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Understanding fan system effects

The air moving industry uses a common term to describe certain inlet and outlet conditions that adversely affect fan performance. The term used is "system effect". Perhaps the term should be "fan installation effect", because system effect results from the difference in how the fan was tested, compared to how it is installed. To minimize system effects, air must enter or leave a fan uniformly.

First let's take a look at how fans are tested and cataloged. Most fans available in today's market bear AMCA Certified Rating Seals. This means that the fan manufacturer followed the test procedures as outlined in AMCA Publication 210 and tested the fan in one of the standardized configurations approved by AMCA. One of the requirements of AMCA, is that directly under the cataloged performance for a given fan model, the fan manufacturer must make a statement as to how that product was tested. Paying attention to these statements is the first step in avoiding system effect problems.

Typical statements for three different product types are:

Roof exhaust fans: Performance shown is for Installation Type A: Free inlet, Free outlet. Power rating (BHP) does not include drive losses. Performance ratings do not include the effects of appurtenances in the airstream.

Tube axial fans: Performance shown is for Installation Type B: Free inlet, ducted outlet. Power rating (BHP) does not include drive losses. Performance ratings do not include the effects of appurtenances in the airstream.

Centrifugal fans: Performance shown is for Installation Type B: Free inlet, ducted outlet. Power rating (BHP) does not include drive losses. Performance ratings do not include the effects of appurtenances in the airstream.

It is important to realize that fan manufacturers can only guarantee the fan to perform as tested. In the examples shown, not one of the fans were tested with obstructions, such as elbows, guards or dampers, directly at the fan inlet or outlet.

These obstructions cause additional losses that are not included in the fan manufacturer's tests, and in many cases, are not included in the designers' usual system resistance calculations. Most designers are well trained in determining the resistance that occurs in the system's ducts, filters, dampers and elbows that are located some distance from the fan, but they pay little attention to obstructions near the fan. The interaction of the air and the obstruction just prior to the fan causes the additional losses known as system effects.

The following figures illustrate how fans are tested in comparison to how they are sometimes installed.

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Roof Exhaust Fans

Figure 1 illustrates how roof exhausters are tested. Additional vertical straight duct would have little if any effect. Figures 2 and 3 illustrate roof exhaust fan installations having system effects. Figure 3 illustrates the worst case, because the damper is located in a turbulent airstream. To improve on installations where horizontal ducts are used directly under the roof line, turning vanes should be installed in the elbows. In addition, a higher curb or extended base should be used. Higher curbs result in the elbow being further from the damper and fan inlet.

Fig. 1

Good

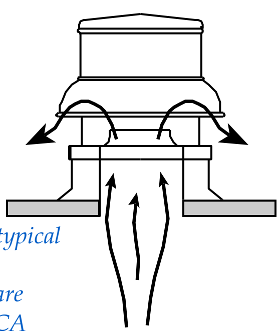

Figure 1 is typical of how roof exhausters are tested. AMCA Publication refers to this set up as "Type A: Free inlet, free outlet."

Fig. 2

Poor

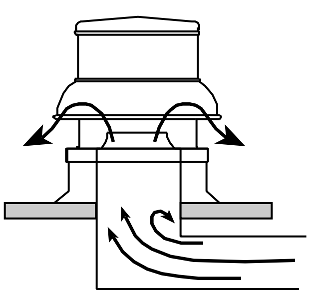

Figure 2 illustrates a poor installation with horizontal duct and an abrupt elbow at the fan inlet causing system effect.

Fig. 3

Poor

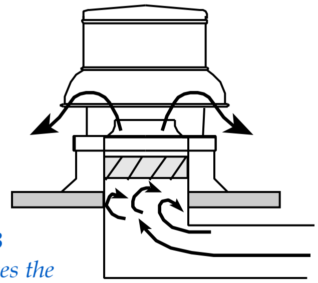

Figure 3 illustrates the same installation as figure 2 with the addition of a damper causing even greater system effect.

Tube Axial Fans

Figure 4 illustrates how tube axial fans are tested. Installations with straight inlet ducts and inlet bells would result in similar performance. Without a discharge duct, a system effect will occur. (See figure 12 on page 4 for recommended discharge.) Inline installations are subject to system effect both at the fan inlet and outlet, as shown in Figures 5 and 6. Figure 5 illustrates a poor inlet condition with an elbow directly at the fan inlet. Figure 6 illustrates a poor outlet condition where the fan discharges too close to a wall. Inline fans require the appropriate length of discharge duct in order to achieve cataloged performance. (Refer to figure 12 on page 4.)

Fig. 4

Good

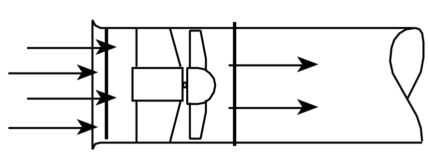

Figure 4 is typical of how tube axial fans are tested. AMCA refers to this set-up as "Type B: Free inlet, ducted outlet."

Fig. 5

Poor

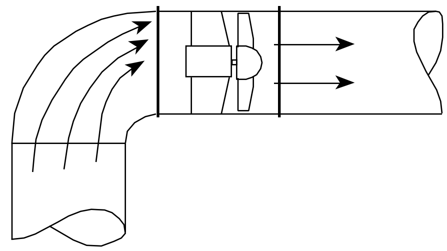

Figure 5 illustrates a poor installation with an elbow directly at the fan inlet. The air entering the fan is forced to one side.

Fig. 6

Poor

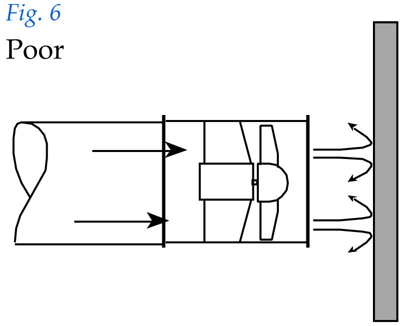

Figure 6 illustrates a poor installation with the fan located too close to a wall.

Centrifugal Fans

Figure 7 illustrates how housed centrifugal fans are tested. Centrifugal fan installations are subject to the greatest possibilities of system effect due to the possibilities of both ducted inlets and outlets, plus multiple available arrangements, discharge positions, and CW or CCW rotations. Figure 8a illustrates a poor installation with an elbow directly at the fan discharge. This type of installation can be avoided by selecting a fan with the correct rotation and

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discharge position as shown in figure 8b. Figure 9 illustrates another poor installation with an abrupt discharge into a plenum. A system effect results if a given length of discharge duct is not present. (See figure 12 on page 4 for recommended discharge ductwork.)

Both figures 10 and 11 illustrate installation with improper inlet conditions. Figure 10 could be improved with at least one fan wheel diameter of straight duct between the fan and the elbow. Installation as shown in figure 11 should be avoided if possible because the effect of inlet spin is difficult to define and correct.

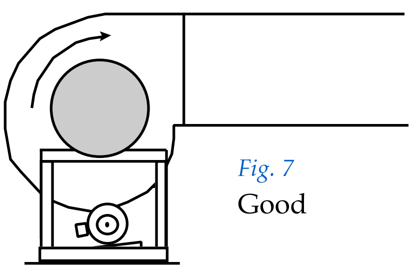

Fig. 7
Good

Figure 7 is typical of how centrifugal fans are tested. AMCA refers to this set up as "Type B: free inlet, ducted outlet."

Fig. 8a
Poor

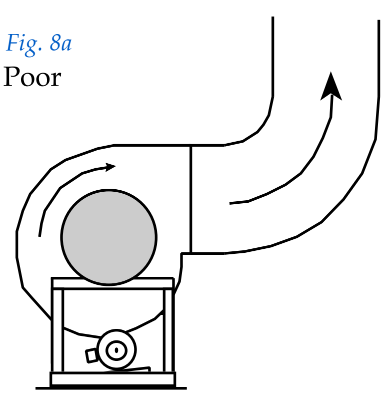

Figure 8a illustrates a poor installation with an elbow directly at the fan discharge.

Fig. 8b
Good

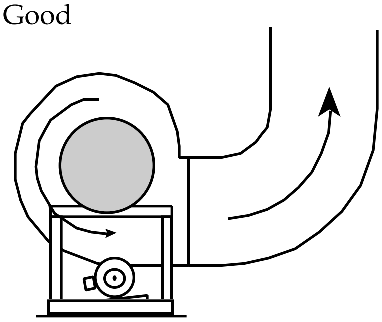

Figure 8b illustrates a typical installation with an elbow directly at the fan discharge. Discharge and rotation have been selected to match the fans field conditions of figure 8a.

Fig. 9
Poor

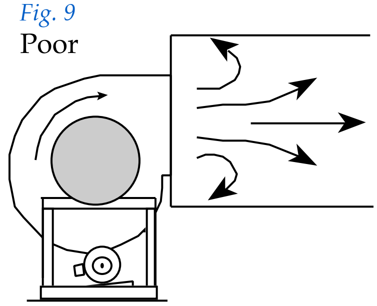

Figure 9 illustrates a poor installation with an abrupt discharge into a plenum.

Fig. 10
Poor

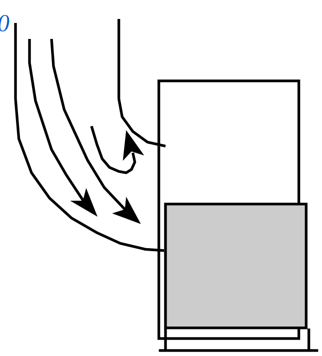

Figure 10 illustrates a poor installation with an elbow directly at the fan inlet.

Fig. 11
Poor

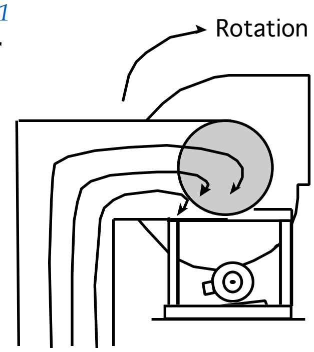

Figure 11 illustrates a poor installation where the duct design is causing inlet spin, resulting in reduced fan performance.

The previous illustrations show only a few of the many installation possibilities that can cause system effect. Remember different types of fans are subject to different considerations based on how they were tested. AMCA Publication 210 shows four basic installation types. However, combining all the fan types, fan arrangements, and manufacturer's choice of how to test, the installation possibilities are far too numerous to cover in this article.

What type of fans are affected by what condition and what condition presents the most common problems?

- Roof exhaust fans are affected by the inlet condition.
- Roof supply fans are affected by outlet conditions.
- Fan types typically affected by both inlet and outlet conditions are inline fans (both axial and centrifugal) and housed single inlet centrifugal fans.

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Fan installations with system effects should be avoided if possible. However, in many cases space constraints or other factors prohibit designers to allow for ideal conditions.

The following conditions (listed by inlet and outlet) cover the most common causes of system effect.

Inlet Conditions:

- elbows too close to fan inlet
- abrupt duct transition
- inlet spin due to duct design
- dampers not fully open
- damper locations
- poorly designed guards
- inlet too close to walls or bulkhead
- inlet boxes

Outlet conditions

- elbows too close to fan outlet
- abrupt transitions
- free discharge
- damper location
- weatherhoods
- discharge guards
- discharge too close to wall or bulkhead

The following recommendations will help in avoiding installation problems:

• Understand how the fan you selected was tested. (Refer to the catalog statements under the fans manufacturer's performance tables.)

• For roof mounted fans where the duct must run horizontal directly under the roof, install turning vanes in the elbow, plus consider using higher curbs or extended bases. This additional height will increase the distance from duct elbows and dampers in relation to the fan inlet.

• Consider different types of fans. For instance, if the duct must turn 90°, a centrifugal fan installed in the turn could be a better choice than an inline fan with an abrupt duct elbow.

• Select housed centrifugal fans with the proper rotation and best discharge position for the situation.

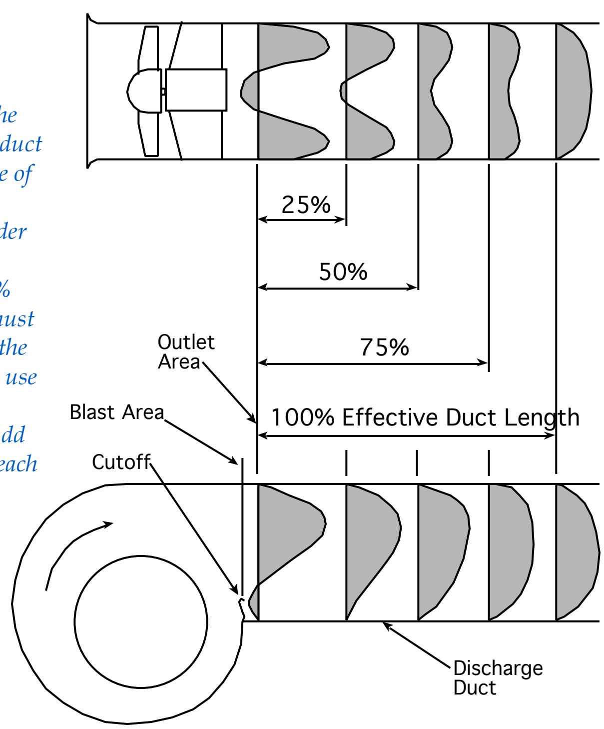
• If an elbow is required at a fan inlet due to space constraints, use an inlet box that has a predicted loss in lieu of the elbow.

• Avoid free discharges for inline and centrifugal fans. Add the duct length required to obtain a uniform velocity profile and to minimize losses (see figure 12).

It's understandable that in many cases an installation will end up having obstruction at the inlet or outlet (or both) causing system effect. If these situations cannot be avoided at the design stage, the system effect should be estimated and added to the calculated system resistance. Keep in mind that the standard procedures for the design of duct systems are all based on the assumptions of uniform flow profiles in the system. The standard adds for resistance of elbows does not account for the loss when the elbow is close to the fan. AMCA has recognized this problem and has published guidelines on how to

Fig. 12

Figure 12 illustrates the need to use a straight duct length on the discharge of both inline and centrifugal fans. In order to achieve a uniform velocity profile, a 100% effective duct length must be used. To calculate the 100% effective length, use 2.5 duct diameters for 2500 FPM (or less). Add one duct diameter for each additional 1000 FPM.



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compensate for system effects in its Publication 201, "Fans and Systems". The designers who fully understand system effects, and design to avoid them, must also follow-up with the installers to make sure the installation is as planned.

What are the penalties of system effect? Even when we recognize causes of system effect and we compensate for their losses, penalties result. The penalty starts with fans selected at higher speeds to compensate for additional losses. Higher speeds result in larger motors, increased cost, reduced efficiencies, increased vibration, and acoustical effects. Acoustical effects are usually completely overlooked even though the acoustic system effect penalty might be quite severe. The severity depends on how inadequate the fan to system connection is. In any case, you cannot expect the fan sound ratings to be as cataloged if system effect exists.

Trouble shooting existing installations when the system is short of air and pressure due to overlooked system effects or poor installation practices can be quite interesting. In most of these cases, it's very difficult to take accurate performance readings because of obstruction and turbulence at the fan inlet or outlet.

And now the finger pointing starts. Is it the fan or is it a system problem? So the big question arises, is the fan installed exactly as tested? In most cases, a visual inspection of the installation will lead you to a clear answer.

To solve deficient fan system performance problems, it helps to have a clear understanding of fan and system curves plus a knowledge of how to apply the fan laws.

Figure 13 uses fan and system curves to illustrate the original design point, the deficient performance reading, and the new fan and system curve with system effect.

Point 1 illustrates the original design point.

Point 2 is the design volume on the corrected system curve.

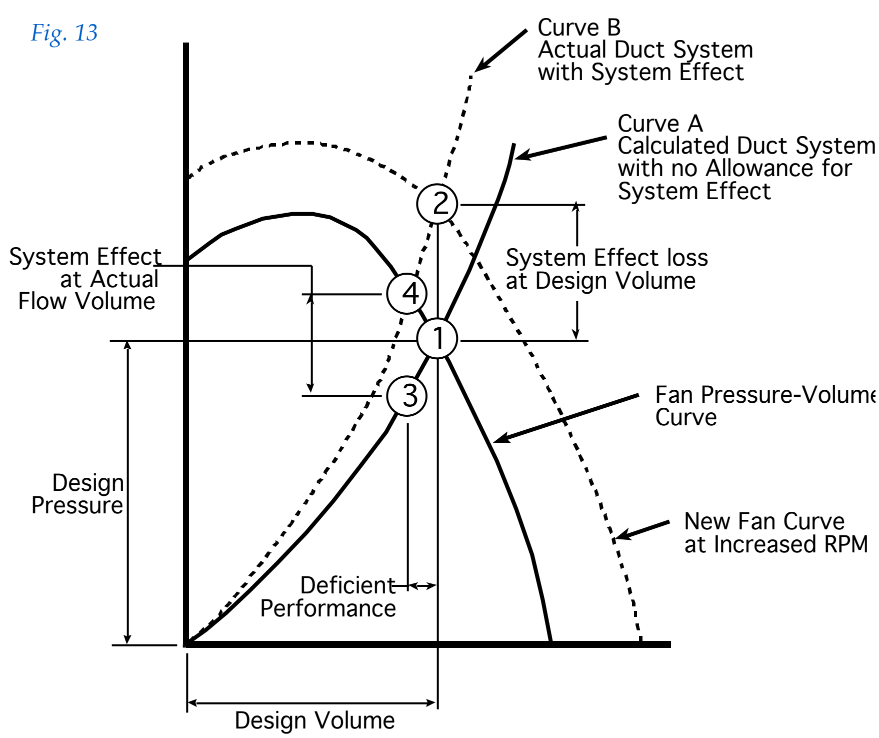
Point 3 is where the deficient volume falls on the original system curve.

Point 4 is where the deficient volume falls on the corrected system curve.

(All of the above is based on the assumption that the air density and the fan's speed are as designed)

To further explain, let's consider an example where the system is delivering 20% less air than design (point 1). The deficient volume is point 3 as shown on the original system curve. The original curve calculation did not include allowance for system effect. The difference in point 3 to point 4 illustrates the system effect at actual flow volume. The difference from point 1 to point 2 illustrates the

Fig. 13



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system effect at the desired volume (design volume). Because system effect is velocity related, the difference between points 1 and 2 is greater than the difference between points 3 and 4.

Points 2 and 4 fall on a new system curve. In order for the existing fan to produce the design volume (point 2) on the new system curve the fan speed must be increased.

Here's where the fan laws come into play. If 20% more air is required, the fan speed will need to be increased 20%. The resulting static pressure will be 44% higher and the BHP will be 73% higher than the original values. A new fan curve is developed as shown in figure 13. The problem may not be over at this point. New fan drives and a larger motor may be required. Hopefully, the original fan can handle these new conditions. Direct drive fans present even a greater problem than belt drive fans. In some cases, it's not possible to use the existing fan unless the installation can be modified to eliminate or reduce the causes of system effect.

Summary

In summary, here are a few points to consider:

• At the design stage, don't try to save a few dollars per square foot of space. The cost of the resulting poor installation could be much greater.

• Carefully design the system so it can operate as intended. Personnel doing the installation and checkout should also be familiar with causes of system effect.

• In correcting installations with system effect, changing the ductwork should not be the last consideration. Remember the penalties of system effect will remain for the life of the project.

From our readers.....

Fan law equations

A number of our readers advised us of an error in our December article, "The basics of fan performance tables, fan curves, system resistance curves and fan laws".

The correct formula for the static pressure is:

$$CFM_2 = \frac{RPM_2}{RPM_1} \times CFM_1$$

$$SP_2 = \left(\frac{RPM_2}{RPM_1}\right)^2 \times SP_1$$

$$BHP_2 = \left(\frac{RPM_2}{RPM_1}\right)^3 \times BHP_1$$

Subscript 1: Describes the existing conditions

Subscript 2: Describes the new conditions

CAPS selection question:

In the DOS version of CAPS you were able to specify if a selection criteria of most efficient or smallest size. Can you still do this in the windows version of CAPS?

Answer:

The windows version of CAPS displays all stable selections where the old DOS version only displayed a limited number of selections.

You can sort the fans by the smallest size, most efficient, or quietest, by clicking on the column heading where the selections are displayed. For example, click a Model and the selections will be displayed smallest fan first (this is the way they originally appear). Click on the Operating Power and the display will be resorted in the order of lowest to highest power. If you click on sones, they will be resorted lowest to highest sone level.

All the selection headings work this way, allowing you to quickly sort and look at the fans that meet your particular selection criteria.

Thank you!

We appreciate your input. Please continue to send in your comments and suggestions.

Fire smoke damper control options

Fire smoke dampers are available with a number of control options designed to meet various code and system requirements. Typical options include one or more of the following:

- Blade position indicator
- Heat responsive link
- Temperature rated electric sensor and
- Dual temperature rated electrical sensor.

This article is a summary of the unique function of each device including the relative advantages, potential disadvantages, and recommendations.

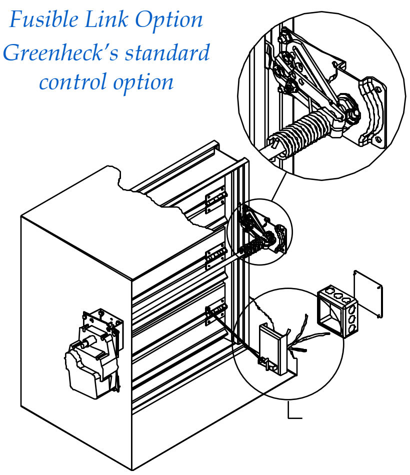
Blade Position Indicator

Blade position indicators include two micro-switches mechanically linked to the damper blade or damper control shaft. One of these switches closes when the damper blades are open (providing indication of an open damper) and the other switch closes when the damper blades are closed (providing indication of a closed damper). The open and closed blade position is typically indicated by means of lights on a control panel located within the building. Technically, the blade position indicator option is not a control option, but is discussed here because it often is combined with, or provided in addition to, one of the following control options. *Greenheck's model is the OCI.*

Heat Responsive Link

With the heat responsive link option, the fire smoke damper's actuator is coupled to the damper through a mechanism that incorporates a spring and fusible link. When air temperature at the damper rises to the temperature rating of the heat responsive link (usually 165°F, but may be 212°F, 286°F or 350°F), the link fuses (or melts) and separates allowing the spring to close the damper and positively latch it in the closed position. Melting of the heat responsive link mechanically disconnects the damper from its actuator. *Greenheck's model is the Fusible Link (standard).*

Fusible Link Option
Greenheck's standard control option



Advantages

1. Use of a heat responsive link designed to melt at a specific temperature is a very simple and reliable method of triggering emergency damper closure. There is very little that can go wrong with a properly installed mechanical heat responsive link option, which makes it the most reliable of available methods to provide emergency damper closure.

Potential Disadvantages

1. When the heat responsive link melts, immediate spring action causes the damper to close very quickly. If this type of closure occurs when high velocity air is flowing through the duct system, there is potential for duct damage due to the dynamics of rapidly interrupting the airflow and the resultant pressure changes. If necessary, appropriately applied duct pressure relief techniques can prevent such damage.

2. When the damper has closed because of heat responsive link operation, the link must be replaced to reopen the damper.

3. In the past, many jurisdictions required the minimum closure temperature of fire rated dampers to be 165°F. As such, many systems are still specified to have a 165°F closure temperature even though the 1996 Edition of NFPA 90A allows primary closure temperatures up to 350°F.

UL 33, Standard for Heat Responsive Links for Fire-Protection Service, describes the maximum ambient temperature that a 165°F rated heat responsive link can be subjected to at the point of installation as 100°F. The potential negative of selecting a 165°F link is that the

Fire smoke damper control options, continued from page 7

link can be exposed to temperatures above 100°F in system start up, shipping, and/or storage and this may cause the links to operate (separate) prior to being called on in a fire emergency. This problem can be eliminated by specifying 212°F links since their maximum ambient temperature at the point of installation, per UL 33, is 150°F.

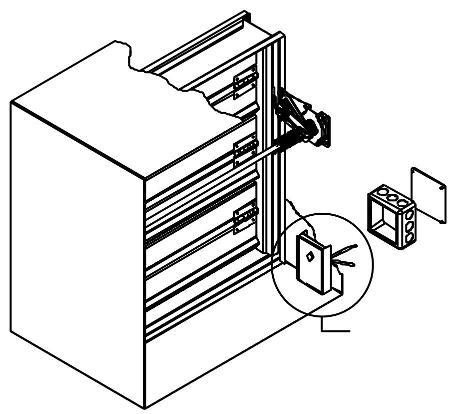
Temperature Rated Electric Sensor

With the temperature rated electric sensor option, an electric thermostat instead of a heat responsive link senses temperature at the fire smoke damper. The thermostat has a fixed set-point (usually 165°F or 212°F, but in accordance with NFPA90A can be any temperature from 160°F to 350°F). When the temperature at the damper reaches this fixed setting, power to the damper's actuator is interrupted and the damper is closed by the actuator's spring return feature. After elevated temperature activates the thermostat (causing emergency damper closure), the temperature must fall below the fixed set-point and someone must push the thermostat's manual reset button at the damper before the fire smoke damper can return to normal operation. *Greenheck's model is the RRL.*

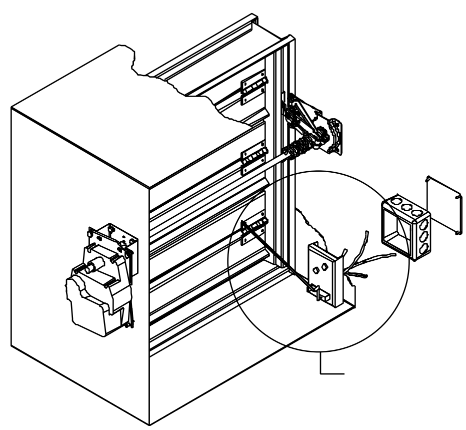
Advantages

1. Damper closure by the fire smoke damper's actuator is usually slower than the action produced by the fusible link

Temperature Rated Electric Sensor
Greenheck's model RRL



Dual Temperature Rated Electric Sensor
Greenheck's model TOR



option. This reduces the potential for duct damage due to emergency or accidental damper closure.

2. Resetting of the damper's thermostat only requires pushing an easily accessible reset button (located at the damper).

Potential Disadvantages

1. The possibility exists that exposure to some higher temperature for some period of time (as during a fire emergency) would damage the thermostat and make it unreliable for continued use. If a thermostat that had been damaged in this manner was manually reset for continued operation, the fire smoke damper may not properly perform its function during the next fire emergency.

Dual Temperature Rated Electric Sensor

This control option is similar to the temperature rated electric sensor option but it incorporates two fixed set-point (manually resettable) electric thermostats. The primary (lower temperature) thermostat's set-point is usually 165°F or 212°F and the secondary, (higher temperature) thermostat's set-point is usually the damper's UL 555S degradation temperature; typically 250°F or 350°F. This option was designed to meet the requirements of NFPA 90A prior to 1996 which designated the maximum primary closure temperature as 286°F.

During the 1980s, the dual temperature rated electric sensor option was developed to allow the fire smoke damper to close at a primary temperature

Fire smoke damper control options, continued from page 8

below 286°F, per NFPA 90A, but then be reopened (by electrically bypassing the primary thermostat) to allow use of the duct system up to a higher temperature, like 350°F, for smoke control purposes. However, the 1996 version of NFPA 90A changed the 286°F limit to the rating of the UL 555S Smoke Control Rating, or a maximum of 350°F. This allows the damper to remain operational for smoke control purposes up to the system's smoke control design temperature. It also eliminates the need for the complex and costly dual temperature rated electric sensor option. Now, the lower cost 350°F heat responsive link, or 350°F temperature rated electric sensor, can be utilized.

The dual temperature rated electric sensor option is still available and continues to be specified primarily due to an unawareness of the temperature change in the 1996 version of NFPA 90A. *Greenheck's model is the TOR.*

Advantages

1. An easy way to comply with building codes that require all fire dampers to close at 165°F or 212°F but still provide the flexibility to override this 165°F damper closure so the duct system can be utilized for smoke control functions.

2. Eliminates the need to obtain approval from the building code authority having jurisdiction when designing and installing a smoke control system that requires fire dampers to remain open until duct temperatures reach 350°F.

Potential Disadvantages

1. Requires much additional electrical wiring with its associated higher cost to provide reopenable capabilities at each installed damper location.

2. Requires a relatively complex system of controls requiring ample training to operate. It is likely that the additional expense of this control option does not result in significant return because the extreme complexity of the resulting system negates its appropriate use.

3. Dual temperature rated electric sensor options have been heavily oversold leaving the mistaken understanding with some engineers, designers, and specifiers that this dual thermostat option is a requirement to meet UL, NFPA, or building code requirements. Consequently, this fire smoke damper control option is often specified even though the building control system design does not provide the override circuitry required to utilize its features.

Recommendations

As the purpose of all fire smoke damper control options is to operate the fire smoke damper appropriately during a building fire emergency, Greenheck recommends selecting the simplest and most straight forward control option that will do the required job.

Dampers in an engineered system

Greenheck recommends using the 350°F rated heat responsive link option, or as a second alternative, the temperature rated electric sensor option with a thermostat set-point of 350°F. The blade position indicator option is recommended if the system provides pilot lights, a status board, or some form of computer interface to make use of the damper open/closed position information. Use of the dual temperature rated electric sensor option is not recommended as it adds considerable cost and complexity and is no longer required for compliance with NFPA 90A.

Dampers not in an engineered system

Greenheck recommends using the 212°F rated heat responsive link option, or as a second alternative, the temperature rated electric sensor option with a thermostat set-point of 212°F. The blade position indicator option is recommended if the system provides pilot lights, a status board, or some form of computer interface to make use of the damper open/closed position information.

Understanding Fan Bearings & Bearing Life

The following information on fan bearings and bearing life is intended to take the mystery out of the many terms the air moving industry uses when discussing the qualities of bearings for various fan applications and their life expectancy.

There is little question when it comes to fan reliability, the fan shaft bearings (for belt drive fans) are the single most important fan component. Also, there is no one single fan manufacturer that has more experience in bearing application than Greenheck. The reason for our broad application knowledge lies in the fact that Greenheck has the most comprehensive product line ranging from small light duty commercial fans to heavy duty, high speed industrial fans. In addition, we have been manufacturing many of these products in excess of fifty years.

Through the years we have worked closely with our bearing supplier partners to establish the quality features required for our full range of products, and to provide the bearing life expectancy required by our industry.

Bearing Life

Bearing life is usually expressed as the number of hours an individual bearing will operate before the first evidence of metal fatigue develops in the rings or rolling elements. In past years, four different terms were used when referring to bearing life. The terms commonly used were B_{10} or L_{10} and B_{50} or L_{50}. The terms B_{10} and L_{10} had the same meaning and the terms B_{50} and L_{50} also had the same meaning. It's time to end the confusion! In today's terminology the preferred term is L_{10}. However, L_{50} is sometimes used, therefore both meanings must be understood.

Air handling quality ball bearings (left) and roller bearings (right)

L_{10} life -

The preferred term in specifying bearing life.

The American Bearing Manufacturers Association (ABMA), formerly the AFBMA defines the Basic Rating Life, L_{10} as the bearing life associated with a 90% reliability when operating under conventional conditions, i.e. after a stated amount of time 90% of a group of identical bearings will not yet have developed metal fatigue. L_{10} life is also referred to by some manufacturers as the "minimum expected life".

L_{50} life -

Or average life.

Although the L_{10} life is the proper method of specifying fatigue life per the ABMA, another term is often used in the industry. The L_{50} or average life is accepted as the bearing life associated with a 50% reliability, i.e., after a stated amount of time, only 50% of a group of identical bearings will not yet have developed metal fatigue. L_{50} life equals five times the L_{10} life.

In other words, to get a L_{50} life equal to a L_{10} 80,000-hour life, you must specify the L_{50} life to be 400,000 hours. The following chart shows a comparison of L_{10} to L_{50} equivalents.

Writing the Bearing Specification

Basic Rating Life, L_{10} is a useful tool when specifying a given level of bearing construction. When required to provide a given L_{10} life, all equipment manufacturers must supply the

Required L_{10} Life Hours	Equivalent L_{50} (avg) Life Hours
20,000	100,000
40,000	200,000
80,000	400,000
100,000	500,000
200,000	1,000,000

Understanding Fan Bearings & Bearing Life, continued from page 10

same capacity bearing for a given RPM and shaft diameter. Also, an 80,000 hour L_{10} bearing will have a theoretical life twice as long as a 40,000 hour L_{10} bearing and hence will last longer in the field.

Here's Greenheck's recommendation for a typical bearing specification:

"Bearings shall be air handling quality, heavy duty grease lubricated, ball or roller type. Bearings shall be selected for a Basic Rating Life, (L_{10}) of 80,000 hours at maximum operating speed and horsepower for each construction level." (Air handling quality means the bearings meet the requirements for use in air handling applications; high speeds, long life and quiet operation. All bearings are 100% tested for excessive noise levels and bore dimensions are verified to be within tolerances.)

Note: If all the fan products you are specifying are from the Greenheck Fan & Vent catalog, you can specify L_{10} 100,000 hour life bearings at no extra charge.

The chart below provides another way to look at the expected bearing life. Assuming you specified Greenheck's standard bearing life of L_{10} 80,000 hours and your fans run an average of 8 hours per day, you can expect 27.5 years of life on 90 percent of the bearings.

In most cases, the Basic Rating Life will be much greater than shown because the bearings are selected for the maximum RPM and horsepower for each size and fan class. Most fans are selected significantly below their maximum fan rpm.

*Bearings with a L_{10} 200,000 hour life are optional for most centrifugal and vane axial products. However, in most cases, it is not practical to specify L_{10} 200,000 because of the associated cost. (And, do you really need the bearings to last for 68 years?) It is more practical for your customer to spend the additional money on maintenance.

Avoid writing bearings specifications without having the correct Basic Rating Life (L_{10} or L_{50}) terms in front of the required hours of life. If your specification reads 200,000 hour bearing life, your chance of getting what you want is minimal. Some suppliers will assume L_{50} 200,000 life is all that's required and you will end up with an inferior bearing system. Other suppliers will assume you are specifying the optional L_{10} 200,000 hour life and add unnecessary cost. The best suppliers will ask for a confirmation of the L10 life required.

L_{10} Life	Average running hours per day		
	8	16	24
80,000	27.5 yrs	13.7 yrs	9.2 yrs
100,000	34.3 yrs	17.2 yrs	11.4 yrs
200,000*	68.6 yrs	34.3 yrs	22.8 yrs

*Years of bearing life for fans running seven days a week, fifty two weeks per year.

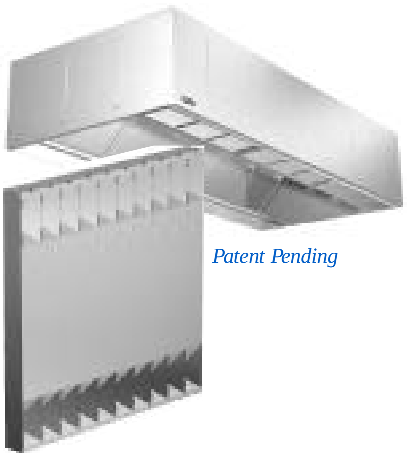
No Guarantee

Bearing Basic Rating Life is theoretical and is based on a collection of statistics. Specifying a L_{10} life does not guarantee that the fan bearings will have a 90% reliability when installed on a fan in the real world. The calculation for Basic Rating Life assumes proper lubrication is provided, no shock or vibration exists, alignment is virtually perfect, no debris enters the bearings and ambient temperatures are not extreme. In the real world, none of these conditions are realistic and the "installed life" of the bearing will depend on the application and maintenance.

To get as close as possible to the specified life, the installer and end user must follow the recommendations in the manufacturers installation and maintenance instructions.

What's new at Greenheck

The GX Series....a totally new concept in grease extraction



Patent Pending

Hood features

The GX Series hood is UL Listed and constructed of 18 gauge type 304 stainless steel with continuously welded external seams in accordance with NFPA 96. Supply air options include front face discharge, air curtain,combination face and air curtain, and short circuit.

Fire Suppression Systems

You may select an Ansul R102® Wet Chemical fire suppression system, water spray fire protection system, or the Ansul Piranha™ Wet Agent fire suppression system.

Greenheck's new GX Series Grease-X-Tractor™ high efficiency filter hoods offer the highest level of mechanical grease extraction available. This revolutionary new hood line includes Greenheck's new proprietary filter design...the most durable, best performing filter in the market .

Performance

The Grease-X-Tractor high efficiency filter hoods extract up to twice as much grease as ordinary baffle filters, and one and a half times more grease than water wash or dry cartridge hoods. These results have been confirmed by independent third party testing, actual case studies, and are based on "real life" commercial kitchen exhaust. The Grease-X-Tractor high efficiency filter is UL 1046 Classified, and NSF certified.

Filter design

This unique filter design is constructed of vertical chambers. Air enters at the top and bottom front of the filter creating a helical or "cyclone" airflow. Centrifugal impingement technology strips the grease and impurities from the airstream, without excessive static pressure. The static pressure requirements are only slightly higher than ordinary baffle filters, and much lower than dry cartridge or water wash extractors.

Benefits

The GX Series offers you and your customers tremendous value. The Grease-X-Tractor filter removes more grease from the air, resulting in less grease in the duct work and less grease accumulated on the roof. This equates to a more reliable ventilation system and lower maintenance costs. And remember, Greenheck is the only manufacturer of a complete kitchen ventilation system.

Greenheck
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