and-dried and depends greatly on equipment's current capabilities, market requirements, and the alternative models available.

Referring to the response for Issue 4: "Utilizing a larger fan of similar type is a viable design option that can be applied to achieve higher FEI levels in non-embedded, non-space-constrained applications. The larger fan will run at a lower speed and have lower exhaust velocities. In many applications, this is an acceptable adjustment."

Issue 7: *DOE requests comment on its approach for estimating the industry-wide conversion costs that may be necessary to redesign fans with forward-curved impellers to meet higher FEI values. Specifically, DOE is interested in the costs associated with any capital equipment, research and development, or additional labor that would be required to design more efficient fans with forward-curved impellers. DOE additionally requests comment and data on the percentage of forward-curved impellers that manufacturers would expect to maintain as a forward-curved impeller relative to those expected to transition to a backward- inclined or airfoil impeller.*

AMCA response:
AMCA has no comment.

Issue 8: *DOE requests comment on the equations developed to calculate the credit for determining the FEI standard for GFBs sold with a motor controller and with an FEPact less than 20 kW and on potentially using an alternative credit calculation based on the proposed equations in section III.C.1.b of this document. Additionally, DOE requests comment on its use of a constant value, and its proposed value, of the credit applied for determining the FEI standard for GFBs with a motor controller and an FEPact of greater than or equal for 20 kW.*

AMCA response:
As stated in GFB 6, AMCA appreciates the efforts by DOE to create an aligned calculation methodology for motor efficiency and accommodating motor-speed-controller losses. The additional complexity of both an equation-based approach and doubling the number of fan categories, however, diminishes some of the virtues of the FEI metric, including simplicity for practitioners and comparability of selectable fans across product classes.

To simplify the process while still encouraging the use of fans with speed controls, AMCA recommends following the approach taken with ANSI/ASHRAE/IES 90.1, which provides a 5-percent credit for variable-air-volume applications (speed-modulated). This is achieved most easily by applying the credit directly to the FEI value of fans sold with a controller, regardless of the methodology used to calculate FEI (i.e., wire-to-air, AEDM, or the method provided by DOE in Section 2.6 of the energy-conservation standard).

Issue 9: *DOE requests comments on whether it should apply a correction factor to the analyzed efficiency levels to account for the tolerance allowed in AMCA 211-22 and if so, DOE requests comment on the appropriate correction factor. DOE requests comment on the potential revised levels as presented in Table IV-12. Additionally, DOE requests comments on whether it should continue to evaluate an FEI of 1.00 for all fan classes if it updates the databases used in its analysis to consider the tolerance allowed in AMCA 211-22.*

AMCA response:
Manufacturer-published data, including values in the AMCA 2021 Fan Shipment Database, are accurate and do not include a tolerance on the power value. Therefore, AMCA does not recommend adjusting FEI-limit values based on the power tolerance included for check tests in AMCA Publication 211-22 (Rev. 01-23). The tolerance is necessary for the performance of a population of products because of both measurement and manufacturing variability under surveillance testing.

Instead, AMCA recommends DOE apply a power tolerance for surveillance testing in alignment with the tolerances provided in AMCA Publication 211-22 (Rev. 01-23). AMCA further recommends the proposed FEI levels not include a tolerance value, as those are representative of nominal performance. Similar to the tolerances for airflow and pressure under Section 2.a, AMCA recommends the measured power at either the operating point or corrected to the certified rating point be acceptable if, for shaft-power measurements, the power (P) is within 7.5 percent of the certified power or 0.05 hp, whichever value is greater. For wire-to-air rated power, the measured or certified point corrected power should be within 10 percent of the certified power or 50 W, whichever value is greater. These tolerances allow for a combination of measurement variability between laboratories and manufacturing variability in a population of fan products.

Issue 10: *Additionally, DOE does not anticipate that the efficiency levels captured in Table IV-12 would impact the cost, energy, and economic analyses presented in this document. As such, DOE considers the results of these analyses presented throughout this document applicable to the efficiency levels with a 5% tolerance allowance. DOE seeks comment on the analyses as applied to the efficiency levels in Table IV-12*.

AMCA response:
AMCA addresses this topic in its response to Issue 9 above.

Issue 11: *DOE requests comment on its method to use both the AMCA sales database and sales data pulled from manufacturer fan selection data to estimate MSP. DOE also requests comment on the use of the MSP approach for its cost analysis for GFBs or whether an MPC-based approach would be appropriate. If interested parties believe an MPC-based* *approach would be more appropriate, DOE requests MPC data for the equipment classes and efficiency levels analyzed, which may be confidentially submitted to DOE using the confidential business information label.*

AMCA response:
AMCA has no comment.

Issue 12: *DOE requests feedback on whether using a more efficient motor would require an ACF redesign. Additionally, DOE requests feedback on what percentage of motor speed change would require an ACF redesign.*

AMCA response:
Use of a more efficient motor likely would require an ACF redesign, unless a seamless, drop-in motor replacement were readily available on the market. Specifically, a drop-in motor replacement would need to match current-state motor mounting, shaft dimensions, frame size, IP rating, motor weight, and enclosure dimensions (especially for overhung motor designs) at a minimum to avoid the need for an ACF redesign. Furthermore, any changes to the fan's operating speed or power transmission (i.e., a belt-driven fan being transitioned to a direct-driven fan) as a result of using a more efficient motor would inherently require an ACF redesign because of potential impacts on the structural and mechanical design of the fan. Based on AMCA's experience, the likelihood of identifying and sourcing a drop-in replacement that meets all of these requirements and is cost-neutral is extremely low. As a result, it is very likely that all ACF would require a redesign to incorporate a more efficient motor and comply with DOE's minimum efficiency levels.

While AMCA cannot comment on a specific percentage of motor-speed change that would require an ACF redesign, it is likely that even a small change would necessitate an ACF redesign. This would be dependent on the design of the fan and the engineering safety factors incorporated in said design. For example, if an ACF was designed with minimal safety factor for speed variance, any change in speed could result in unsafe operation, unwanted vibration and noise, reduced life expectancy, and unexpected performance or utility losses. Additionally, the change in fan speed that would be incurred when transitioning from a belt-driven fan to a direct-driven fan would invariably necessitate a fan redesign because of the magnitude of the speed change required to maintain fan performance and utility.

Issue 13: *DOE requests feedback on whether setting an ACF standard using discrete efficacy values over a defined diameter range appropriately represents the differences in efficacy between axial ACFs with different diameters, and if not, would a linear equation for efficacy as a function of diameter be appropriate.*

AMCA response:
AMCA does not recommend the use of discrete values in binned diameter ranges or the use of a linear equation for ACF efficiency. This topic is covered in ACF 3.

Issue 14: *DOE seeks comment on the distribution channels identified for GFBs and ACFs and fraction of sales that go through each of these channels.*

AMCA response:
AMCA has no comment.

Issue 15: *DOE seeks comment on the overall methodology and inputs used to estimate GFBs and ACFs energy use. Specifically, for GFBs, DOE seeks feedback on the methodology and assumptions used to determine the operating point(s) both for constant and variable load fans. For ACFs, DOE requests feedback on the average daily operating hours, annual days of operation by sector and application, and input power assumptions. In addition, DOE requests feedback on the market share of GFBs and ACFs by sector (i.e., commercial, industrial, and agricultural).*

AMCA response:
AMCA has no comment but encourages DOE to gather additional information from individual manufacturers and large retailers of ACF through confidential interviews.

Issue 16: *DOE requests feedback on the price trends developed for GFBs and ACFs.*

AMCA response:
AMCA has no comment.

Issue 17: *DOE requests feedback on the installation costs developed for GFBs and on whether installation costs of ACFs may increase at higher ELs.*

AMCA response:
AMCA has no comment.

Issue 18: *DOE requests feedback on whether the maintenance and repair costs of GFBs may increase at higher ELs. Specifically, DOE requests comments on the frequency of motor replacements for ACFs. DOE also requests comments on whether the maintenance and repair costs of ACFs may increase at higher ELs and on the repair costs developed for ACFs.*

AMCA response:
AMCA has no comment.

Issue 19: *DOE requests comments on the average lifetime estimates used for GFBs and ACFs.*

AMCA response:
AMCA generally agrees with average-lifetime estimates for GFB and ACF. AMCA encourages DOE to gather additional information on product lifetimes from individual manufacturers through confidential interviews.

Issue 20: *DOE requests feedback and information on the no-new-standards case efficiency distributions used to characterize the market of GFBs and ACFs. DOE requests information to support any efficiency trends over time for GFBs and ACFs.*

AMCA response:
AMCA has no comment but encourages DOE to gather additional information from individual manufacturers through confidential interviews.

Issue 21: *DOE requests feedback on the methodology and inputs used to project shipments of GFBs in the no-new-standards case. DOE requests comments and feedback on the potential impact of standards on GFB shipments and information to help quantify these impacts.*

AMCA response:
AMCA has no comment.

Issue 22: *DOE requests feedback on the methodology and inputs used to estimate and project shipments of ACFs in the no-new-standards case. DOE requests comments and feedback on the potential impact of standards on ACF shipments and information to help quantify these impacts.*

AMCA response:
AMCA has no comment but encourages DOE to gather additional information from individual manufacturers through confidential interviews.

Issue 23: *DOE requests comment and data regarding the potential increase in utilization of GFBs and ACFs due to any increase in efficiency.*

AMCA response:
AMCA has no comment regarding GFB.

Regarding ACF, AMCA believes the likelihood of consumers utilizing ACF more often because of an increase in efficiency vs. current offerings is low. Because of the utility and simplistic nature of ACF, consumers are unlikely to see value in a more efficient and more costly version. That said, as it is difficult to understand the increase in cost for more efficient ACF designs, manufacturers run the risk of pricing ACF in a manner justifying the consideration of alternative options (e.g., portable air-conditioning units). Portable air-conditioning units are less efficient when compared with elevated air speeds produced by ACF with regard to occupant comfort. This would be a detriment to the desired energy-conservation impact of this rulemaking. The CBE Thermal Comfort Tool by California Berkley demonstrates how elevated air speeds impact occupant comfort. Table 22 lists an example of how energy use could increase if this shift were to happen.

Fan Type	Power (W)	Impacted Area (ft²)	W/ft²
ACF	460	3,000	0.15
Portable AC	930	250	3.7

TABLE 22. Comparison of energy usage between different cooling solutions.

Issue 24: *DOE requests comment on the number of end-use product (i.e., a product or equipment that has a fan or blower embedded in it) basic models that would not be excluded by the list of products or equipment listed in Table III-1.*

AMCA response:
Products potentially not listed in Table III-1, most of which have industrial applications, are shown in Table 23. Photographs of exemplar embedded fan equipment are provided (Figure 43). AMCA encourages DOE to contact manufacturers for a more complete list of products with fans or blowers embedded in them.

End Use Product	Equipment Class Used
Ovens	Housed and Unhoused Centrifugal, Axial Inline, Centrifugal Inline, Axial PRV, Centrifugal RPV
Dust Collectors	Housed and Unhoused Centrifugal,Radial Housed
Grain Dryers	Housed and Unhoused Centrifugal, Axial Inline
Process Dryers	Housed and Unhoused Centrifugal,Radial Housed
Industrial Flash Freezers	Housed and Unhoused Centrifugal, Axial Inline
Paint Finishing Systems	Housed and Unhoused Centrifugal
Boilers	Housed Centrifugals, Radial Housed
Gas Tubines	Housed and Unhoused Centrifugal
Large Fan,Blower Coils	Housed and Unhoused Centrifugal
Custom Air Handlers	Housed and Unhoused Centrifugal, Axial Inline, Centrifugal Inline, Axial Panel
Custom Packaged Equipment	Housed and Unhoused Centrifugal, Axial Panel
Custom Dedicated Outdoor Air Systems	Housed and Unhoused Centrifugal, Axial Panel
Clean Rooms	Housed and Unhoused Centrifugal, Axial Inline, Centrifugal Inline
Industrial Burers (Combustion Air)	Housed Centrifugal, Radial Housed
Thermal Oxidizers	Housed Centrifugal
Food Processing	Housed and Unhoused Centrifugal, Axial Inline, Centrifugal Inline, Axial PRV, Centrifugal RPV
Paint Booths	Housed Centrifugal, Axial Inline, Centrifugal Inline
Odor Control Systems	Housed Centrifugal, Axial Inline, Centrifugal Inline
Fume Scrubbers	Housed Centrifugal, Axial Inline, Centrifugal Inline
Filtration Systems	Housed Centrifugal,
Swamp / Evaporative Coolers	Unhoused Circulating Fan
Protable Evaporative Coolers	Unhoused Circulating Fan
Misting / Fogging Fans	Housed and Unhoused Air Circulating Fans
Transformers	Unhoused Circulating Fan
Wind Machines / Frost Protestion Fans	Unhoused Circulating Fan
Air Supported Structures (Veolodromes, Ice Rinks, Bouncy Castles)	Housed Centrifugal, Radial Housed

TABLE 23. Partial list of end-use products having embedded fans.

General air-filtration unit

Boiler combustion air

Grain drying

Paint finishing system

Dust collector

FIGURE 43. Photographs of exemplar embedded fan equipment. These types of applications are primarily industrial and usually not in mind when discussing embedded fans.

Issue 25: *DOE requests information regarding the impact of cumulative regulatory burden on manufacturers of fans and blowers associated with multiple DOE standards or product-specific regulatory actions of other Federal agencies.*

AMCA response:

AMCA is particularly concerned about the cumulative regulatory burden imposed by this rulemaking and respectfully requests that DOE appreciate that, in this case, it is very important to go beyond the traditional scope of its loosely defined obligation to avoid crafting a regulation that subjects those regulated to “multiple related federal standards for the same product or manufacturer.”

DOE is encouraged to consider not only the burden of accumulating relevant federal regulations, but also the varying approaches the State of California is taking to regulating, marketing, and labeling the same products. This is particularly important because DOE's inaction on the ASRAC consensus recommendations for GFB during the last year of the Obama administration and the following Trump administration directly led to California feeling compelled to move forward with its own regulation of most of the same products. The effort, through preemption (which, in the case of the test procedure, California recognized very late in the process), to harmonize the federal and state laws has fallen short, and manufacturers are faced with the combined burden of having to live with conflicting state and federal approaches from the outset of the effective date of the California standard.

This is a first-of-its-kind energy-efficiency standard and test procedure that requires fan manufacturers to not only improve or eliminate specific products but re-educate market actors throughout the buying chain about highly technical and nuanced characteristics of the FEI metric and its application. Fan manufacturers and their customers now must engage in a complex exchange of information and analysis about how a fan will be used to determine if any particular fan will be compliant with the new energy-efficiency standard, considering what may end up being fans rated under variable conditions, such as whether they are tested or sold with an electronic drive.

While the industry has championed this approach as the best way to deliver real and meaningful energy savings, this should not minimize the challenge and burden placed on manufacturers and customers to deliver these energy savings. The new regulations do not just require manufacturers to no longer sell certain products for certain applications; they require manufacturers and others to restructure how they market and sell every product.

The cumulative-regulatory-burden analysis should be conducted with careful consideration of the type of manufacturer experiencing the burden. In this case, regardless of whether a manufacturer's number of employees meets the traditional categorization of "small" vs."medium" size (AMCA members are well-represented in both categories), the number of highly trained technical laboratory technicians and professional fan engineers in these companies is quite small. They have been subjected to continuous distraction from their core responsibilities to the business to focus on preparing for massive, simultaneous, and often unclear regulatory changes driven by DOE and the State of California. The traditional analysis of the cumulative regulatory burden for large corporations that manufacture a variety of different products that are all subject to overlapping regulatory timetables (for example, an appliance company making regulated washing machines, dishwashers, and refrigerators) get some relief for being forced to have to deal with simultaneous regulations. However, for the much smaller AMCA members, nearly every category of product they sell in the "within scope" GFB and ACF categories are being burdened by this new regulation.

AMCA asks that DOE reduce the burden in this case by being more flexible with respect to energy-efficiency levels, effective dates, and exercise of its discretion with respect to the timing of enforcement.

Regarding ACF, the fan industry and its associated industries have been simultaneously working on the following DOE standards/regulatory actions:

- Electric motors—test procedure and energy-conservation standard.
- Ceiling fans—test procedure and energy-conservation standard.
- Ceiling-fan light kits—test procedure and energy-conservation standard.
- Fans and blowers—test procedure and energy-conservation standard.
- ACF (under fans and blowers)—test procedure and energy-conservation standard.

While ceiling fans and ceiling-fan light kits are covered by revisions to existing regulations, the other rulemakings cover new product classes. New regulations, especially ones where re-testing and re-rating of products changes published performance data, create a significant burden on manufacturers. Technical staff often are working to complete testing to the new standard while simultaneously responding to comments on the energy-standard NOPR for those products.

Components of fans also being newly regulated creates additional issues. For further consideration, DOE Energy Conservation Standards for Electric Motors 88 FR 72347 went into effect June 1, 2023. The scope of electric motors covered was expanded to include totally enclosed air-over (TEAO) motors and expanded-scope electric motors, among others. Motor manufacturers are having to redesign some motors to meet the new efficiency standards. Historically, motors nameplate-rated at 230V have had the statement “Usable at 208V” on the nameplate, even if the motor did not meet the DOE efficiency standard at 208V. This no longer is allowed, forcing motor manufacturers to develop a separate motor with the 208V rating. This requires fan manufacturers to stock two motor SKUs (230V and 208V) when only one was required in the past and to certify ACF performance twice when, in the past, it had to do so only once.

Issue 26: *DOE requests comment on the proposed standard level for axial PRVs, including the design options and costs, as well as the burdens and benefits associated with this level and the industry standards/California regulations FEI level of 1.00.*

AMCA response:
In the NOPR (Page 3845, Column B), DOE states:

> *DOE is proposing an FEI level of 0.85 (EL4) for axial PRVs. In section IV.C.1.b, DOE developed the MSP-efficiency relationship based on data from the AMCA sales database as well as performance data from manufacturer fan selection software and performance data provided from confidential manufacturer interviews. From its analysis, DOE estimated that EL4 for axial PRVs would be achieved by implementing two impeller diameter increases. Based on the MSP-efficiency results, EL4 for axial PRVs is the highest level with positive life-cycle costs savings. Furthermore, as discussed in section IV.C.1.b, ASHRAE 90.1–2022 set an FEI target of 1.00 for all fans within the scope of*

that standard, which includes axial PRVs. CEC requires manufacturers to report fan operating boundaries that result in operation at a FEI of greater than or equal to 1.00 for all fans within the scope of that rulemaking, which includes axial PRVs. DOE also notes that, based on its shipments analysis, 50-percent of axial PRVs have an FEI of at least 1.00. Additionally, based on its review of the market, DOE has found that most manufacturers offer models of APRVs that have an FEI of at least 1.00 at a range of diameters. Based on this, DOE expects that the market is already shifting towards an FEI of 1.00 for axial PRVs and that this level may not be unduly burdensome for manufacturers to achieve.

DOE is proposing an FEI level of 0.85 (EL4) for axial PRV.

DOE estimated that EL4 (FEI = 0.85) for axial PRV would be achieved by implementing two impeller-diameter increases. Note that this design option is relative to baseline and not EL3, as shown in Table 24.

Axial PRV	Design Option	Baseline	Impeller change 1	Impeller change 2	Diameter Increase*	Diameter Increase*	1st Aero redesign	2nd Aero redesign	3rd Aero redesign
	FEI	0.66	0.69	0.72	0.75	0.85	1.00	1.25	1.49

TABLE 24. Axial-PRV design options.

Based on the LCC analysis of 10,000 sample points, 72.9 percent of the duty points are above 0.85 (green line), while only 49.5 percent of the duty points are above 1.00 (orange line); thus, half of the sample selections are eliminated (Figure 44).

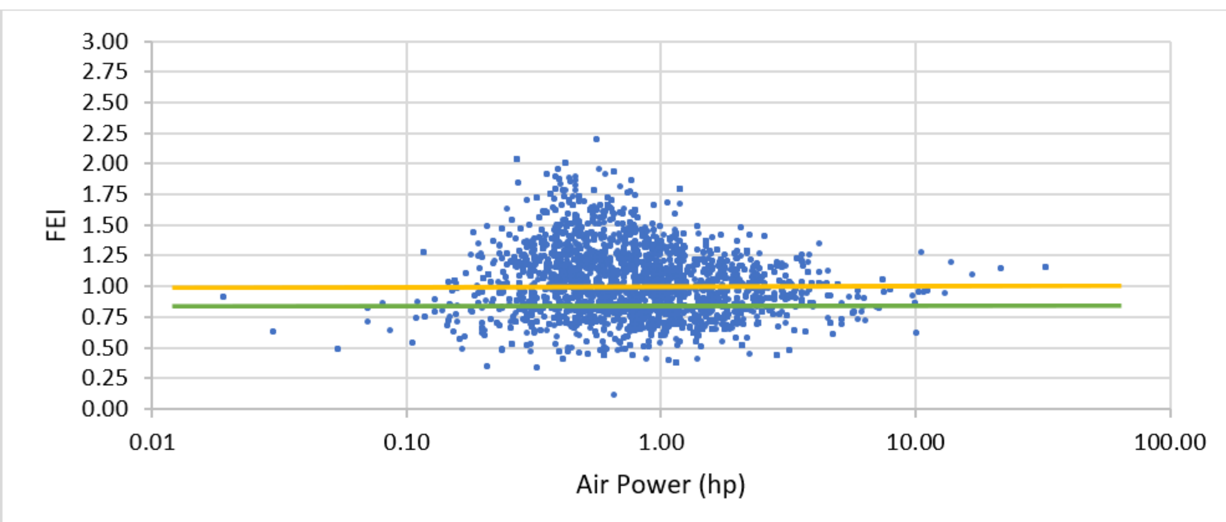


FIGURE 44. Axial-PRV air power vs. FEI, with FEI thresholds of 0.85 (green line) and 1.00 (orange line) shown.

	Diameter (in.)	Air Pressure (in w.g.)	Airflow (cfm)
Axial PRV	36	0.25	10,000
	48	0.50	30,000
	60	0.50	50,000

TABLE 25. Axial-PRV representative duty points from TSD Table 5.5.1.

Using manufacturer selection software and the representative duty points given in Table 25 at the diameters provided, selections generally were available for the two lowest air powers of 10,000 cfm at 0.25-in. w.g. and 36-in. diameter and 30,000 cfm at 0.5-in. w.g. and 48-in. diameter without additional blade-topology changes or size increases. The representative duty point of 50,000 cfm at 0.5-in. w.g. and 60-in. diameter, however, found far fewer options meeting the proposed minimum FEI of 0.85 or the 1.00 reporting requirements in California. Diameter increases are limited for this duty point, as 60 in. tends to be the largest diameter provided by many manufacturers. A blade topology change, entire model change, or system redesign is needed in many circumstances to meet this representative duty point. It still may be untenable to achieve the highest air-power values requested in all cases without diameter increases, which add cost to the unit and installation as well as building modifications.

From a cost standpoint, the improvement of changing blade topology from a baseline fan selection to one meeting FEI requirements at the high-air-power list price is roughly a 1.60x increase, based on one manufacturer's selection software. Table 5A.2.3 of the TSD (Table 26 below—note that diameters listed in the table are incorrectly labeled) shows the difference between baseline MSP ($5,524) and required EL4 MSP ($9,747) is a 1.76x increase. This has general alignment with what is calculated using the referenced software selection.

Table 5A.2.3 Price-Efficiency Data for Axial PRV Fans

Efficiency Level	Design Option	FEI	10-in. MSP	12-in. MSP	15-in. MSP
0	Baseline	0.66	$3,198	$4,180	$5,524
1	Impeller Change 1	0.69	$4,733	$6,144	$8,040
2	Impeller Change 2	0.72	$4,795	$6,222	$8,140
3	Diameter Increase 1	0.75	$3,687	$5,106	$7,135
4	Diameter Increase 2	0.85	$4,389	$6,491	$9,747
5	1st Aero Redesign	1.00	$4,795	$6,222	$8,140
6	2nd Aero Redesign	1.25	$4,795	$6,222	$8,140
7	3rd Aero Redesign	1.49	$4,795	$6,222	$8,140

TABLE 26. Table 5A.2.3 from the TSD.

The burden for manufacturers and purchasers will be increased, as will material and labor costs for larger-diameter fans at high air-power requirements. Axial-PRV fans are used to move high volumes of air at low pressures and can be quite large at airflows of 50,000 cfm and higher. As shown in Figure 44, few selections can meet an FEI of 0.85 above 7 hp, showing the industry has limited capabilities at the high-end range, and an FEI limit of 0.85 is exceeding the FEI_{max} previously calculated for this performance range.

Issue 27: *DOE requests comment on the number of small-business OEMs identified that manufacture fans and blowers covered by this proposed rulemaking.*

AMCA response:
For GFB, AMCA has no comment.

For ACF, in TSD Table 3.5.2, DOE identified 48 ACF OEMs. To that list, AMCA would like to add the following small businesses:

- Airmaster
- American Coolair Corp.
- Vostermans Ventilation Inc. (Multifan)
- Rapid Engineering
- QFFI
- Absolut Aire
- Cambridge Engineering
- Mestek
- Engineering Air
- Modine

Small businesses that assemble impellers and cones into OEM products (fans with a minimum testable configuration) will become fan manufacturers by definition.

The number of soon-to-be designated fan manufacturers surely will be in the hundreds and very likely thousands.

Issue 28: *DOE requests comment on the estimated small business costs and how those may differ from the costs incurred by larger manufacturers.*

AMCA response:
Small businesses are less likely to be able to invest in an on-site testing laboratory and the associated staff, including engineering resources. This generally means testing needs to be conducted at an external test facility (BESS Lab or AMCA). Contract testing with independent laboratories can be significantly more expensive and comes with longer lead times, especially when multiple rulemakings are simultaneously consuming testing resources.

Issue 29: *Additionally, DOE welcomes comments on other issues relevant to the conduct of this rulemaking that may not specifically be identified in this document.*

AMCA response:

AMCA has provided commentary in Sections 1 and 2 to issues beyond those that DOE raised as those it seeks comment.

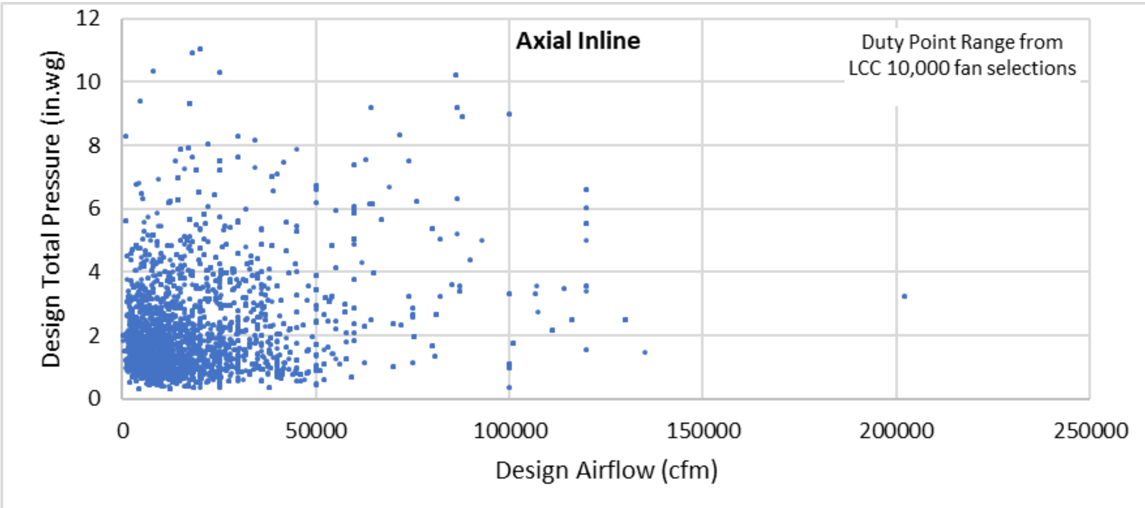
APPENDIX A – Supplemental Information for Proposed FEI Levels for Equipment Classes

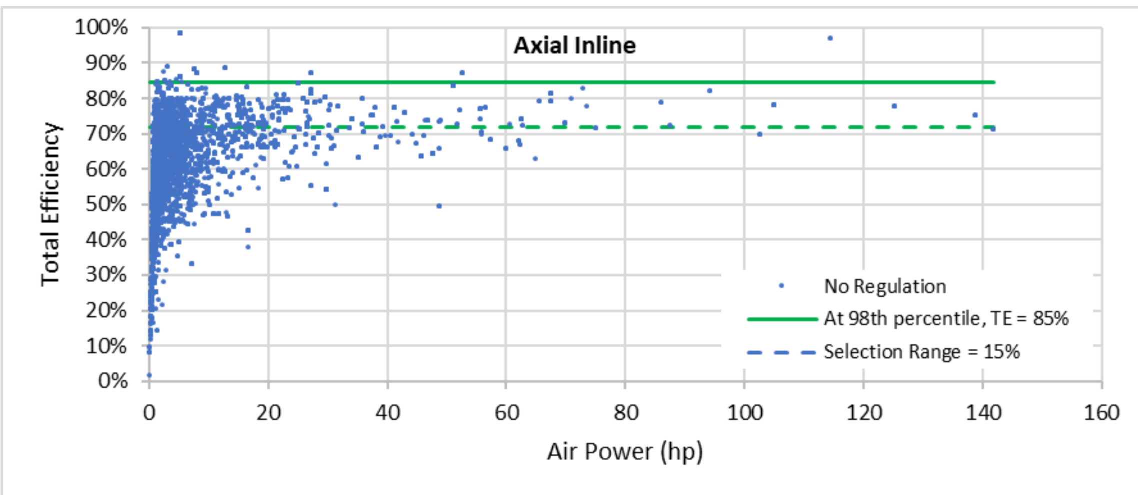
Analysis for each equipment class follows the proposed FEI-determination methods explained in detail under previous comments (Issue #1). Each equipment class is analyzed using the LCC duty points provided by DOE. Subsequent analysis can be performed, if additional or modified data are presented. The methodology would remain the same; however, it is possible the proposed FEI level could change, if there are significant differences between the datasets.

Each equipment class shown below has the same format of charts, with three charts per page:

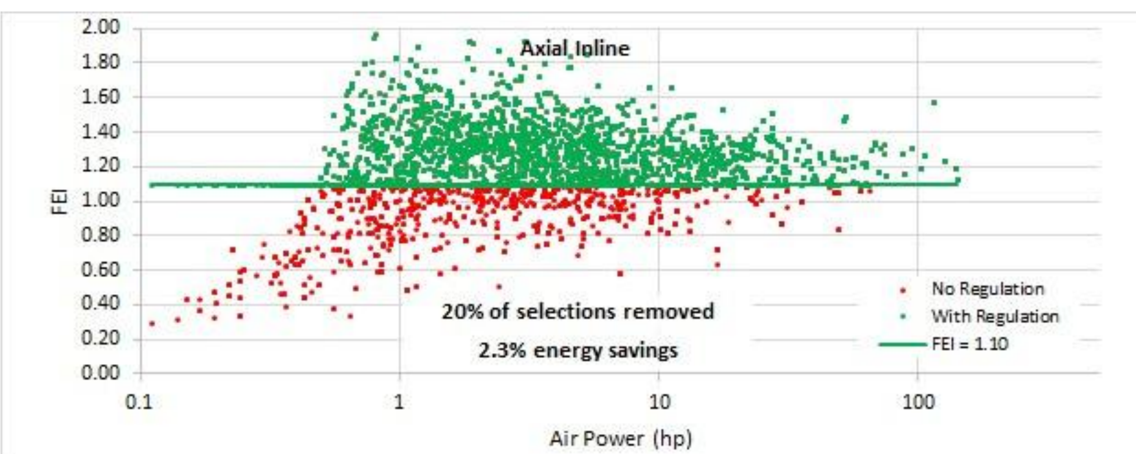
- **Top:** Plots the roughly 10,000 sample points used in the LCC analysis for pressure (in. w.g.) and volume (cfm).
- **Middle:** Converts the points used in the top chart to efficiency vs. air power. The chart indicates the 98th percentile (max-tech) for efficiency and a 15-percentage-point selection range.
- **Bottom:** Converts the same points used in the previous two charts to FEI vs. air power. The chart shows the AMCA-proposed FEI level and indicates which selections would be above and below that level, along with a percentage of selections removed and percent energy saved based on LCC analysis.

Product Class: Axial Inline

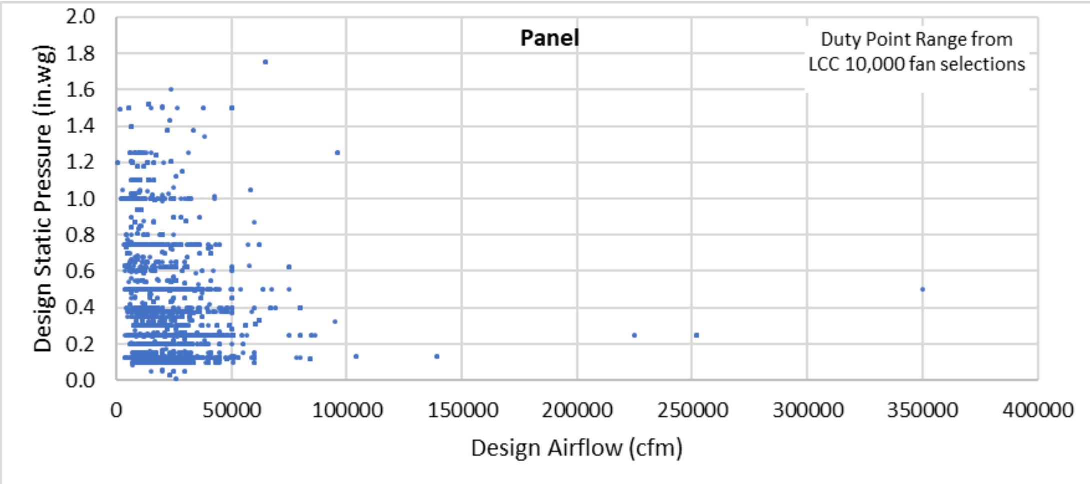


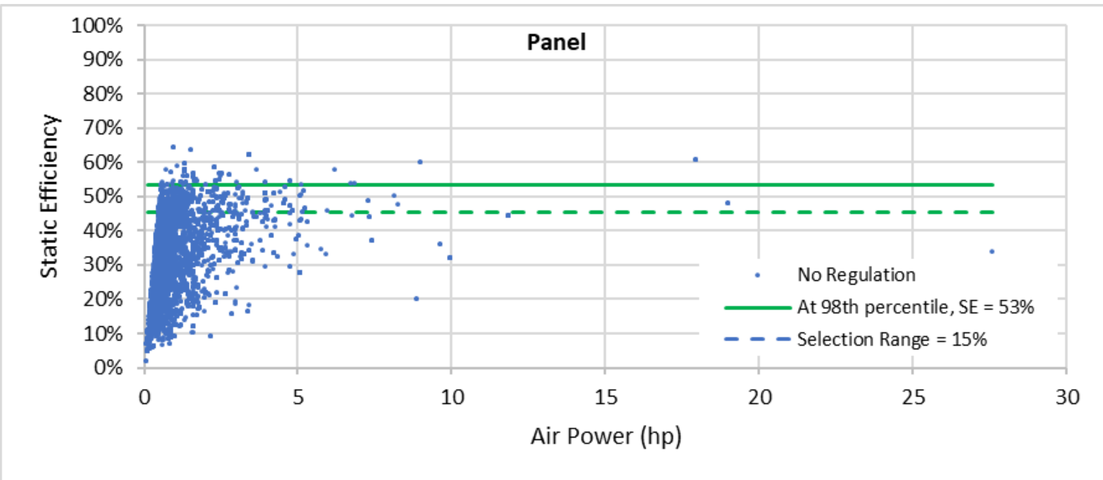


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

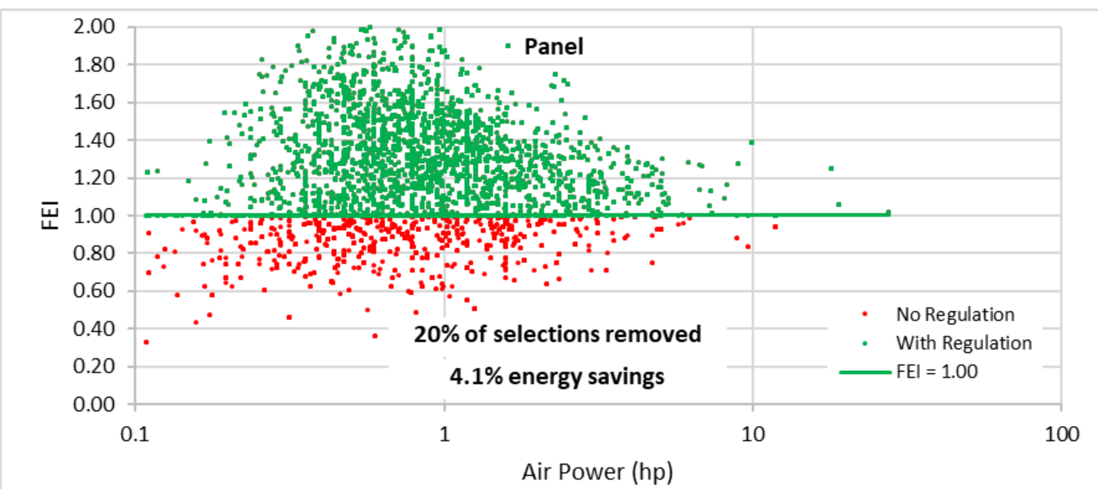


Product Class: Axial Panel

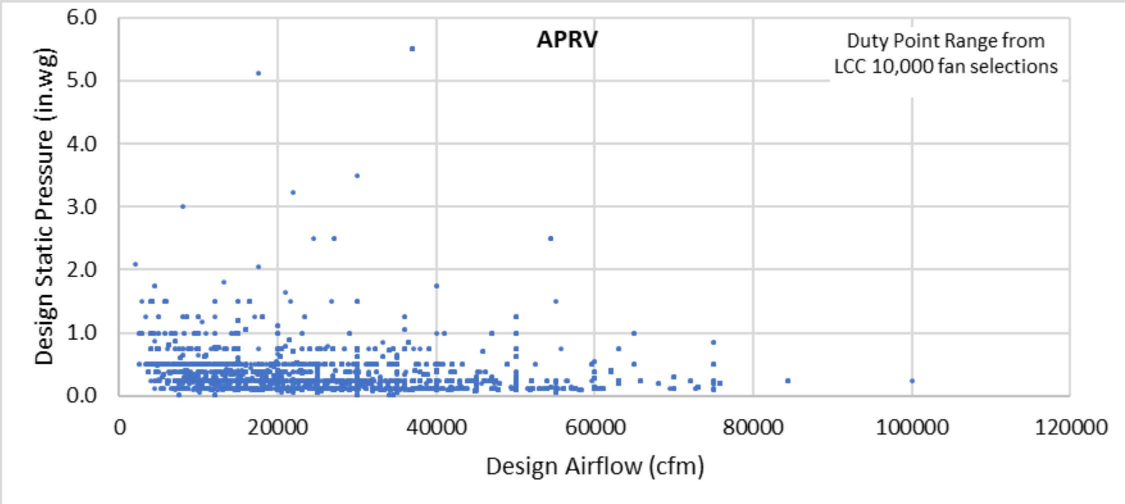


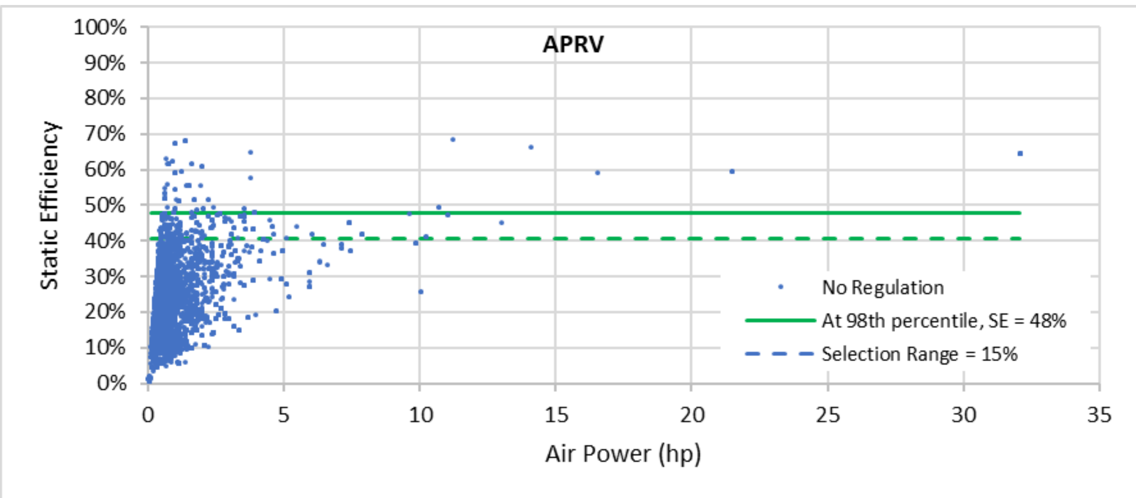


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

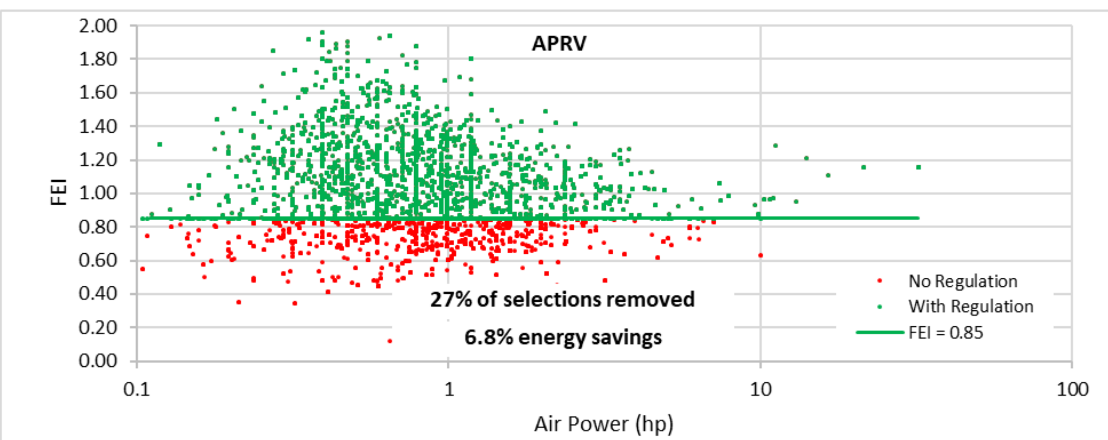


Product Class: Axial Power Roof Ventilator (APRV)

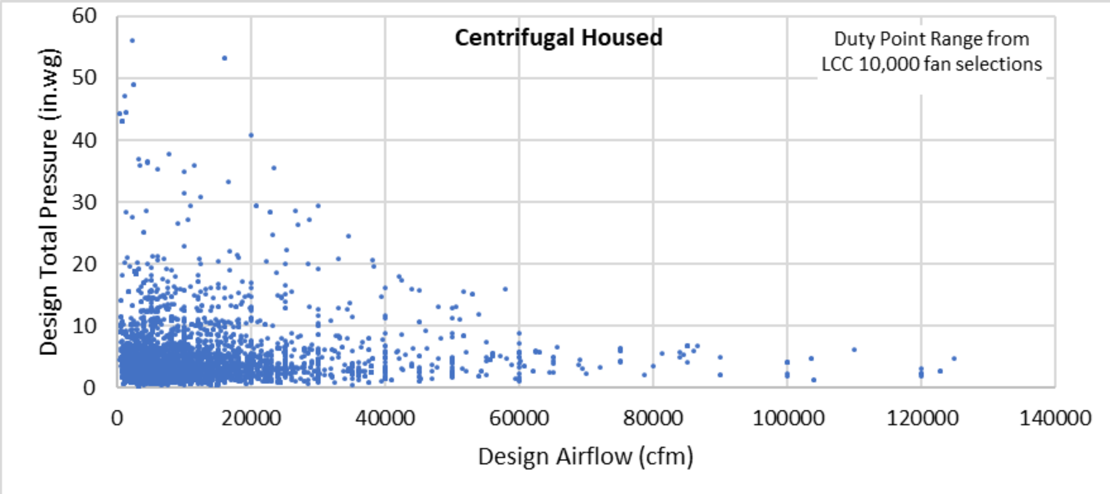


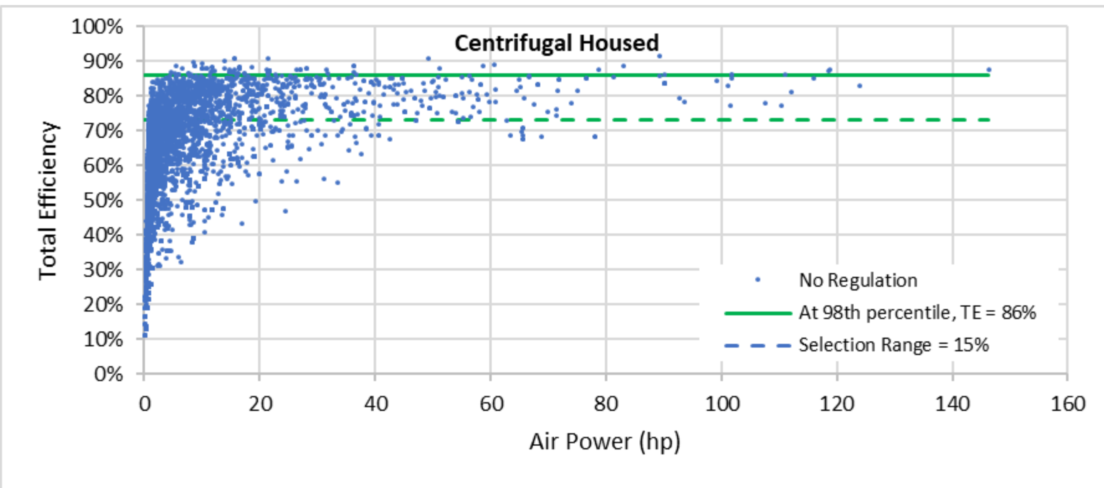


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

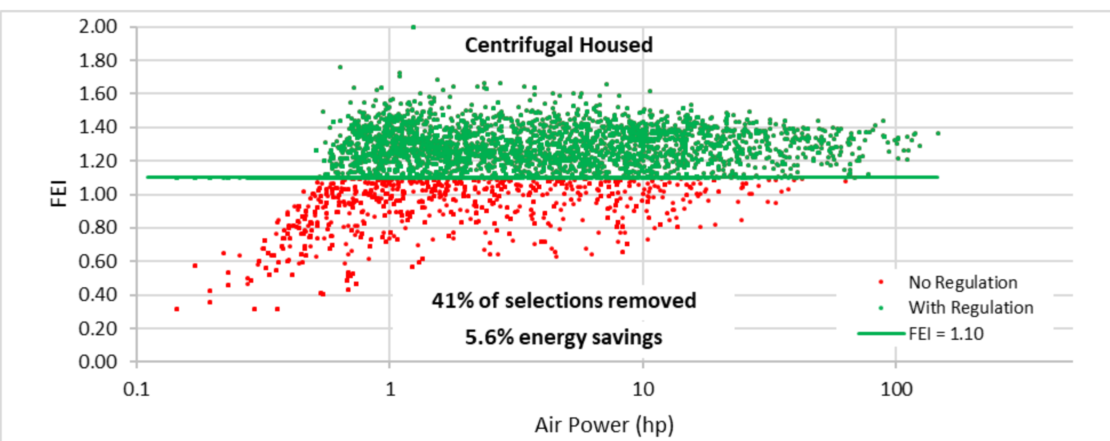


Product Class: Centrifugal Housed

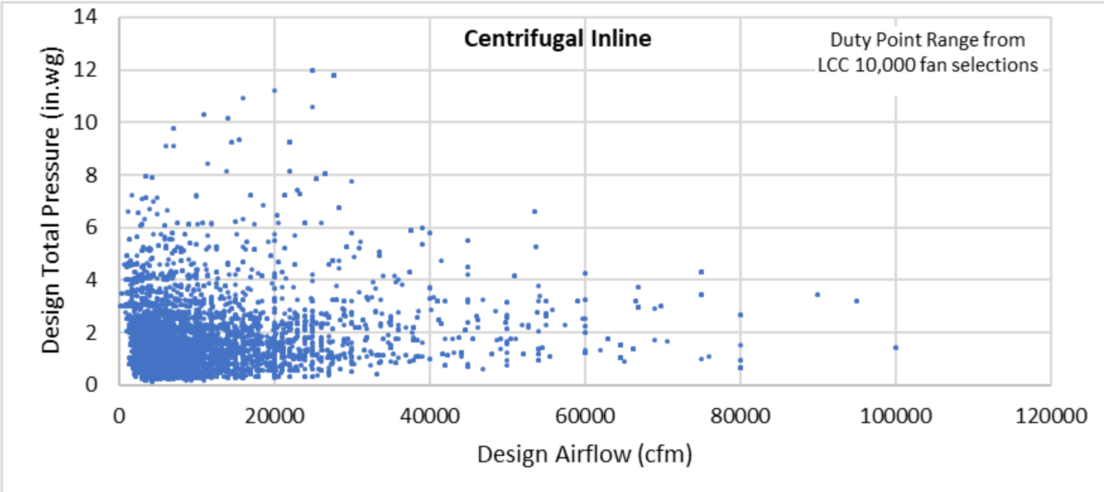


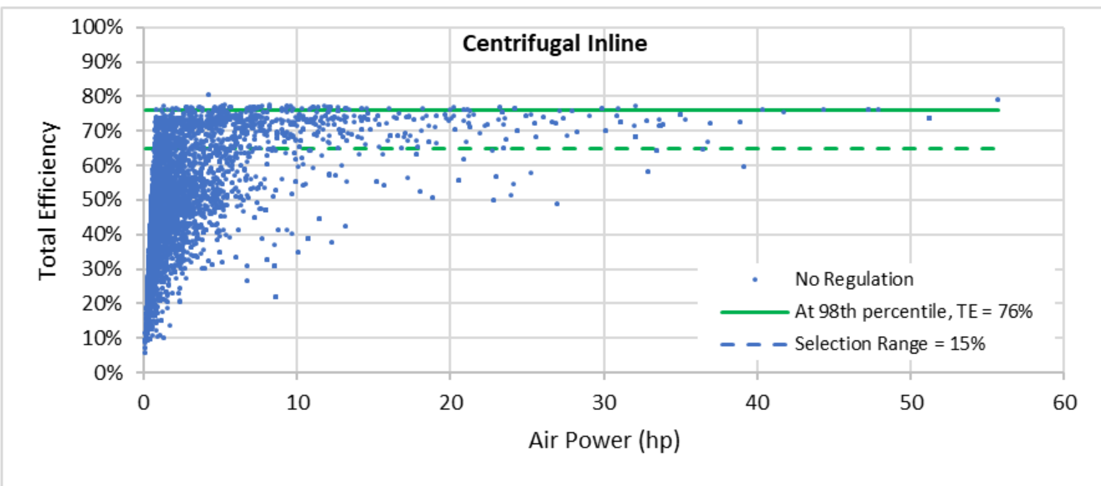


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

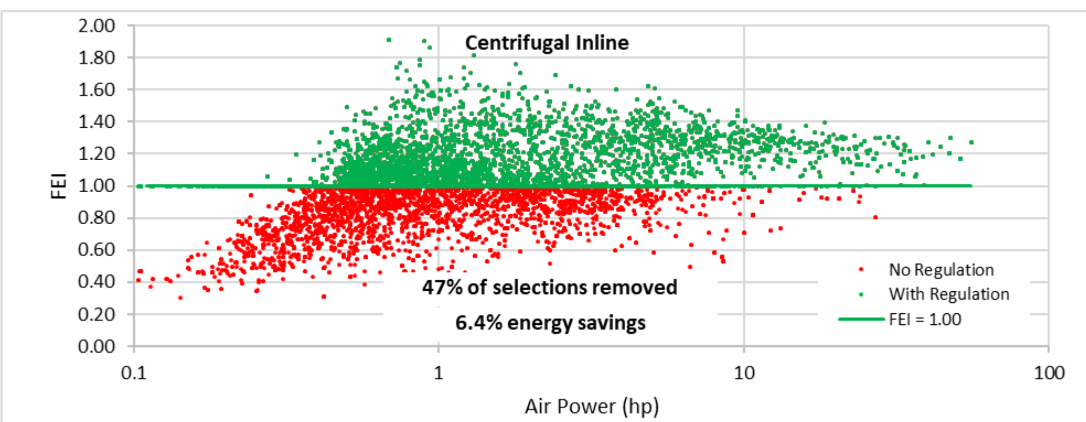


Product Class: Centrifugal Inline

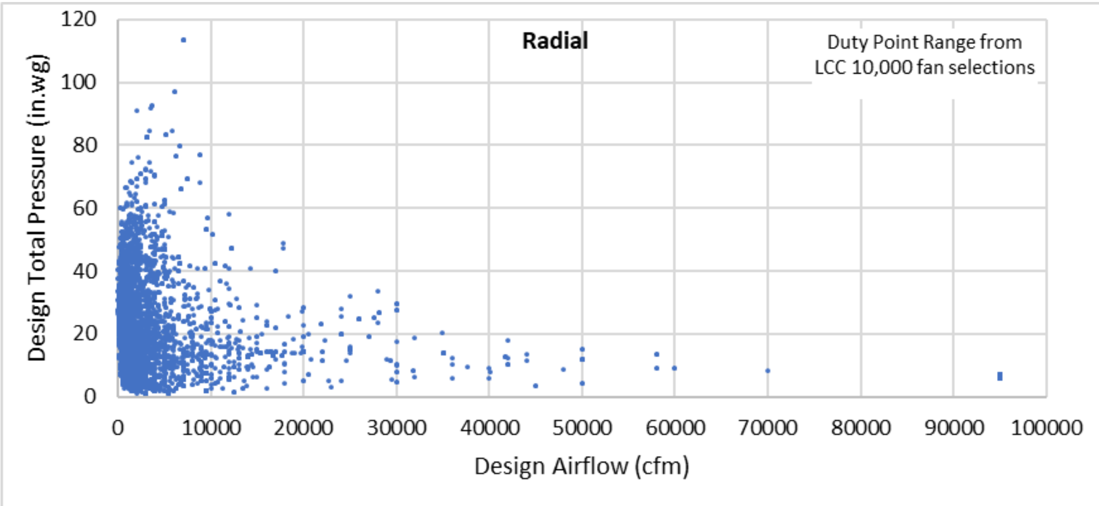


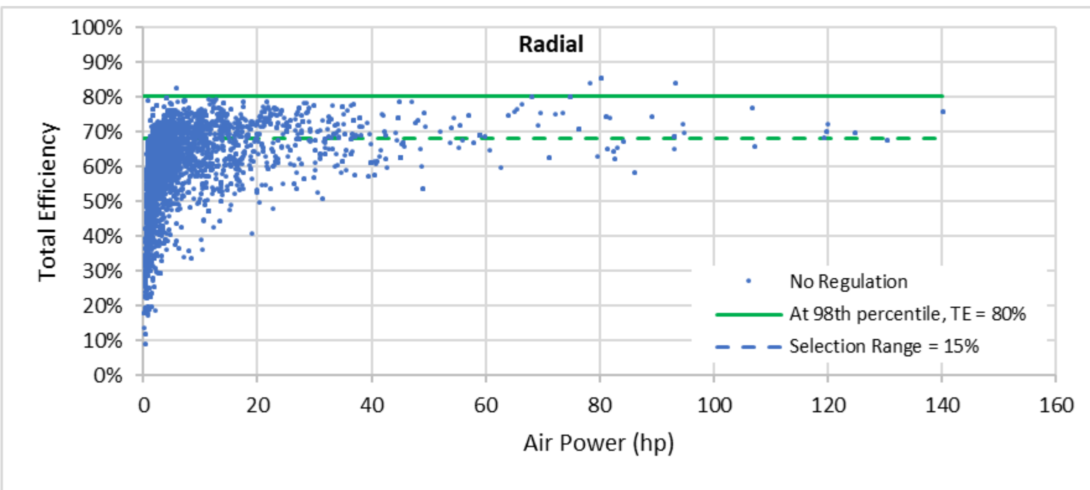


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

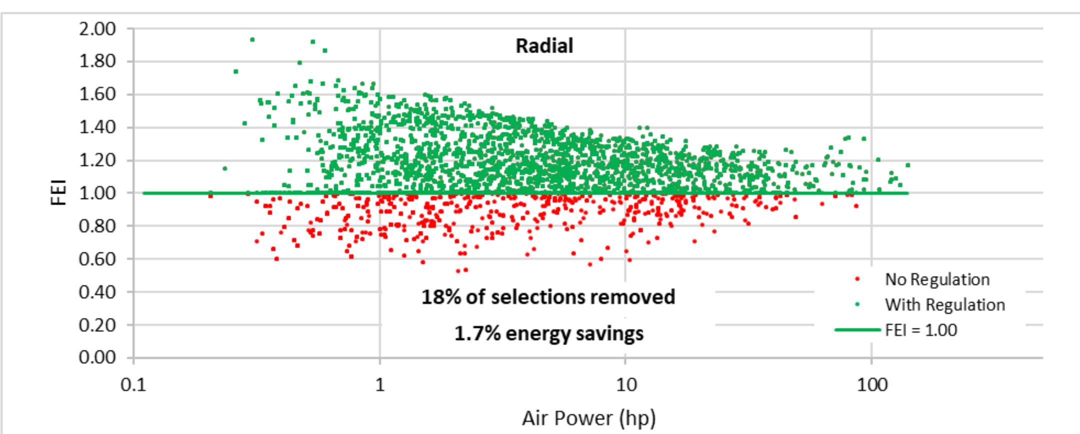


Product Class: Radial Housed

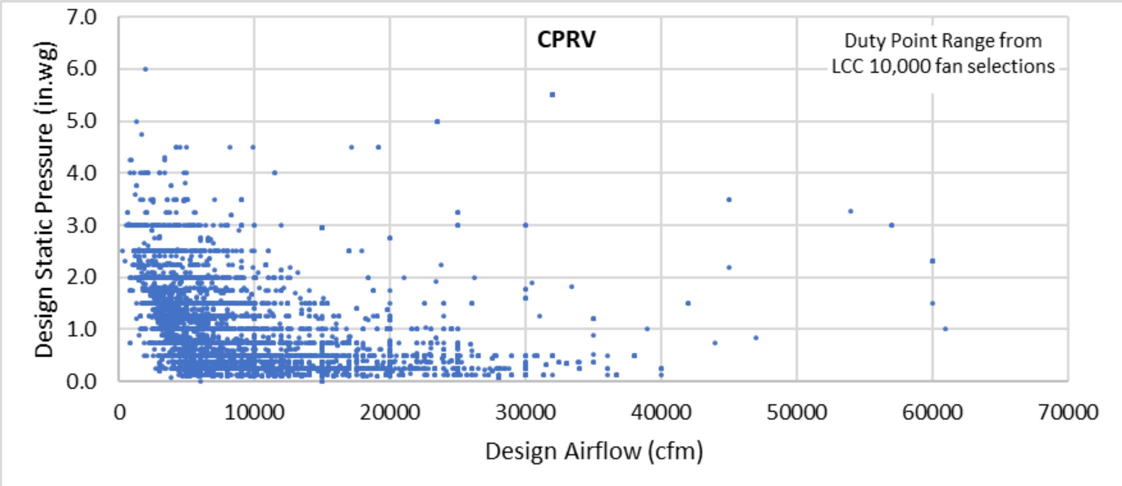


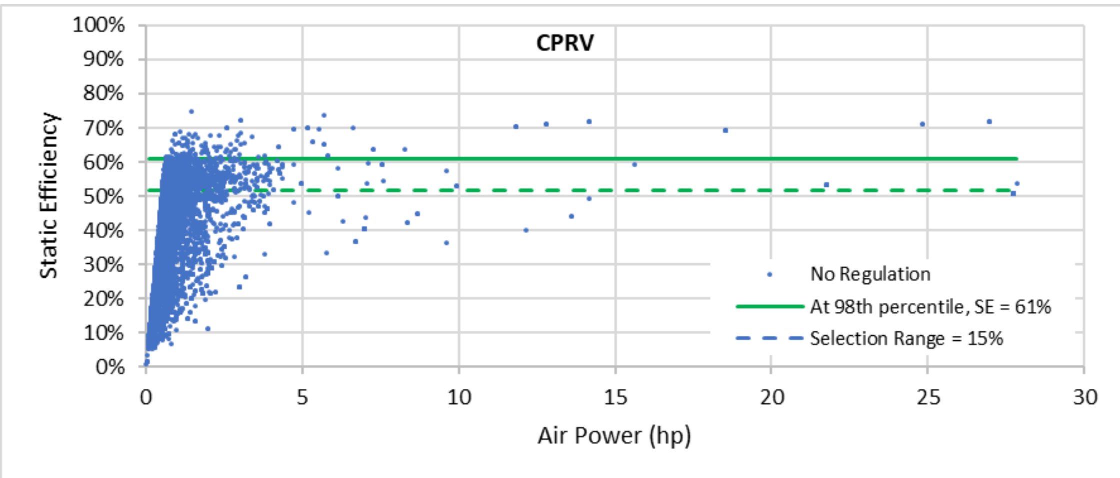


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

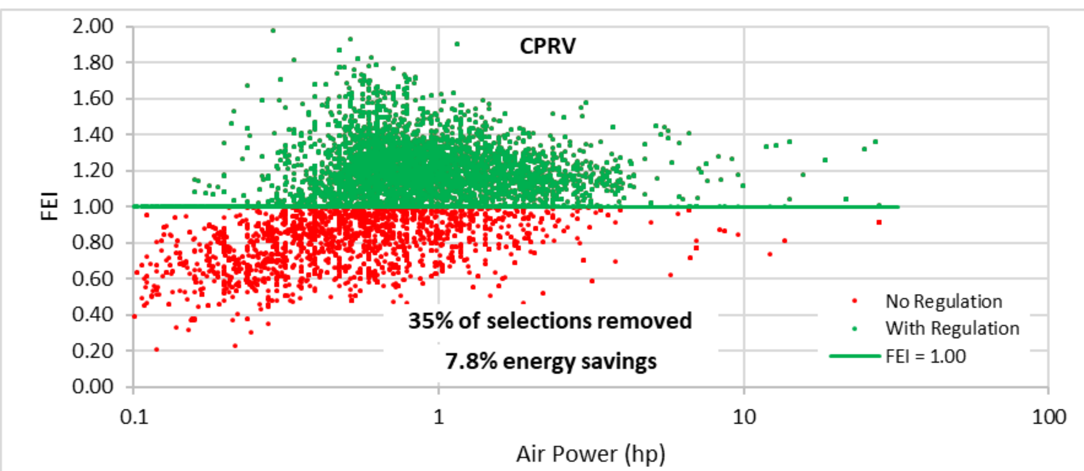


Product Class: Centrifugal Power Roof Ventilator (CPRV) Exhaust

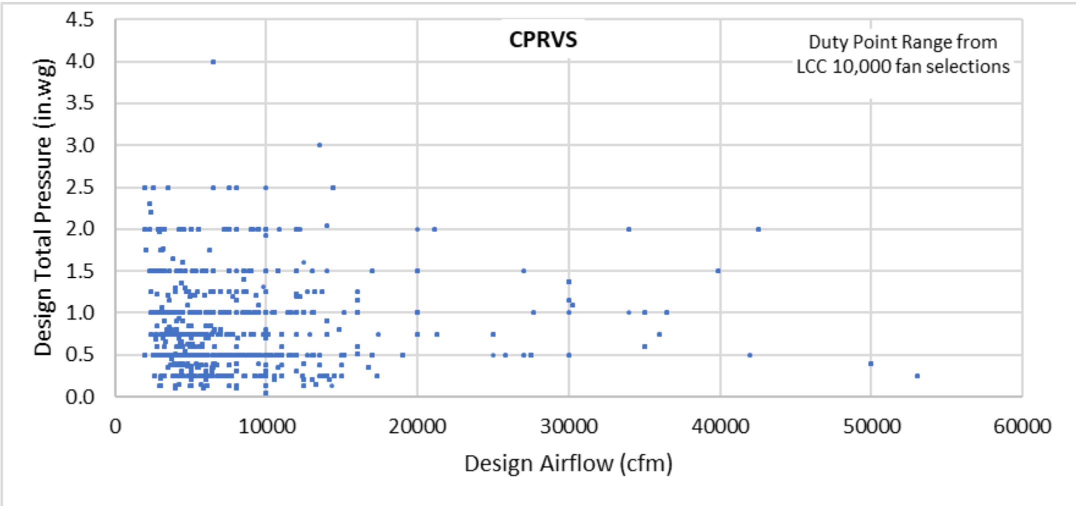


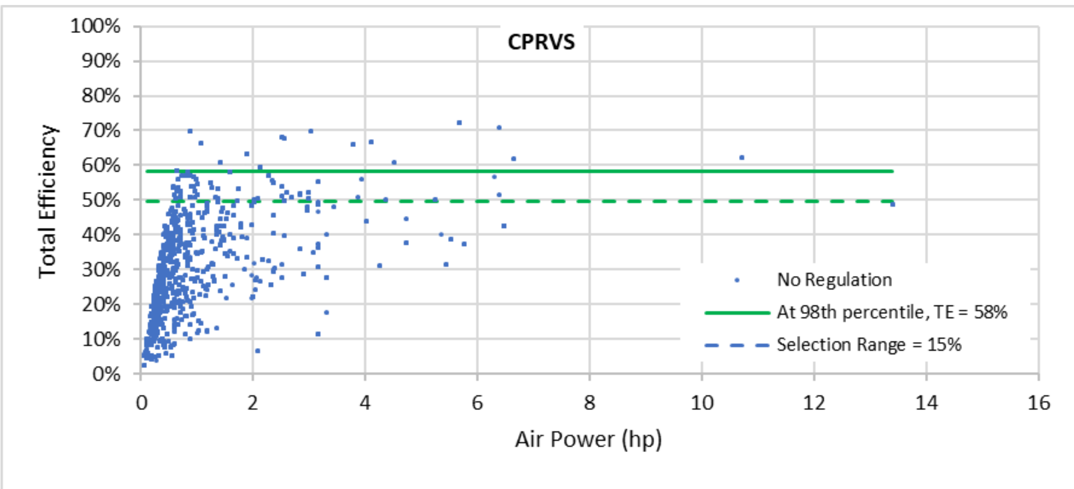


Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.

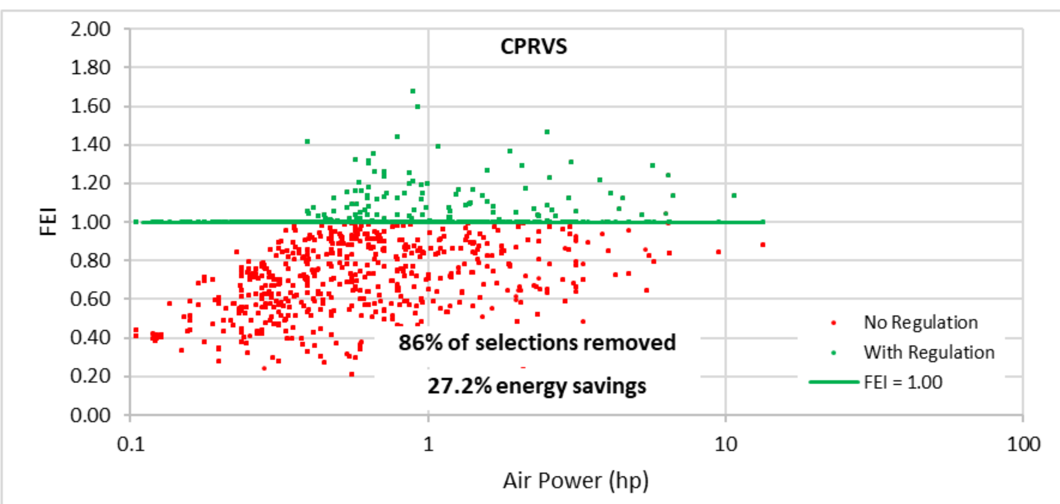


Product Class: Centrifugal Power Roof Ventilator Supply (CPRVS)





Note: Only sample points with densities between 0.07 and 0.08 were used, for clarity.



APPENDIX B – Proposed Modifications to ANSI/AMCA Standard 214 Section 6.4.2.4

The following procedures are proposed for use of modified coefficients for ANSI/AMCA Standard 214-21 Section 6.4.2.4. The objective of these modified coefficients is to obtain the same estimated motor and VFD efficiency as the modified IEC Standard 61800-9-2:2023 interpolation method outlined in the energy-standard NOPR.

Calculation Based on Shaft-to-Air Testing for Fans with Motors and Motor Controllers. The provisions of Section 6.4 of ANSI/AMCA Standard 214-21 apply. These include sections 6.4.1.4 and 6.4.2.4, which previously were excluded. Exceptions are constants *g*, *h*, and *i* determined in Section 6.4.2.4.1.2, which shall be replaced with those in revised ANSI/AMCA Standard 214-21 Table B.1 or B.2 below, and the constants *d*, *e*, and *f* determined in ANSI/AMCA Standard 214-21 Section 6.4.2.4.2.1, which shall be replaced with those in revised Table C.1 or C.2 below.

REVISED TABLE B.1. Polyphase-induction-motor performance constants (horsepower-rated motors with VFD).

HP	2 POLE			4 POLE		
	g	h	i	g	h	i
1	1.08580	0.08250	-0.03070	1.06340	0.06010	-0.02710
1.5	1.06883	0.06644	-0.02726	1.05326	0.04958	-0.02417
2	1.05840	0.05670	-0.02497	1.04678	0.04309	-0.02221
3	1.04580	0.04508	-0.02195	1.03861	0.03521	-0.01962
5	1.03313	0.03360	-0.01849	1.02982	0.02720	-0.01664
7.5	1.02534	0.02669	-0.01601	1.02398	0.02220	-0.01450
10	1.02090	0.02280	-0.01440	1.02040	0.01930	-0.01310
15	1.01600	0.01856	-0.01234	1.01612	0.01603	-0.01130
20	1.01338	0.01631	-0.01102	1.01359	0.01421	-0.01014
25	1.01178	0.01493	-0.01008	1.01190	0.01306	-0.00931
30	1.01073	0.01400	-0.00937	1.01070	0.01227	-0.00868
40	1.00948	0.01285	-0.00835	1.00909	0.01126	-0.00776
50	1.00881	0.01216	-0.00764	1.00809	0.01065	-0.00711
60	1.00843	0.01170	-0.00711	1.00742	0.01026	-0.00663
75	1.00814	0.01123	-0.00654	1.00677	0.00987	-0.00610
100	1.00800	0.01070	-0.00590	1.00620	0.00950	-0.00550
125	1.00802	0.01030	-0.00549	1.00595	0.00928	-0.00510
150	1.00809	0.00994	-0.00521	1.00587	0.00822	-0.00482
200	1.00826	0.00926	-0.00487	1.00594	0.00714	-0.00447
250	1.00839	0.00859	-0.00469	1.00619	0.00700	-0.00426
300	1.00847	0.00791	-0.00459	1.00643	0.00700	-0.00426
350	1.00850	0.00791	-0.00455	1.00874	0.00700	-0.00426
400	1.00850	0.00791	-0.00454	1.01105	0.00700	-0.00426
450	1.00850	0.00791	-0.00456	1.01185	0.00700	-0.00426
500	1.00850	0.00791	-0.00459	1.01264	0.00700	-0.00426

REVISED TABLE B.1 (continued)

HP	6 Pole			8 Pole		
	g	h	i	g	h	i
1	1.05220	0.05950	-0.01630	1.11330	0.11030	-0.03910
1.5	1.04237	0.04825	-0.01499	1.08772	0.08840	-0.03372
2	1.03639	0.04150	-0.01411	1.07273	0.07519	-0.03039
3	1.02924	0.03355	-0.01294	1.05555	0.05953	-0.02632
5	1.02216	0.02582	-0.01160	1.03962	0.04424	-0.02210
7.5	1.01789	0.02125	-0.01063	1.03074	0.03514	-0.01936
10	1.01550	0.01870	-0.01000	1.02610	0.03010	-0.01770
15	1.01291	0.01593	-0.00918	1.02140	0.02469	-0.01568
20	1.01156	0.01444	-0.00865	1.01907	0.02187	-0.01443
25	1.01075	0.01350	-0.00827	1.01768	0.02017	-0.01354
30	1.01022	0.01284	-0.00798	1.01673	0.01905	-0.01157
40	1.00960	0.01195	-0.00756	1.01542	0.01767	-0.01065
50	1.00927	0.01134	-0.00726	1.01446	0.01687	-0.01107
60	1.00908	0.01087	-0.00703	1.01361	0.01633	-0.00996
75	1.00892	0.01030	-0.00678	1.01243	0.01577	-0.00970
100	1.00880	0.00950	-0.00650	1.01050	0.01510	-0.00870
125	1.00874	0.00950	-0.00631	1.00856	0.01455	-0.00789
150	1.00870	0.00950	-0.00617	1.00856	0.01455	-0.00789
200	1.00860	0.00950	-0.00599	1.00856	0.01455	-0.00789
250	1.00848	0.00950	-0.00589	1.00856	0.01455	-0.00789
300	1.00990	0.00950	-0.00589			
350	1.01150	0.00950	-0.00589			
400						
450						
500						

REVISED TABLE B.2. Polyphase-induction-motor performance constants (kilowatt-rated motors with VFD).

kW	2 POLE			4 POLE		
	g	h	i	g	h	i
0.75	1.08556	0.08227	-0.03065	1.06326	0.05995	-0.02706
1.1	1.06953	0.06710	-0.02740	1.05368	0.05001	-0.02429
1.5	1.05822	0.05654	-0.02493	1.04666	0.04298	-0.02217
2.2	1.04632	0.04556	-0.02207	1.03895	0.03554	-0.01973
3	1.03852	0.03849	-0.01996	1.03356	0.03061	-0.01791
4	1.03178	0.03240	-0.01806	1.02881	0.02633	-0.01627
5.5	1.02566	0.02697	-0.01611	1.02422	0.02241	-0.01459
7.5	1.02083	0.02274	-0.01437	1.02034	0.01925	-0.01307
11	1.01620	0.01873	-0.01242	1.01630	0.01616	-0.01137
15	1.01334	0.01627	-0.01100	1.01355	0.01418	-0.01012
18.5	1.01184	0.01498	-0.01011	1.01196	0.01310	-0.00934
22	1.01083	0.01409	-0.00944	1.01081	0.01234	-0.00874
30	1.00946	0.01283	-0.00833	1.00906	0.01124	-0.00774
37	1.00883	0.01218	-0.00766	1.00812	0.01067	-0.00713
45	1.00842	0.01169	-0.00710	1.00740	0.01025	-0.00662
55	1.00816	0.01127	-0.00658	1.00682	0.00990	-0.00614
75	1.00800	0.01069	-0.00589	1.00619	0.00949	-0.00549
90	1.00802	0.01036	-0.00555	1.00599	0.00931	-0.00516
110	1.00808	0.00997	-0.00524	1.00588	0.00832	-0.00485
132	1.00819	0.00955	-0.00501	1.00591	0.00760	-0.00462
160	1.00830	0.00905	-0.00481	1.00602	0.00710	-0.00440
200	1.00842	0.00833	-0.00465	1.00628	0.00700	-0.00426
250	1.00849	0.00791	-0.00456	1.00810	0.00700	-0.00426
315	1.00850	0.00791	-0.00455	1.01142	0.00700	-0.00426
355	1.00850	0.00791	-0.00458	1.01227	0.00700	-0.00426
375	1.00850	0.00791	-0.00459	1.01268	0.00700	-0.00426

REVISED TABLE B.2 (continued)

kW	6 Pole			8 Pole		
	g	h	i	g	h	i
0.75	1.05206	0.05934	-0.01628	1.11294	0.10999	-0.03902
1.1	1.04278	0.04871	-0.01504	1.08878	0.08930	-0.03394
1.5	1.03629	0.04139	-0.01409	1.07249	0.07497	-0.03033
2.2	1.02953	0.03388	-0.01299	1.05626	0.06018	-0.02649
3	1.02517	0.02911	-0.01217	1.04640	0.05075	-0.02390
4	1.02142	0.02503	-0.01143	1.03808	0.04266	-0.02163
5.5	1.01807	0.02144	-0.01067	1.03111	0.03552	-0.01947
7.5	1.01546	0.01866	-0.00999	1.02603	0.03002	-0.01767
11	1.01302	0.01604	-0.00921	1.02159	0.02491	-0.01576
15	1.01154	0.01442	-0.00864	1.01903	0.02183	-0.01441
18.5	1.01078	0.01353	-0.00828	1.01773	0.02023	-0.01357
22	1.01027	0.01290	-0.00801	1.01682	0.01915	-0.01175
30	1.00959	0.01193	-0.00755	1.01540	0.01765	-0.01066
37	1.00928	0.01136	-0.00727	1.01449	0.01690	-0.01106
45	1.00908	0.01086	-0.00702	1.01358	0.01632	-0.00995
55	1.00893	0.01034	-0.00680	1.01252	0.01581	-0.00972
75	1.00880	0.00950	-0.00650	1.01045	0.01509	-0.00868
90	1.00875	0.00950	-0.00634	1.00886	0.01464	-0.00802
110	1.00870	0.00950	-0.00618	1.00856	0.01455	-0.00789
132	1.00864	0.00950	-0.00607	1.00856	0.01455	-0.00789
160	1.00856	0.00950	-0.00596	1.00856	0.01455	-0.00789
200	1.00903	0.00950	-0.00589	1.00856	0.01455	-0.00789
250	1.01105	0.00950	-0.00589			
315						
355						
375						

REVISED TABLE C.1. VFD performance constants (horsepower capacity).

HP	d	e	f
1	0.98960	0.06390	-0.00988
1.5	0.99135	0.05151	-0.00973
2	0.99132	0.04399	-0.00962
3	0.99016	0.03501	-0.00947
5	0.98781	0.02614	-0.00927
7.5	0.98580	0.02080	-0.00912
10	0.98448	0.01780	-0.00901
15	0.98288	0.01454	-0.00886
20	0.98196	0.01281	-0.00875
25	0.98137	0.01176	-0.00867
30	0.98097	0.01105	-0.00860
40	0.98046	0.01019	-0.00849
50	0.98015	0.00969	-0.00841
60	0.97994	0.00937	-0.00834
75	0.97972	0.00904	-0.00825
100	0.97946	0.00870	-0.00814
125	0.97923	0.00845	-0.00806
150	0.97902	0.00822	-0.00799
200	0.97858	0.00780	-0.00788
250	0.97864	0.00780	-0.00788
300	0.97867	0.00780	-0.00788
350	0.97870	0.00780	-0.00788
400	0.97872	0.00780	-0.00788
450	0.97873	0.00780	-0.00788
500	0.97874	0.00780	-0.00788

REVISED TABLE C.2. VFD performance constants (kilowatt capacity).

kW	d	e	f
0.75	0.98963	0.06372	-0.00988
1.1	0.99128	0.05202	-0.00973
1.5	0.99130	0.04386	-0.00962
2.2	0.99021	0.03538	-0.00947
3	0.98881	0.02992	-0.00936
4	0.98746	0.02522	-0.00925
5.5	0.98589	0.02102	-0.00913
7.5	0.98446	0.01775	-0.00901
11	0.98294	0.01467	-0.00887
15	0.98194	0.01278	-0.00875
18.5	0.98139	0.01179	-0.00867
22	0.98100	0.01112	-0.00860
30	0.98045	0.01018	-0.00849
37	0.98016	0.00971	-0.00841
45	0.97994	0.00936	-0.00833
55	0.97974	0.00907	-0.00826
75	0.97945	0.00869	-0.00814
90	0.97927	0.00849	-0.00807
110	0.97904	0.00825	-0.00800
132	0.97877	0.00798	-0.00793
160	0.97860	0.00780	-0.00788
200	0.97865	0.00780	-0.00788
250	0.97869	0.00780	-0.00788
315	0.97872	0.00780	-0.00788
355	0.97874	0.00780	-0.00788
375	0.97874	0.00780	-0.00788

APPENDIX C – Supplemental Information for Proposed Exemption of Unshrouded Radial Fans

Industrial-process fans—dust- and material-handling fans in particular—are challenged to achieve high efficiency levels. Their applications require high air-stream velocities and rugged impellers, which degrade the efficiency capability of designs.

Figure C.1 shows two examples of industrial-process material-handling fan impellers. The one on the left is designed for a rugged application, such as moving aluminum cans or wood or even metal chips. Although the wear may seem dramatic, the impeller has served its useful life and is ready for replacement. The example on the right is that of a high-efficiency radial fan misapplied into an application that has abrasive fly-ash materials in the air stream. The blade was eaten away in very short order (less than a month).

FIGURE C.1. Left: normal wear and tear on an unshrouded radial fan. Right: breakage within a shrouded radial fan installed in an application for which an unshrouded radial fan would have been better-suited.

These examples show the impacts of rugged applications—even on fans suited for those applications—and that higher-efficiency radial impellers cannot be applied in such environments.

Velocity Requirements in Material Handling

High velocities are required to keep materials suspended in an air stream, as opposed to settling at the bottom of a duct or fan casing (Table C.1).

Dust Collecting and Fume Removal Duct Velocities

Material	Velocity in FPM	Material	Velocity in FPM
1. Grinding Dust	5000		
2. Foundry Dust	4500	20. Jute Dust	3500
3. Sand Blast Dust	4000	21. Grain Dust	3000
4. Wood Flour	2000	22. Shoe Dust	4000
5. Sander Dust	2000	23. Rubber Dust	3500
6. Shavings, Dry	3000	24. Rubber Buffings	4500
7. Shavings, Wet	4000	25. Bakelite Moulding	
8. Sawdust, Dry	3000	Powder	3500
9. Sawdust, Wet	4000	26. Bakelite Moulding	
10. Wood Blocks	4500	Dust	2500
11. Hog Waste	4500	27. Oven Hood	2000
12. Buffing Lint, Dry	3000	28. Tail Pipe Exhaust	3000
13. Buffing Lint, Wet	4000	29. Melting Pot and	
14. Metal Turnings	5000	Furnace	2000
15. Lead Dust	5000	30. Metallizing Booth	3500
16. Cotton	3000	31. Soldering Fumes	2000
17. Cotton Lint	2000	32. Paint Spray	2000
18. Wool	4000	33. Carbon Black	3500
19. Jute Lint	3000	34. Paper	3500

Figure 1

Material Conveying Duct Velocities

Material	Velocity in FPM	Material	Velocity in FPM
1. Wood Chips	4500	12. Cotton	4000
2. Rags	4500	13. Wool	4500
3. Ground Feed	5000	14. Jute	4500
4. Powdered Coal	4000	15. Hemp	4500
5. Sand	7500	16. Vegetable Pulp,	
6. Wood Flour	4000	Dry	4500
7. Sawdust	4000	17. Paper	5000
8. Hog Waste	4500	18. Flour	3500
9. Pulp Chips	4500	19. Salt	6000
10. Wood Blocks	5000	20. Grain	5000
11. Cement	6000	21. Coffee Beans	3500
		22. Sugar	6000

Figure 4

TABLE C.1. Industrial-process duct velocities. Source: Engineering Letter EL-09, The New York Blower Co.

HVAC equipment typically is designed to keep air velocity well below 2,500 fpm to avoid excessive losses from pressure drop and to keep noise to a minimum, considering most HVAC applications have people in the reasonably near vicinity.

Figure C.2 shows a material-handling fan that did not maintain air-stream velocity at a level high enough to keep material suspended. The material settled at the bottom of the fan casing and eventually clogged the fan inlet and duct system.

FIGURE C.2. Material settled at the bottom of a fan casing. Image courtesy of The New York Blower Co.

BHP Increase Attributed to Material Handling

Another negative impact on the FEI calculation is the additional power required to move a dust- or material-laden air stream compared with a clean-air application. The additional power required to move the material in the air stream results in a lower FEI value.

The increase in horsepower can be estimated by including a material-loading factor. For example:

Estimating horsepower in a material-handling application:

Actual airflow = 26,000 acfm

Material loading = 19,500 lb/hr

Inlet density = 0.05 lbm/ft^3

BHP at conditions = 112 BHP

Mass flow rate of air = 26,000 * 0.05 = 1,300 lbm/min

Mass flow rate of material = 19,500/60 = 325 lbm/min

Material loading = 325/1,300 = 0.25

BHP correction = 1 + material loading = 1.25

BHP with material = 112 x 1.25 = 140 BHP

Loss of Utility by Including Material-Handling Fans with all Other Radial Impellers

Rugged material-handling designs and higher-efficiency shrouded radial designs are in the radial-equipment class. AMCA has had conversations about separating the products from their application. The current discussion involves describing characteristics of a particular set of impeller designs that can be segregated within the radial-equipment class. Higher-performing radial impeller designs can be left in the radial-equipment class to be assessed with the broader range of radial equipment under an FEI requirement.

Impeller designs specific to The New York Blower Co. with their typical applications are shown in Figure C.3.

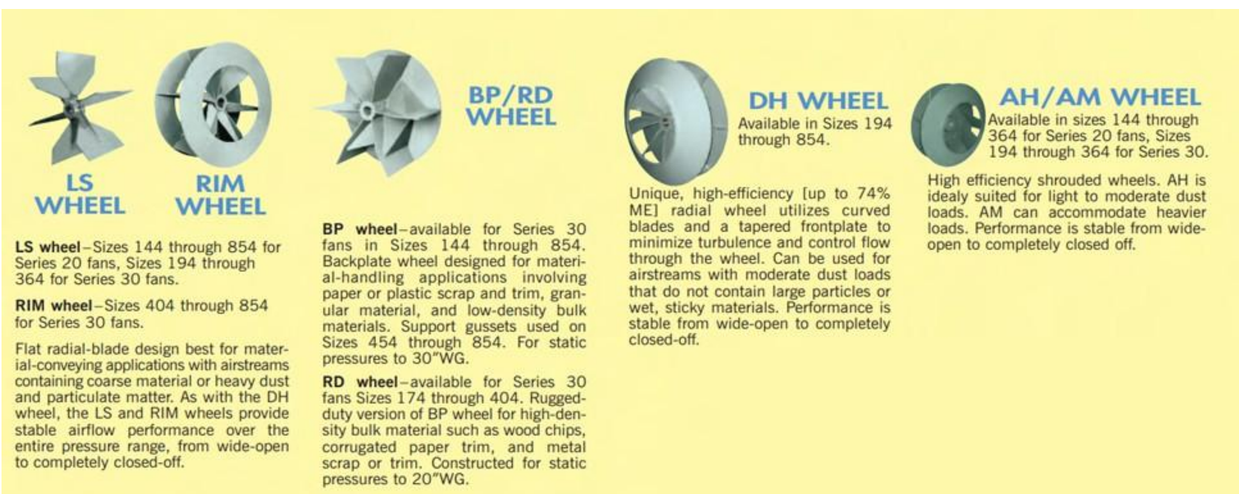


FIGURE C.3. Impeller designs. Courtesy of The New York Blower Co.

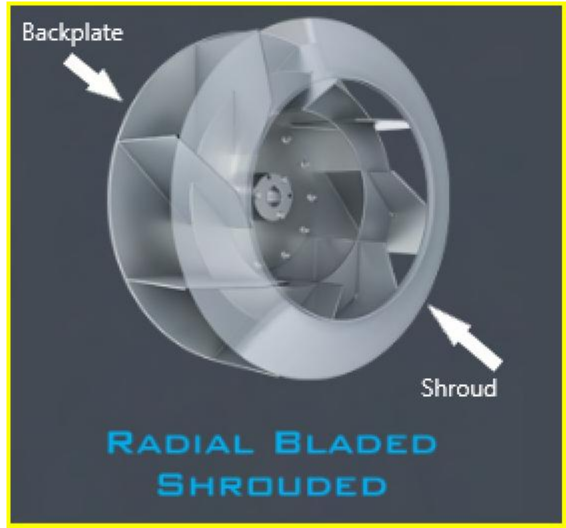
Wheel Type (increasing efficiency capability)	Material Handling Application and/or Market
RD	Heavy objects (aluminum cans/metal scrap) (rugged design)
BP	Pneumatic Conveying (Paper industry, pulp mill)
LS	Material Handling/Long Strands, Small Particles, Dust or sticky particles, Abrasive (diaper industry, wood industry-cutting, fiberglass, roofing products, gypsum)
RIM	Pollution Control / Coarse Material / Heavy Dust
AM	Moderate Dust / Air slides
DH	Food Dryers / Aeration / Process cooling / Fluidized systems / Oven exhaust / Moderate Dust
AH	Light Dust / Dust control / combustion air / farm aeration / Industrial dryers

TABLE C.2. Typical applications by wheel type corresponding to Figure C.3.

Impeller designs generally can be grouped into two categories: shrouded and unshrouded. In Figure C.3, the LS and BP/RD designs clearly are unshrouded. The DH and AH/AM (along with other designs referred to as radial-tipped—not shown) clearly are shrouded.

The RIM impeller is a design common to the industry. One can see the clear rims placed on the outside of the impeller. While these rims may look like shrouds, their purpose is mechanical—to increase the impeller's structural integrity—rather than aerodynamic. This design can be clearly distinguished from, for example, a radial shrouded fan impeller by the inclusion of a solid backplate (Figure C.4).

Radial unshrouded



Radial shrouded

FIGURE C.4. RIM-type unshrouded radial fan and typical shrouded radial fan.

If the current exemption for unshrouded radial impellers were expanded to include the RIM-style impeller and extended to all sizes, material-handling fans could be accommodated. Consequently, AMCA's ask is to extend the exemption for radial, unshrouded impellers to include all sizes (or just remove the size limitation) and classify a RIM-style radial impeller as a radial, unshrouded impeller design. This will exempt a measurable portion of the radial-equipment class; however, the exemption would be limited to industrial-process/material-handling applications.

AMCA believes the risk of substitution of an unshrouded impeller in a clean-air-handling application to avoid regulation to be extremely low. While the material cost may be lower in very inexpensive applications, the inefficiency of the design and the power curve of the radial impeller would quickly make such an application economically unfeasible. Applying a shroud in these applications would return the design to being in the scope of the regulation. As such, AMCA believes the exemption would not create a loophole.

To examine the impact of separating unshrouded radial fans from the population of the radial-housed equipment class, The New York Blower Co. and Twin City Fan Companies Ltd. examined selections from fiscal year 2023. Figure C.5 indicates that, with FEI being a duty-point metric, FEI comparisons are best executed at similar duty points of flow and pressure. Because this approach creates an overly complicated analysis, air power is used as a proxy for duty point. FEI and efficiency values are plotted against air power.

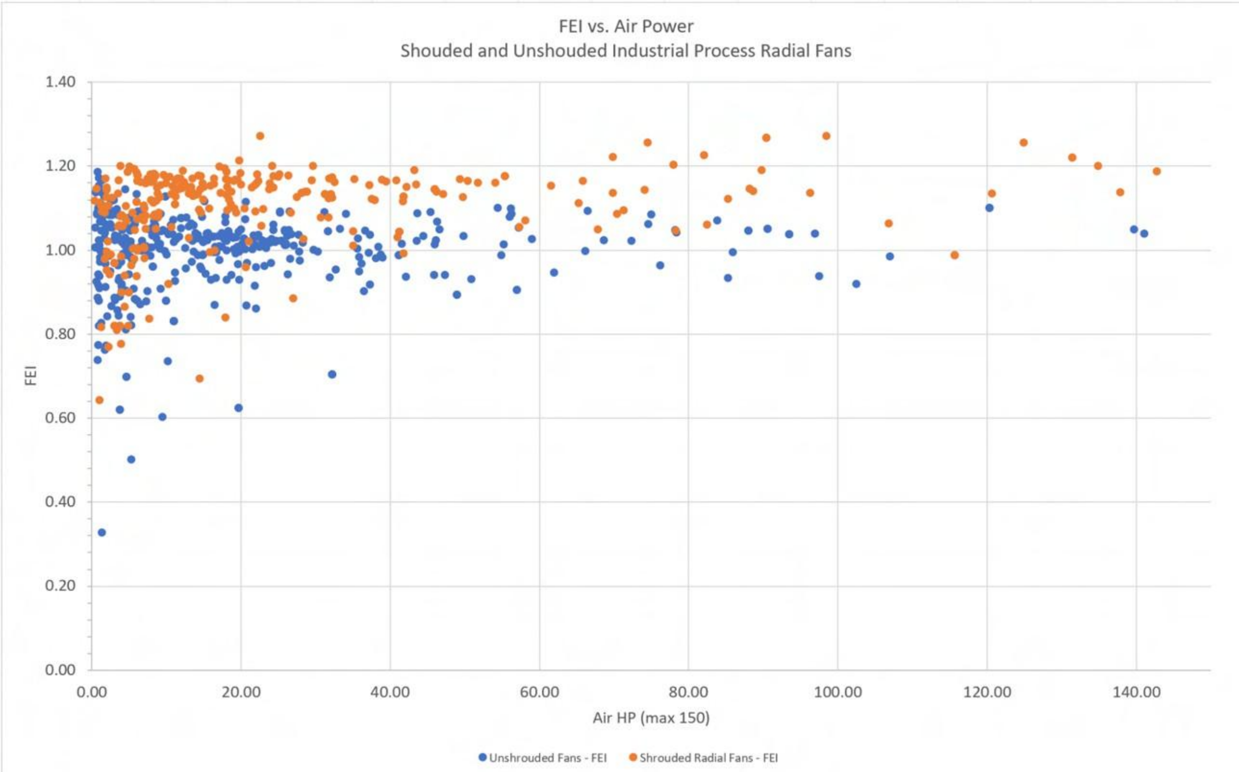


FIGURE C.5. FEI vs. air power, shrouded and unshrouded industrial-process radial fans.

Shrouded radial industrial-process fans are shown in orange. Unshrouded radial industrial-process fans are shown in blue. One can observe the clear distinction between the capabilities of the shrouded radial fans to achieve higher FEI levels. A similar analysis using total efficiency is shown in Figure C.6.

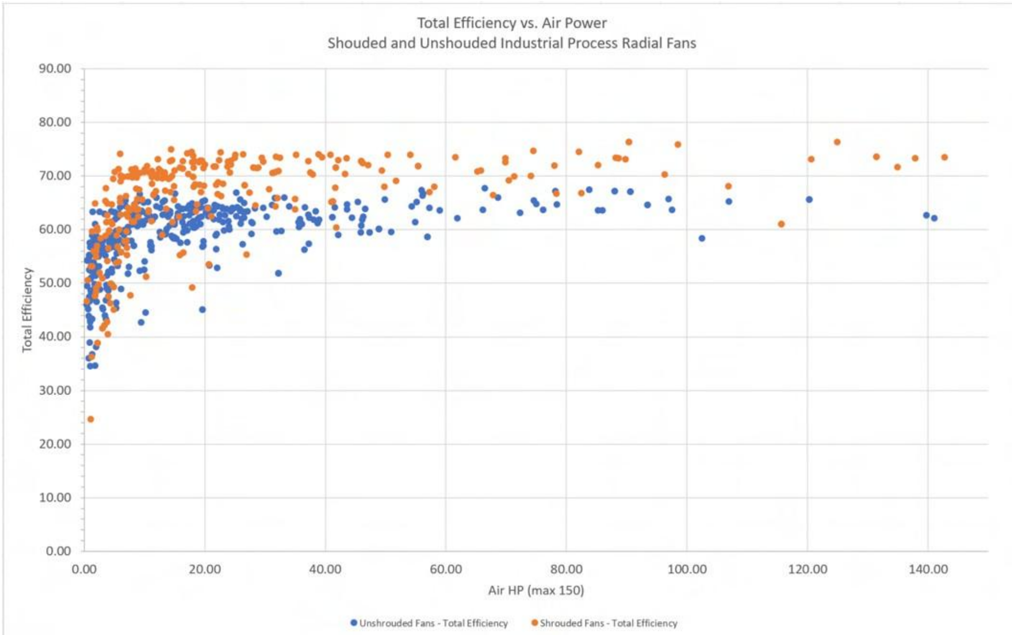


FIGURE C.6. Total efficiency vs. air power, shrouded and unshrouded industrial-process radial fans.

Figure C.6 clearly shows the distinction between the efficiency capabilities of shrouded and unshrouded radial industrial-process fans. Figure C.6 also shows size effect is prevalent in lower-air-power solutions. One can see the “max tech” available across the power spectrum is appreciably lower than with the remaining radial designs.

To reiterate, AMCA requests an exemption be implemented for unshrouded radial-impeller designs, the equipment class with unshrouded radial designs be studied, and the issue be reviewed in a future round of regulatory review.

APPENDIX D – Derivation of Equation 1 for ACF

Base Equations

Circulating-Fan Efficacy

(ANSI/AMCA Standard 230-23, Page 14, Equation 8.16)

$$Eff_{circ} = \frac{Q_0}{W_E} \qquad \text{I-P} \quad [1]$$

Where:
Q_0 = fan airflow rate, cfm
W_E = electrical input power, W

Overall Fan Efficiency

$$\eta = \frac{W_O}{W_E} \qquad \text{I-P} \quad [2]$$

Where:
W_O = output airflow power, W
W_E = electrical input power, W

Output Airflow Power

$$W_O = \frac{745.7 * (Q_0 * P_t)}{6343} \qquad \text{I-P} \quad [3]$$

Where:
W_O = output airflow power, W
Q_0 = fan airflow rate, cfm
P_t = total fan pressure, in. w.g.
Conversion factor = 745.7 W / 1 hp

Note: The coefficient 745.7 is applied to convert the units of output airflow power from the native unit of horsepower (hp) to Watts (W).

Fan Total Pressure

$$P = \rho * \left(\qquad \right)^2 \; \frac{\quad}{\quad} \qquad Q_0$$

$1097.8 * A$

I
-
P
[
4
]

Where:
A = fan outlet or discharge area, ft^2
P_t = total fan pressure, in. w.g.
Q_0 = fan airflow rate, cfm
ρ_0 = air density, lbm/ft^3

Fan Swept Area

$$A = \pi * \frac{\left(\frac{D}{12}\right)^2}{4} \qquad \text{I-P} \quad [5]$$

Where:
A = fan outlet or discharge area, ft^2
D = fan impeller diameter, ft

<u>Derivation of Final Equation</u>

Beginning with Equation 1:

$$Eff_{circ} = \frac{Q_0}{W_E}$$

Equation 2 can be rewritten as:

$$W_E = \frac{W_O}{\eta}$$

And substituted in, resulting in:

$$Eff_{circ} = \frac{Q_0}{\frac{W_O}{\eta}}$$

Which can be simplified to:

$$Eff_{circ} = \frac{Q_0 * \eta}{W_O}$$

Equation 3 then can be substituted for the denominator, resulting in:

$$Eff_{circ} = \frac{Q_0 * \eta}{\frac{745.7 * (Q_0 * P_t)}{6343}}$$

Which can be simplified to:

$$Eff_{circ} = \frac{6343 * Q_0 * \eta}{745.7 * (Q_O * P_t)}$$

Q_0 is a multiple in the numerator and denominator and, thus, can be canceled out, resulting in:

$$Eff_{circ} = \frac{6343 * \eta}{745.7 * P_t}$$

At this point, the equation for fan total pressure (Equation 4) can be substituted in as well. This results in the following equation:

$$Eff_{circ} = \frac{6343 * \eta}{745.7 * \rho_0 * \left(\frac{Q_0}{1097.8 * A}\right)^2}$$

The square in the denominator can be distributed and the function simplified to:

$$Eff_{circ} = \frac{6343 * 1097.8^2 * A^2 * \eta}{745.7 * \rho_0 * Q_0^2}$$

The equation is now written in terms of efficiency, airflow, and area. Area can be converted to diameter using Equation 5:

$$Eff_{circ} = \frac{6343 * 1097.8^2 * \left(\pi * \frac{\left(\frac{D}{12}\right)^2}{4}\right)^2 * \eta}{745.7 * \rho_0 * Q_0^2}$$

The inner square can be distributed:

$$Eff_{circ} = \frac{6343 * 1097.8^2 * \left(\pi * \frac{\frac{D^2}{144}}{4}\right)^2 * \eta}{745.7 * \rho_0 * Q_0^2}$$

And then simplified:

$$Eff_{circ} = \frac{6343 * 1097.8^2 * \left(\pi * \frac{D^2}{576}\right)^2 * \eta}{745.7 * \rho_0 * Q_0^2}$$

Then, the outer square can be distributed:

$$Eff_{circ} = \frac{6343 * 1097.8^2 * \pi^2 * \frac{D^4}{576^2} * \eta}{745.7 * \rho_0 * Q_0^2}$$

The resulting equation can be simplified to:

$$Eff_{circ} = \frac{6343 * 1097.8^2 * \pi^2 * D^4 * \eta}{745.7 * 576^2 * \rho_0 * Q_0^2}$$

To further simplify this formula, we can assume a value of 3.14159 for π, and all of the constants can be combined into a single coefficient, rounded to two decimal places:

$$Eff_{circ} \approx \frac{304.95 * D^4 * \eta}{\rho_0 * Q_0^2}$$

This equation calculates the efficacy of a circulating fan in cfm/W using airflow, impeller diameter, and efficiency as inputs. At this point, the equation still allows the user to input air density at the test location.

By assuming a standard air density, ρ_{std}, of 0.075 lbm/ft^3 (ANSI/AMCA Standard 230-23, Page 4, Section 4.1.6.3), the variable ρ_0 can be replaced with a constant. The coefficient then can be simplified into a single term again:

$$Eff_{circ} \approx \frac{4066 * D^4 * \eta}{Q_0^2}$$

Appendix E – Fan-Guard References for ACF

OSHA References

1910.212(a)(5)
Exposure of blades. When the periphery of the blades of a fan is less than seven (7) feet above the floor or working level, the blades shall be guarded. The guard shall have openings no larger than one-half (½) inch.

https://www.osha.gov/laws-regs/regulations/standardnumber/1910/1910.212

The subject paragraph means all fans within 7 ft of the floor or working level must be guarded. The guard must not have openings greater than one-half inch in width. The use of concentric rings with spacing between them not exceeding one-half inch is acceptable, provided that sufficient radial spokes and firm mountings are used to make the guard rigid enough to prevent it from being pushed into the fan blade during normal use. The use of nylon mesh or similar materials with holes not exceeding one-half inch to modify a substandard guard on an existing fan is acceptable, provided the combination of the two provides adequate protection so that the mesh cannot be pushed into the fan blade during normal use.

https://www.osha.gov/enforcement/directives/std-01-12-001

UL 507 10th Edition (8.23.34) References

9.1 General
9.1.1 The rotor of a motor, a pulley, a belt, a gear, an impeller, or other moving part shall be enclosed, guarded,or installed at sufficient height per 9.2, Portable Fans and Window Fans, or 9.3, Stationary Fans and Permanently Connected Fans, as applicable to reduce the risk of injury.
……
9.2 Portable Fans And Window Fans
……
9.2.5 Conventional designs of impellers meet the requirement of being guarded when:
……
b) The guarding is such that the probe illustrated in Figure 9.1 [Figure E.1 below] cannot touch the leading edge of the blade and when inserted as described in 9.2.6 ….

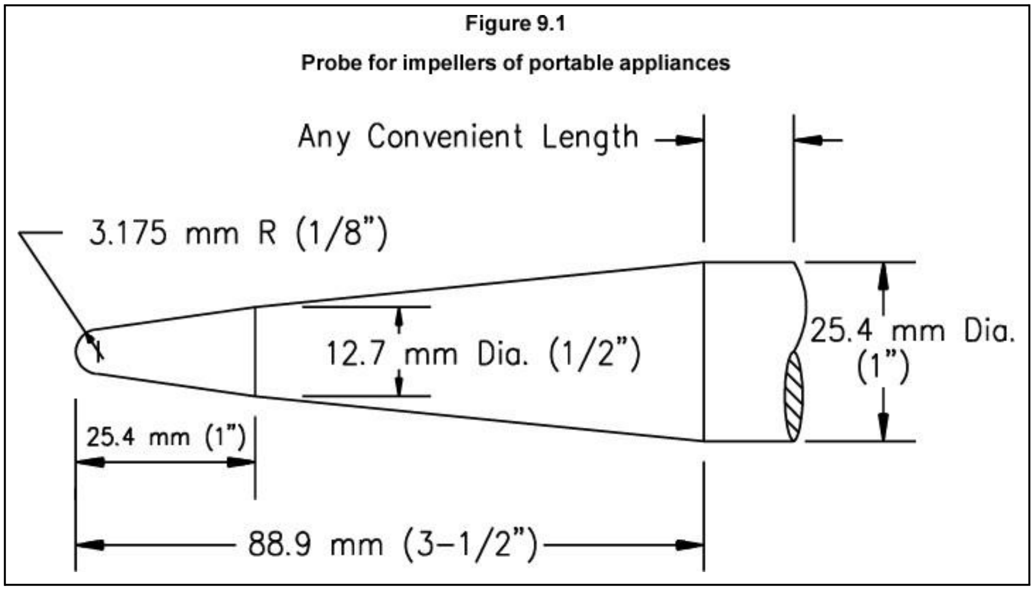


FIGURE E.1. **Probe for impellers of portable appliances.**

9.3 Stationary Fans And Permanently Connected Fans

….

9.3.2 The impeller of a stationary or permanently connected fan shall be constructed so that it cannot be contacted by the probe illustrated in Figure 9.2 [Figure E.2 below].

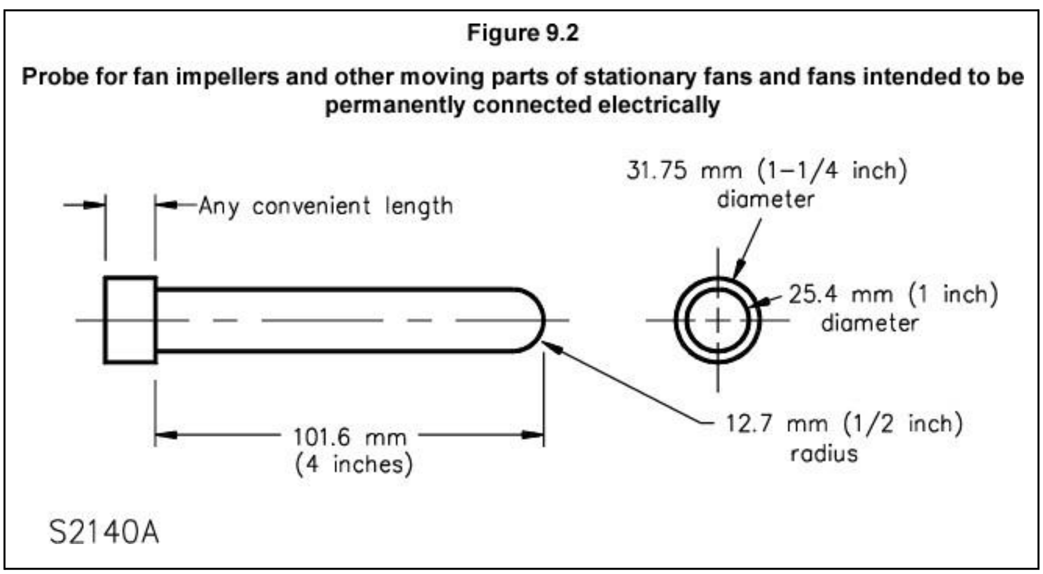


FIGURE E.2. **Probe for fan impellers and other moving parts of stationary fans and fans intended to be permanently connected electrically.**