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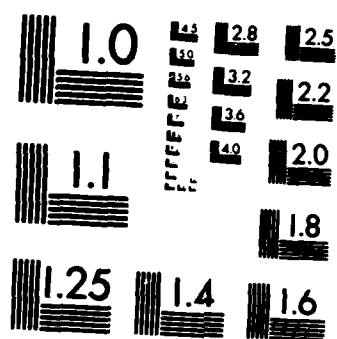
MODIFICATION OF CABINET FANS WITH INLET AIR GUIDE
FAIRINGS TO IMPROVE PERFORMANCE(U) CONSTRUCTION
ENGINEERING RESEARCH LAB (ARMY) CHAMPAIGN IL W H DOLAN
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TECHNICAL REPORT E-181
April 1983
(Energy Conservative Operation of
Existing Buildings and Facilities)

MODIFICATION OF CABINET FANS WITH INLET
AIR GUIDE FAIRINGS TO IMPROVE PERFORMANCE

by
William H. Dolan

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number) Cabinet fans are commonly used for central station air-handlers in commercial size heating, ventilating, and air-conditioning (HVAC) systems. This report describes the conception, construction, and testing of a device to improve the efficiency of cabinet fans by improving the fan inlet conditions. By observing airflow within the cabinet, a		

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— region of streamline flow and the boundary of flow separation were outlined. Fan efficiency was improved by enhancing streamline flow and eliminating flow separation within the cabinet by installing a fiberglass fairing which guided inlet air.

A fan was tested with and without the fairing in place; an overall 20 percent improvement in fan efficiency was observed with the fairing in place.

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FOREWORD

This work was performed for the Assistant Chief of Engineers under Project 4A762781AT45, "Design, Construction and Operation and Maintenance Technology for Military Engineers"; Task B, "Installation Energy Conservation Strategy"; Work Unit 004, "Energy Conservative Operation of Existing Buildings and Facilities." Mr. Bernard Wasserman, DAEN-ZCF-U, was the Technical Monitor.

The work was performed by the Energy Systems (ES) Division of the U.S. Army Construction Engineering Research Laboratory (CERL). Mr. R. G. Donaghy is Chief of CERL-ES. Appreciation is expressed to Mr. Richard Rundus and Mr. Victor Storm of CERL for their contributions to this work.

COL Louis J. Circeo is Commander and Director of CERL, and Dr. L. R. Shaffer is Technical Director.

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MODIFICATION OF CABINET FANS WITH INLET AIR GUIDE FAIRINGS TO IMPROVE PERFORMANCE

1 INTRODUCTION

Background

The fans within the central station air handlers that help heat, ventilate, and cool Army buildings use substantial amounts of electrical energy. The Army, as well as the private sector, operates many air-handler fans that often consume more energy over a single heating or cooling season than all other HVAC components combined.

The fan manufacturing industry markets centrifugal fans as efficient as 80 percent; however, it is common for fans with central air handlers to operate at static efficiencies of 50 percent or less. Several factors contribute to this poor performance:

1. Fan operation outside the region of efficient performance.
2. Nonoptimal outlet conditions.
3. Nonuniform inlet flow.
4. Swirl at the fan inlet.

To find ways to reduce fan electrical consumption by improving fan efficiency, the U.S. Army Construction Engineering Research Laboratory (CERL) examined these causes of poor fan performance. CERL determined that although replacing existing fans and rebuilding central station air handlers would address the first two factors, these options were too costly and usually impossible considering the existing space in most Army buildings. However, improving the inlet conditions using a fan fairing (the third and fourth problem areas) could improve fan operating efficiency without requiring any physical modification of the fan cabinet or air handler. Presently, cabinet fan manufacturers do not market such devices.

Objective

The overall objective of this study was to (1) determine if inserting an inlet fairing within a fan cabinet could improve fan inlet conditions and fan efficiency and (2) develop design guidance and specifications for an inlet fairing.

This report describes the results of investigation (1), above.

Approach

1. Procure and install a cabinet fan with an appropriate drive and duct network and operate it over a typical range.

2. Observe the inlet airflow within the cabinet by using various flow visualization techniques.

3. Based on the inlet flow patterns observed in Step 2, fabricate a fairing to be positioned within the cabinet.

4. Procure and install instrumentation to generate fan performance curves and fan efficiency.

5. Operate the test fan with and without the fairing in place to determine the effective increase in fan efficiency.

6. If results from Step 5 are positive, determine and acquire fans of the most common type used in Army construction.

7. Develop a mathematical expression to describe the fairing shape and determine an acceptable means of manufacturing.

8. Substantiate the improvement in fan efficiency and fan performance on the fans acquired in Step 6 outfitted with fairings fabricated according to Step 7.

9. Develop design guidance and specifications for fan cabinet fairings.

This report describes Steps 1 through 5, above.

Mode of Technology Transfer

It is recommended that the results of this study be incorporated into an Engineering Technical Note on retrofitting cabinet fans with inlet air guide fairings.

2 DISCUSSION

General

Most fans used for HVAC applications are the centrifugal type (Figure 1). A typical built-up air handler for a multizone system has three sections:

1. A mixing box which accepts either of two airstreams, or mixtures of each (usually outdoor air and return air).

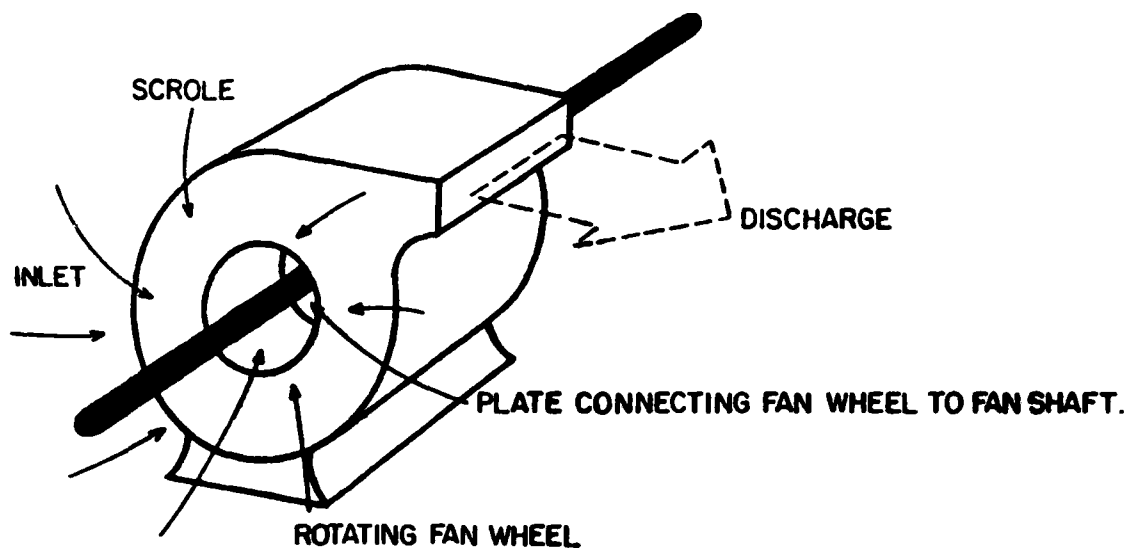


Figure 1. Double Width, Double Inlet (DWDI) centrifugal fan.

2. A fan drawing air from the mixing box.
3. Coils into which the fan discharges and over which air passes (heating and cooling).

To control the air moved by the air-handler fan, the fan is contained within a cabinet which, in the case of a multizone air handler, connects to the mixing box and coils.

Operating a centrifugal fan in this arrangement is far from ideal. Although this category of fan can operate at static efficiencies of 80 percent, optimum operating conditions are needed to produce this high efficiency.

The standard test conditions for testing centrifugal fans are:

1. A free, unobstructed inlet.
2. A long, slowly diverging duct section at the fan outlet ending with a uniform register which will not cause an air swirl (Figure 2). It should also be noted that the fan must be rotated within a certain speed range and the static pressure rise developed across the fan must also be within a certain range, depending on the characteristics of the fan (wheel diameter, blade type, scroll design).

Practical considerations of space and expense prohibit excessively large fan cabinets in air handlers. Thus, the inlet flow is not free and unobstructed and long diverging outlets which convert high-velocity discharge air efficiently to static pressure are not possible. In practice, operating a fan within an air-handler enclosure results in a substantial drop in fan efficiencies as compared with the nearly ideal conditions under which the fan is tested.

Literature from the Air Moving and Conditioning Association (AMCA) specifically states that three common causes of poor fan performances are (1) improper outlet conditions, (2) nonuniform inlet flow, and (3) swirl at the fan inlet.¹ Fans within built-up air handlers are subject to those three problems. The AMCA guides show the losses resulting from poor fan inlet conditions can be equivalent to a system static pressure drop of 0.25 in. water column (wc).^{*} For example, a system with a total static pressure drop of 2 in. wc would realize a reduction of up to 12 percent in the power imparted to the airstream and a similar savings in motor power if a fairing in the fan cabinet could reduce or eliminate this 0.25-in. wc loss.

¹ *Fans and Systems*, Publication 201 (Air Moving and Conditioning Association).

^{*} Metric conversion table is on p 15.

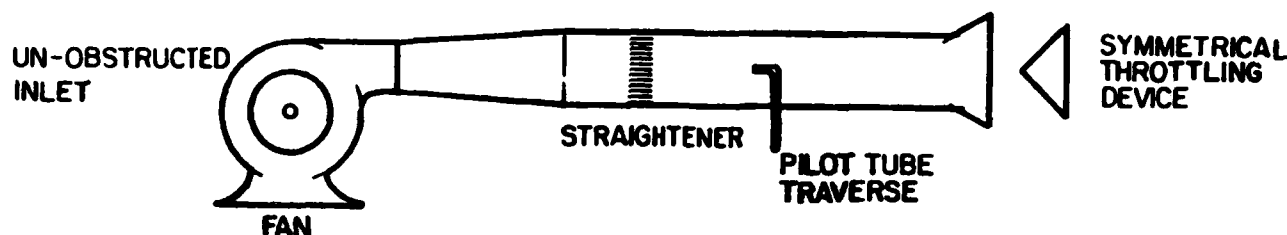


Figure 2. Standard configuration used for rating centrifugal fans.

The purpose of a fairing positioned within the fan cabinet is to direct the airstream entering the fan in a laminar manner. Without a fairing, airflow within a fan cabinet is laminar in a region around the center of the fan inlet and nonlaminar or separated in the corners and center of the cabinet, creating stagnant regions.

CERL reviewed several approaches to a fairing design. One approach was to analytically predict the flow contours which could occur without separation; basically, the inlet airstream has to accelerate around the fan scroll and turn 90 degrees to enter the fan.

Another approach was to study and map the flow patterns occurring in the fan enclosure without a fairing, and then base the fairing design on the observed shape of that part of the flow stream that did not separate. This latter approach was adopted because:

1. The cabinet fan used for this study had to be modified. Consequently, the rotative speed and flow rate associated with the range of best efficiency was not certain and so had to be determined experimentally. Analytical prediction of optimal fan performance for a modified fan plus flow patterns was considered to overcomplicate the study objective.

2. Flow separation is a relatively simple concept which can be observed using a variety of techniques.

Experimental Design

The equipment listed below was used to evaluate the merit of enhancing flow characteristics by inserting a contoured fairing in a cabinet fan:

1. A cabinet fan that could be fitted with a fairing.
2. Instrumentation to evaluate the fan performance.
3. An air duct system capable of adjusting the total system resistance.

4. A fairing.

A double width, double inlet (DWDI) backward incline-blade cabinet fan was available at CERL. Instrumentation was acquired to monitor the following parameters: fan speed, motor speed, motor torque, air temperature rise, static pressure rise, and total flow. CERL also designed and built an air duct system, including dampers. The fiberglass fairing, also designed by CERL, was fabricated on a foam mold.

Fan Modification and Ductwork System

Figure 3 shows the apparatus used for this experiment. The first section is the horizontal ducting fit to the fan inlet to keep the flow entering the fan one-directional and free from the swirl which sometimes results when a vertical duct section is fixed to the inlet of a horizontal fan.

The second section is the specially modified DWDI cabinet fan. Because one of the two fan inlets was obstructed by bearings, scroll supports, and frame members, the fan was capped, which resulted in a no-flow stall condition in one-half of the fan. (Test results proved this was not to be a problem; see Chapter 3.) The active half of the fan performed in a manner consistent with the fan laws.

The fan was driven by a 5-hp motor powered by a variable frequency power supply. Fan and motor speed were regulated by a potentiometer located near the motor. Torque developed by the fan motor was measured by a balancing-type scale (Figure 4).

The final section of the test facility was the outlet section which:

1. Spanned about 15 equivalent diameters after the fan discharge (intended to ensure a uniform outlet flow profile).

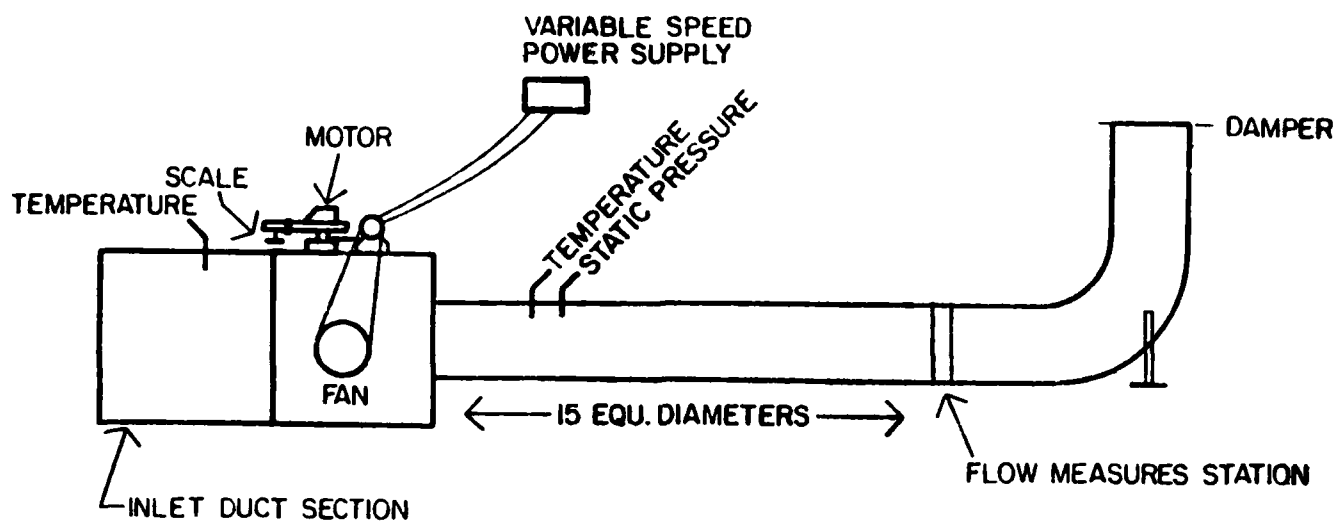


Figure 3. Experimental facility.

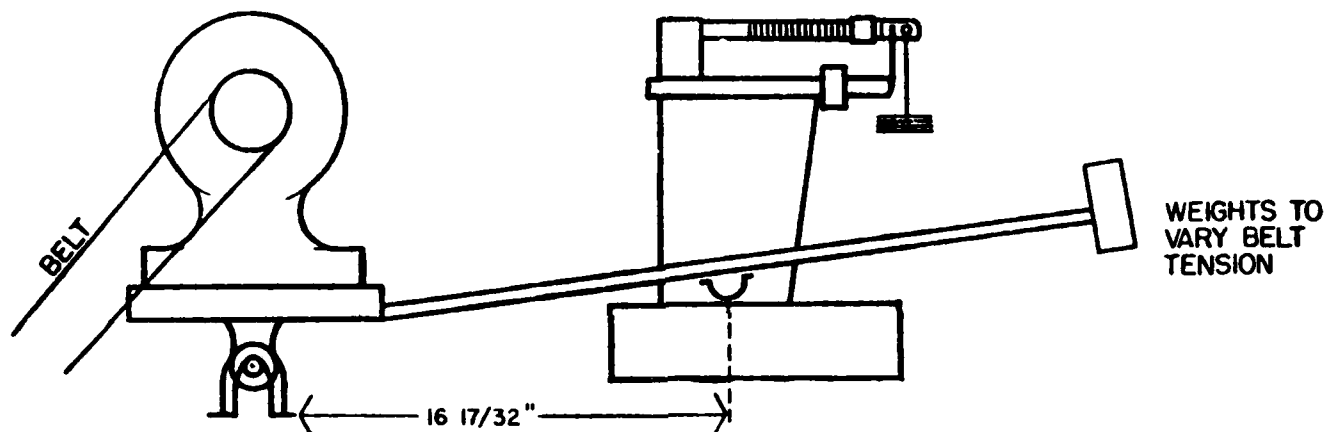


Figure 4. Torque measurement.

2. Incorporated a pitot tube rack to sample the air velocity at 24 points uniformly spaced across the duct cross section.

3. Discharged through two slide dampers located at the end of the ductwork. These dampers were used to control the airflow rate by varying the overall resistance of the duct section.

Flow Visualization

The methods considered for observing the airflow within the fan cabinet were:

1. Particulate tracing with laser anemometry.
2. Smoke generation by (a) fixed multiple-point sources, (b) a traversing point source, and (c) a single high-output source.

3. Yarn sampling by a single traversing probe.

Although laser anemometry is a sophisticated, noninterfering method, it was rejected because it is overly complex and expensive, and had a resolution greater than the needs of this study. Smoke generators, both single high-output source and single-traversing probe, successfully identified the overall characteristics of the airflow as well as the boundaries of streamline and turbulent flow.

The high-output smoke source was bottled carbon dioxide with a moderately convergent nozzle which produced condensate or clouds in the airstream. Pictures and notes were recorded of the streamline flow pattern observed through plexiglass windows on the fan cabinet.

The single traversing probe was made from 1/8-in. OD copper tubing about 3 ft long; it output smoke at the tip. The probe's small diameter had no appreciable effect on the flow and clearly identified the boundary of streamlined and separated flow.

The point-source traversing smoke generator, plus small pieces of yarn suspended from a thin probe, were used to map the shape of the unseparated streamline flow entering the fan.

Fairing Construction and Design

The fairing shape was based on the results of the flow visualization tests and elementary principles of low-velocity airflow; i.e., the duct design of smooth uniform sections and gradual turns to impede flow separation. Figure 5 is a two-dimensional representation of airflow within the cabinet. Layered airflow from the free stream into the fan inlet described an arc closely approximated by a circle with tangents at the left border of the fan cabinet and at the rear of the fan inlet.

The fairing's physical shape was defined by the arc on the converging section. The inlet was flared to form a rectangle the same size of the cabinet inlet (Figure 6).

The fairing was made of fiberglass formed over a mold. Fiberglass proved to be well suited to this application; it has excellent strength, flexibility which eased the fairing installation, and a smooth interior finish. The mold was carved from a block of rigid polyurethane foam, then painted with multiple coats of epoxy paint and waxed. The fiberglass was placed over the mold, allowed to harden, then separated from the mold with 120 psi air.

The fairing did not have to be fastened in place because it fit snugly into the cabinet. A hole for the

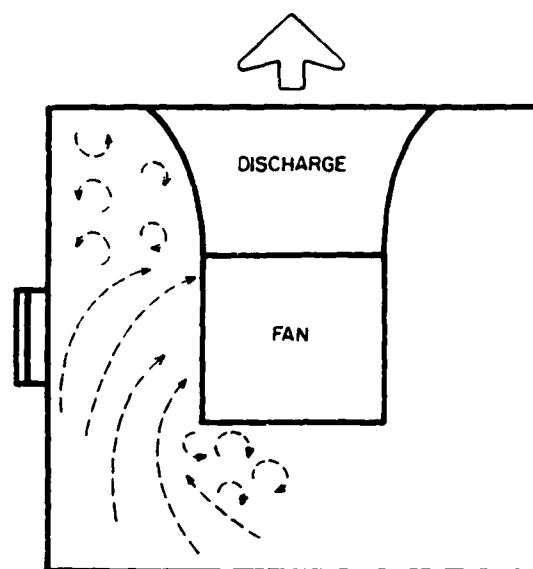


Figure 5. Flow within cabinet.

fan shaft was cut during assembly; a slice to the hole as well as the hole were cut during installation with a low-power reciprocating saw.

Static Pressure

Static pressure was sampled at the fan discharge using several static port pitot tubes. Static pressure at the fan inlet was assumed to be at atmospheric pressure because of the low velocities in the 5-ft section of ducting on the inlet. The static pressure was observed on a manometer with a finely graduated rule.

Temperature Measurement

It was of interest to measure the rise in temperature experienced by the airstream as it passed through the

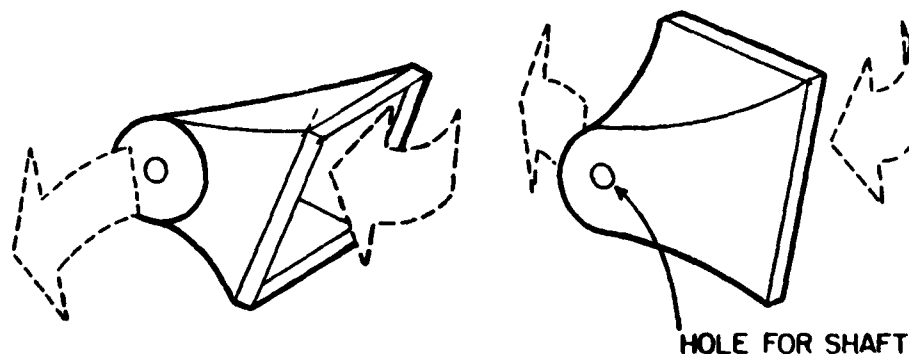


Figure 6. Fairing.

fan. This temperature rise can be used to calculate fan efficiency (temperature rise from ideal compression divided by actual temperature rise). Two temperature-measuring methods were tried:

1. Four-wire platinum probes were used along with a Hewlett-Packard 3052A data acquisition system. For the purpose of a differential temperature measurement, these probes displayed excellent accuracy. (The probes were matched within an adiabatic environment of 0.05°F agreement.) However, the fast response of these probes sensed minor fluctuations occurring in the inlet airstream, which meant minor temperature fluctuations had to be averaged out in the inlet airstream.

2. High-accuracy glass bulb thermometers (graduations to 0.18°F) were matched within an adiabatic environment to 0.09°F agreement. The relatively slow response of the glass bulb thermometers tended to average out the minor temperature variations observed in the inlet airstream by the four-wire platinum probes.

The glass bulb thermometers were selected for use for the remainder of the study.

Power/Torque Measurement

Fan shaft power was determined by measuring the rotative speeds of the fan and motor shafts and the torque developed by the motor. Although this method did not account for the belt drive losses, those losses were believed to be negligible. They also would be the same for each experimental condition.

Motor torque was measured by recording a force transmitted by a rigid member to a balance scale with a contact point 17.07 in. from the motor shaft center.

The rotative speeds of the fan and motor were measured using a digital photo tachometer accurate to within 1 rpm. Power was calculated as the product of torque and rotative speed.

Airflow Measurement

An airflow measuring station manufactured by Cambridge Filter Corporation was used to record flow. This device uses 36 pitot probes evenly positioned across a 1 x 1 ft cross section. Stagnation and static pressures are transmitted by averaging manifolds. The measuring station was located 15 equivalent diameters downstream of the fan discharge to ensure a uniform flow profile. The manometer supplied by the manufacturer was calibrated in units of flow intended specifically for the flow station.

Testing

Fan performance was recorded with and without the fairing in place by generating standard fan curves at various rotative speeds (static pressure vs flow for a fixed rotative speed). Air temperature rise across the fan and motor torque were also recorded. Thus, fan efficiency could be calculated by two means:

1. Power imparted to the air divided by motor shaft power.
2. Temperature rise associated with ideal compression divided by actual temperature rise.

Efficiency Measurement

If the process of the compression of air by the test fan is assumed to occur at constant density, a simple expression for fan efficiency as a function of pressure rise and temperature rise can be derived. For ideal gas compression, the density will increase according to:

$$\frac{P_1}{P_2} = \left(\frac{\rho_1}{\rho_2} \right)^k \quad [\text{Eq 1}]$$

where:

P = absolute pressure

ρ = density

k = gas constant.

However, the actual process of compression by a fan will involve additional heat added to the gas because of fan inefficiencies, which will cause a temperature rise and corresponding reduction in density according to Eq 2:

$$\frac{T_2}{T_1} = \frac{\rho_1}{\rho_2} \quad [\text{Eq 2}]$$

where T = absolute temperature.

Conveniently, the process occurring in the test fan having an approximate efficiency of 35 percent results in the effects of ideal compression and constant pressure heating on gas density negating one another. This is evident by examining the test fan at two operating points in the range of interest of this study:

1. A relatively high compression of 2.38 in. wc where a temperature rise of 2.36°F was observed.

2. A relatively low compression of 1 in. wc where a temperature rise of 1.62°F was observed.

Using ideal gas laws

$$\frac{\rho_1}{\rho_2} = \frac{P_1 T_2}{P_2 T_1} \quad [\text{Eq 3}]$$

and laboratory conditions of absolute pressure (29.21 in. Hg), the resultant change in $\frac{P_1}{P_2}$ for Case 1 was 0.999 and Case 2 was 1.000. Thus, over the operating range of interest of the test fan, the change in density of the air was negligible.

Continuing in the derivation of an expression for fan efficiency, Eq 4 defines the rate at which energy is delivered by the fan shaft and realized by a temperature rise in the airstream (i.e., change in enthalpy of the airstream):

$$P = (\text{cfm})(\rho)(c_p)(\Delta T) \quad [\text{Eq 4}]$$

where:

P = rate of shaft work

cfm = volumetric flow rate

ρ = density

ΔT = temperature rise

c_p = specific heat of the gas

Using a density for air of 0.071 lbm/cu ft (based on absolute pressure of 29.21 in. Hg, 75°F, and 65 percent relative humidity) and a specific heat of 0.024 Btu/lbm°F, Eq 4 simplifies to

$$P = 0.000402 \text{ cfm } \Delta T \quad [\text{Eq 5}]$$

where P = power (hp).

Eq 6 is a hydraulic definition for power imparted to a fluid as a function of volumetric flow rate and pressure rise.

$$P = 0.0001573 \text{ cfm } \Delta P \quad [\text{Eq 6}]$$

where: P = power (hp)

ΔP = pressure rise (in. wc)

An expression for an efficiency is formed by combining Eqs 5 and 6:

$$n, \text{ efficiency} = \frac{\text{power imparted the air}}{\text{shaft power}}$$

$$n = \frac{0.0001573 \text{ cfm } \Delta P}{0.000402 \text{ cfm } \Delta T}$$

$$n = 0.391 \frac{\Delta P}{\Delta T} \quad [\text{Eq 7}]$$

where:

ΔP = pressure rise (in. wc)

ΔT = temperature rise °F

To verify Eq 7, the case of an ideal fan having an efficiency of 1 was examined. A relationship of $\Delta T = 0.391 \Delta P$ should exist. From ideal gas laws, the relationship of pressure and temperature is as follows:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \quad [\text{Eq 8}]$$

Using the laboratory conditions of $T_1 = 535^\circ \text{R}^*$, $P_1 = 397$ in. wc, and k for air = 1.4, a change in P of 1 in. wc ($P_2 = 398$ in. wc) produces a change in temperature of 0.386°F, which is in excellent agreement with the expression stated by Eq 7 derived independent of ideal gas laws.

In conclusion, excellent confidence is held in the measurement of fan efficiency using the observed temperature and pressure rise and assuming a constant density process.

3 TEST RESULTS

1. Fan performance improved with the inlet air guide fairing in place. For a fixed rotative speed and fixed airflow rate, the fan developed an additional static pressure rise with the inclusion of the fairing. This indicates an increase in the fan capacity. More significantly, the fan efficiency improved and the region of efficient operation increased. This would

*°R = °F+460.

be reflected in reduced power requirements and operating costs.

2. The experimental test apparatus did not truly represent a typical DWDI centrifugal cabinet fan since one fan inlet was capped off. However, the test results show the positive effects of the inlet air guide fairing and it is expected a conventional DWDI fan would benefit to an even greater extent from the inclusion of two inlet guide fairings.

3. The testing determined that fan performance could be improved using a flow contouring insert within the fan cabinet. The test fan operated at a peak static efficiency of 38 percent in a standard configuration. With the inlet air guide fairing in place, the test fan operated at 43 percent peak static efficiency. This represents an improvement in peak efficiency of 13 percent. The average efficiency of the test fan improved by 20 percent with the fairing in place due to the increase in the region of efficient operation.

4. Testing involved operating the fan at three rotative speeds, through a range of no-flow to wide-open flow. An operating range of 25 to 75 percent of

wide-open flow was selected as operational test points. Since centrifugal fans are designed to move air against resistance, it is inappropriate to test fan performance near wide-open flow. Operating centrifugal fans in a range of less than 25 percent wide-open flow is not recommended because surging usually occurs. Table 1 summarizes the results of operating points over the flow range recorded for each of three rotative speeds. Although an absolute value of improved efficiency could not be quantified for this series of trials, 10 of the 11 trials resulted in an increase in fan capacity and efficiency with the fan retrofit with the inlet air guide fairing (Table 2). The average efficiency for the

Table 1
Summary of Fan Testing

Rotative Speed (rpm)	Flow Range (% of Wide-Open Flow)	Average Efficiency With Fairing/ Without Fairing	Peak Efficiency With Fairing/ Without Fairing
1500	30-73	41/31	43/38
1750	44-78	42/35	43/37
1990	29-69	37/35	42/38

Table 2
Fan Testing Data

Rotative Speed (rpm)	Flow (cfm)	Static Pressure With Fairing (in. wc)	Static Pressure Without Fairing (in. wc)	Static Efficiency With Fairing	Static Efficiency Without Fairing
1500	1200	2.00	1.94	43	28
1500	1500	1.75	1.75	42	38
1500	2300	1.47	1.50	40	36
1500	2900	1.09	.87	40	21
1750	2000	2.44	2.38	41	37
1750	2500	2.19	2.06	43	37
1750	3000	1.88	1.75	41	35
1750	3500	1.38	1.34	43	32
1990	1500	3.50	3.38	32	29
1990	2000	2.88	2.75	42	38
1990	3500	2.19	2.13	37	38
Average:				40.4%	33.6%

11 trials without the fairing was 33.6 percent; with the fairing in place the average efficiency was 40.4 percent.

5. For a system where energy to operate fans is half the total annual HVAC energy usage, a 20 percent reduction in fan power would result in an overall energy consumption reduction of 10 percent for the HVAC system, a substantial and favorable improvement.

6. Further refinement and evaluation of fairing inserts is necessary to more accurately quantify efficiency improvements and cost-benefit ratios.

4 CONCLUSION

The results of this study indicate that the efficiency of cabinet fans common to Army air-handler systems can be improved 20 percent overall by inserting a fiberglass fairing to guide inlet air.

METRIC CONVERSION CHART

$$1 \text{ in. wc} = 249 \text{ Pa}$$

$$1 \text{ in. Hg} = 3385 \text{ Pa}$$

$$1 \text{ hp} = 0.74 \text{ kW}$$

$$1 \text{ in.} = 25.4 \text{ mm}$$

$$1 \text{ ft} = 0.3048 \text{ m}$$

$$1 \text{ psi} = 6.894 \text{ kPa}$$

$$(^{\circ}\text{F}-32)0.55 = ^{\circ}\text{C}$$

$$1 \text{ rpm} = 0.104 \text{ rad/s}$$

$$1 \text{ lbm/cu ft} = 16 \text{ kg/m}^3$$

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