INLET INSTALLATION EFFECTS ON DIFFERENT TYPES OF FANS AND DUCTWORK DESIGNS

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SUMMARY

This paper presents an analysis of test results obtained by NEL and AMCA to quantify the fan system effect due to various duct obstructions at the inlet of axial or centrifugal fans of different types. A new definition of system effect factor based on the average relative drop of the flow rate on the fan curve is proposed, which appears more physical and easier to use than the SEF adopted in AMCA Publication 201.

INTRODUCTION

Fan system effect (or installation effect) refers to both the cause and the quantitative result of the difference often observed between the performance of a fan measured on site and that which is anticipated from the curve of this fan measured in a laboratory test configuration and the resistance curve of the system. The system effect may affect not only the aerodynamic performance of the fan but also its acoustic and vibration performances. This paper essentially deals with aerodynamic installation effects.

Aerodynamic installation effects have been the subject of numerous publications due to the severe possible consequences encountered if they are neglected. AMCA Publication 201 [1] provides an insight of the main causes of fan system effect and proposes a methodology to quantify the system effect factor that is recalled in section 2. This methodology is applied to assess the influence of components close to the fan inlet or outlet, such as 90° elbows, inlet boxes, volume control dampers, walls, cabinets, … on the fan performance of axial and centrifugal fans. Zaleski [2] presents additional results of a test program undertaken by AMCA on axial flow fans to determine system effect factors for mitred elbows fitted in close proximity to the fan inlet and outlet. Riera et al. [3] shows results of tests carried out at CETIAT to quantify the system effect of eleven duct configurations at the inlet of a forward curved centrifugal fan.

An extensive experimental programme has been conducted by NEL (National Engineering Laboratory) in order to quantify installation effects for various fans of different types connected to ductwork fittings at their inlet or outlet. A summary of the results of this work was published in a guide of the Fan Manufacturer's Association [4] and in an IMechE Seminar paper [5].

Several experimental research projects conducted by AMCA and financially supported by ASHRAE had as an objective to obtain a set of system effect data for various appurtenances at the inlet of fans of different types. The results of two of these projects ([6] and [7]) are analyzed below.

Greenzweig et al. [8] study the effect of an inlet flow distortion on the performance of a backward-curved centrifugal fan. They quantify the fan installation effect on the total pressure and total efficiency due to this distortion.

The objective of this paper is to analyze the test data obtained by NEL and AMCA mentioned above with a new definition of the system effect factor (SEF), based on the flowrate drop of the fan due to the inlet disturbance. This new definition of SEF is presented and justified in section 2, then applied to NEL test data in section 3 and AMCA results in section 4. An example of acoustic system effect based on a similar approach and deduced from AMCA sound test data is also presented in section 4.

DEFINITION OF THE SYSTEM EFFECT FACTOR

Previous definitions of System Effect Factor (SEF)

In AMCA Publication 201 [1] the SEF is represented by letters, each letter being associated to a system effect coefficient C defined by:

$$C = SE / p_{dyn}$$

where: SE system effect loss, i.e. difference in static pressure between points 1 and 2 or points 3 and 4 in ***Figure 1***, depending on the airflow that is chosen (design airflow or reduced airflow due to system effect) and p_{dyn} , dynamic pressure at the fan inlet (when the disturbance is at the inlet)

As stated in [7] ***Figure 1*** means that system effect is considered as a change to the system, i.e. the resistance of the system is increased due to the installation effect. This situation is certainly not far from truth when the disturbing component is at the fan discharge since in this case the pressure loss of the component (a bend for instance) is certainly higher that the loss calculated from the usual databases, such as IDELCIK or MILLER . This is due to the fact that the flow behind a fan is highly non uniform while it is considered as uniform at the entrance of the singularity in the guides dealing with pressure losses of ductwork components.

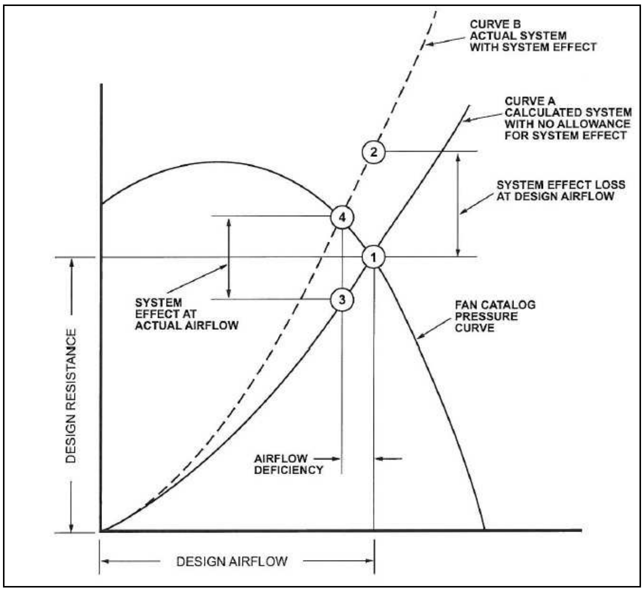

Figure 1: Definition of System Effect loss according to AMCA 201 (from [1])

Conversely, when the fitting is upstream of the fan its pressure loss calculated from the reference books is probably right but the flow disturbance induced by this component may lead to a deterioration of the fan performance curve as shown schematically in Figure 2. Without system effect, the volume flow of the fan in the system is Q_{v0} at the intersection of the fan and ductwork resistance curves. When an installation effect occurs, the fan curve deteriorates and the volume flow is reduced to Q_{v1}.

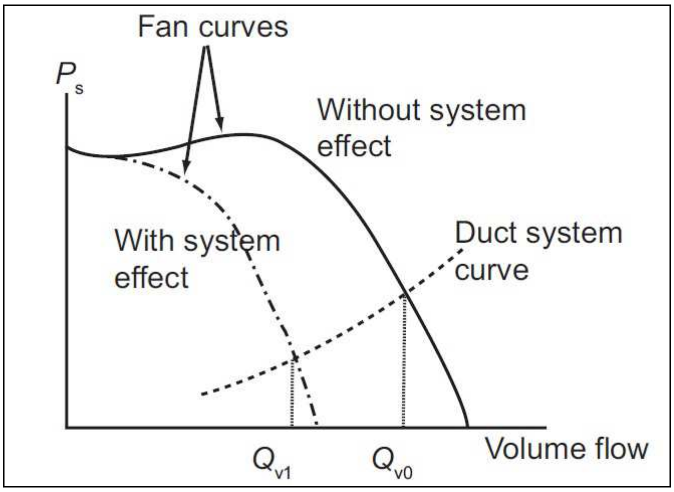

Figure 2: Influence of the system effect due to an inlet disturbance on the fan curve

The definition of the system effect factor taken by NEL in their research programme is explained in *Figure 3*, taken from [4].

The "fan only" curve is the performance curve of the fan alone tested on a standardised test rig. The "fan + fitting" curve is obtained from measurement of the performance of the fan with the fitting. The "fan + fitting (predicted)" curve is deduced from the "fan only" curve, from which the pressure loss of the fitting is subtracted at each flowrate. The "70% curve" is a parabolic system resistance line which intersects the "fan only" curve at flowrate Q1 = 0.7 Q0, where Q0 is the volume flow of the fan at zero total pressure. For the fans tested in this study Q1 is equal or close to their best efficiency point. This 70% resistance line intersects "fan + fitting" and "fan + fitting (predicted)" curves at flowrates Q3 and Q5 respectively.

If the fitting had no influence on the fan performance, then Q3 and Q5 would be identical. If Q3 and Q5 are different, like in *Figure 3*, then the fitting has induced an installation effect. NEL ranks the amplitude of the system effect according to the fan type and inlet fitting with the parameter A = 1-Q3/Q5. They consider the installation effect as insignificant if A is below 5%, significant when A is between 5 and 10% and large or excessive if A exceeds 10%.

In their paper Greenzweig et al. [8] analyze the effect of an inlet distortion on the performance curve of a centrifugal fan. The method to quantify the system effect due to this distortion is more or less similar to that adopted by NEL [4] except that it is based on the relative total pressure change between the "fan + fitting (predicted)" and "fan + fitting" curves..

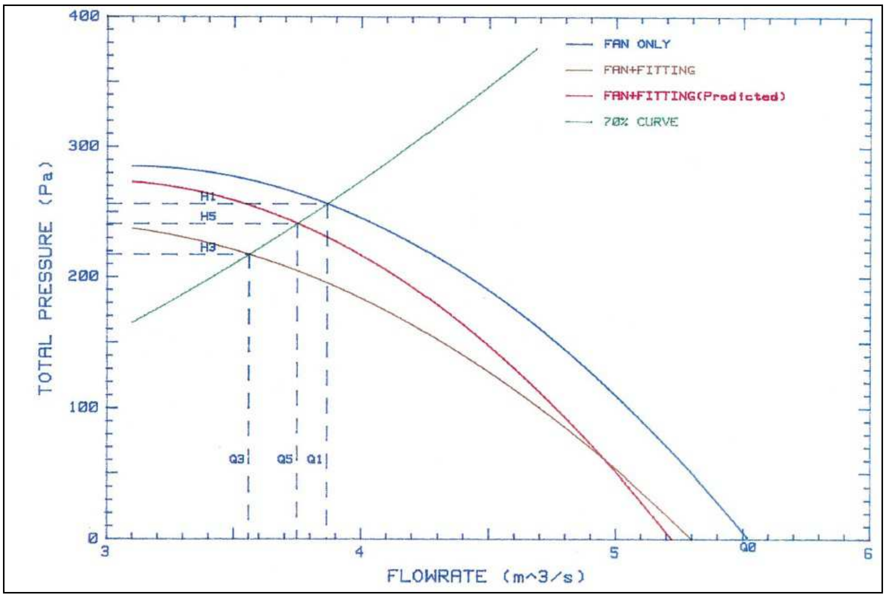

Figure 3: Analysis of test data by NEL (from [4])

With the notations of *Figure 3* the system effect factor based on total pressure is then written:

$$SEF = \frac{H5 - H3}{H5}$$

In [8] SEF is averaged on three parabolic system lines around the best efficiency point on the fan curve without system effect, which is the "fan +fitting (predicted)" curve in *Figure 3*

New definition of SEF

This new definition of system effect factor is similar to that adopted in the NEL work [4] but it looks more straightforward to use for quantifying fan system effect. It is based on the relative drop in flowrate due to installation effect when a fitting is connected to a fan inlet. With the notations of Figure 4 the relative flow decrease on a given system line is $\Delta Qv/Qv_0$. All the fan curves in this paper are based on static pressure instead of total pressure like in NEL study. The pressure loss of the inlet component is always added to the measured "fan + fitting" curve, so that the curve with system effect in Figure 4 is the "fan + fitting (predicted)" curve of Figure 3.

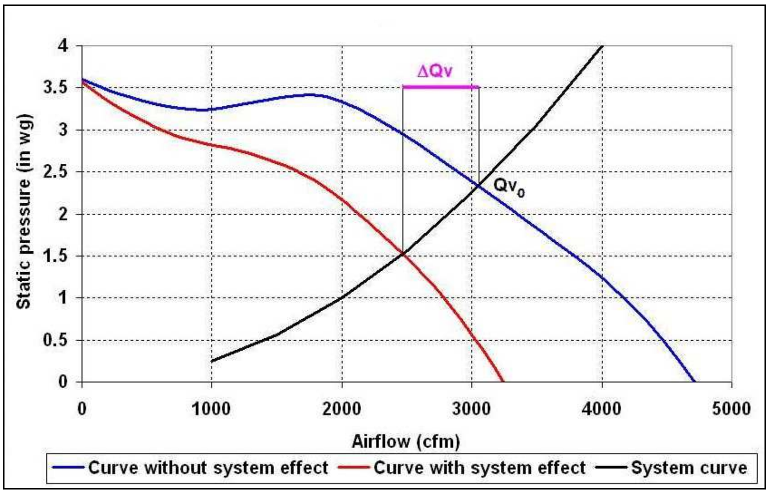

Figure 4: Definition of Qv_0 and ΔQv on a given system line

To assess the system effect on the whole fan curve the quantity $\Delta Qv/Qv_0$ is plotted as a function of ξ in Figure 5, where $\xi = Qv/\sqrt{Ps}$ is the system resistance coefficient and P_s the fan static pressure at flow rate Qv. To quantify the fan installation effect for each test setup $\Delta Qv/Qv_0$ is averaged over the ξ range, which provides the system effect factor that is presented in percent in the next sections for different fans and inlet fitting configurations.

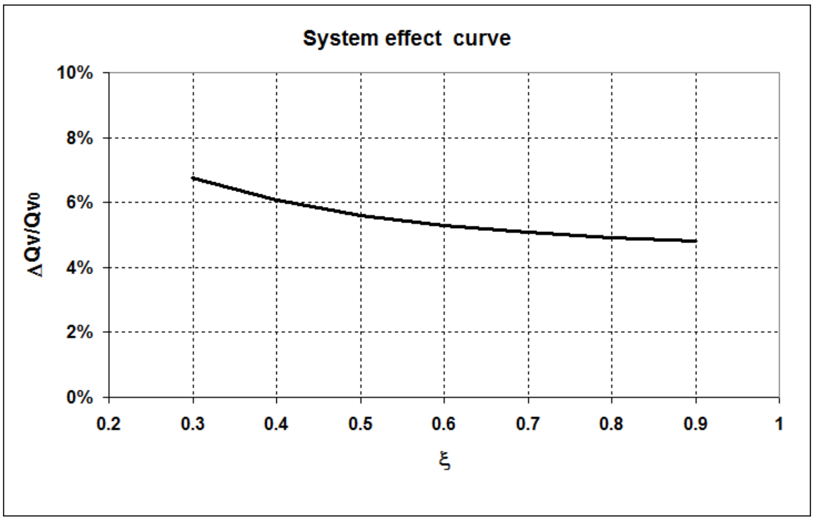

Figure 5: $\Delta Qv/Qv_0$ versus system resistance coefficient ξ

ANALYSIS OF NEL TEST DATA

Test setups

NEL performed an extensive experimental study on 11 fans of different types and 7 ductwork fittings located either at the inlet or the outlet of the fan [5]. With the inlet fittings only combinations of 9 fans and 6 fittings have been tested. The distance between the component and the fan inlet or outlet has been varied from 0D to 2D, where D is the duct diameter (D = 630 mm in the whole study). Figure 6, taken from [4], shows an example of test setup for the determination of the installation effect of a 90° bend directly fitted to the fan inlet.

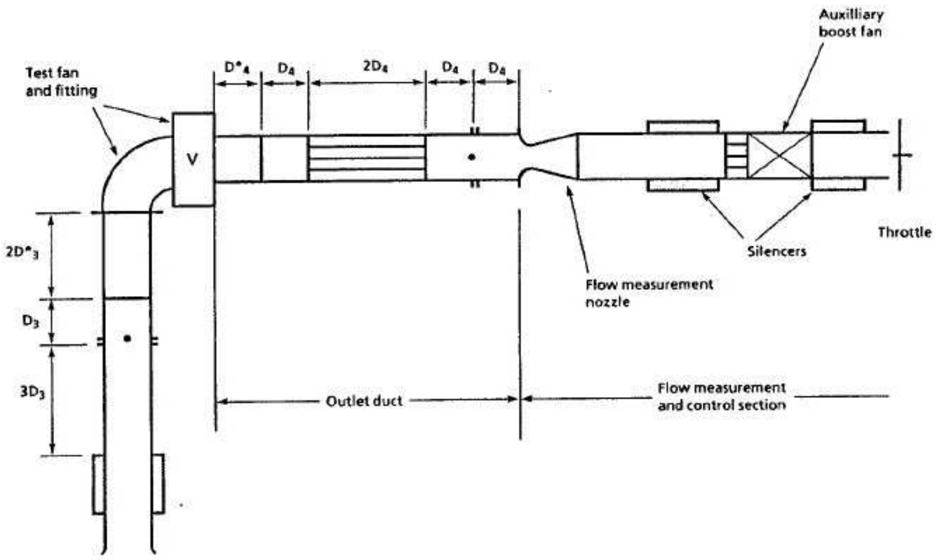

Figure 6: Test rig for determination of inlet installation effect (from [4])

Details on the experimental programme and measurement procedure are given in [4], [5] and other private reports. The test data used in the present analysis are the performance curves of the fan alone and fan + inlet fitting and the measured pressure loss of the fittings. All these curves, based on total pressure, were transformed into static pressure curves by subtracting the dynamic pressure at the fan outlet.

Table 1 gives the main characteristics of the test fans and Figure 7 shows views of these fans, some of them being used only with fitting at their outlet. Figure 8 presents sketches of the fittings that were connected to the fan inlet via transition elements.

Table 1: Main characteristics of the fans tested by NEL (from [4])

fan	fan type	diameter (mm)	hub/tip ratio	speed (rpm)
1	tubeaxial, blade setting 24°	630	0.223	1440
2	tubeaxial, blade setting 30°	630	0.223	1440
3	vaneaxial, blade setting 24°	630	0.389	1440
4	vaneaxial, blade setting 32°	630	0.389	1440
5	tubeaxial, blade setting 24°	630	0.389	2900
6	tubeaxial, blade setting 32°	630	0.389	2900
7	mixed-flow with guide vanes	630		1470
8	FC centrifugal, single inlet	630		850
9	BC centrifugal, single inlet	610		2600

Figure 7: Views of the test fans (from [4])

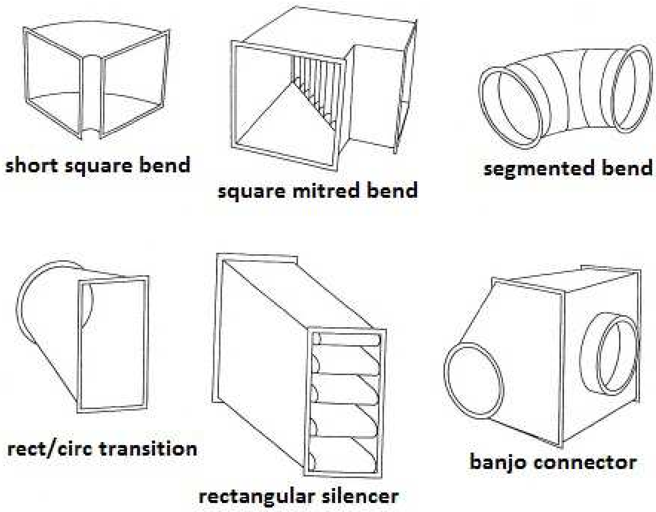

Figure 8: Inlet fittings of the NEL study (from [4])

Results

Figure 9 shows examples of curves $\Delta Qv/Qv_0$ (ξ) obtained with Fan 2 (see Table 1) and the short square bend. As anticipated the flowrate drop caused by the installation effect is larger when the bend is closer to the fan inlet.

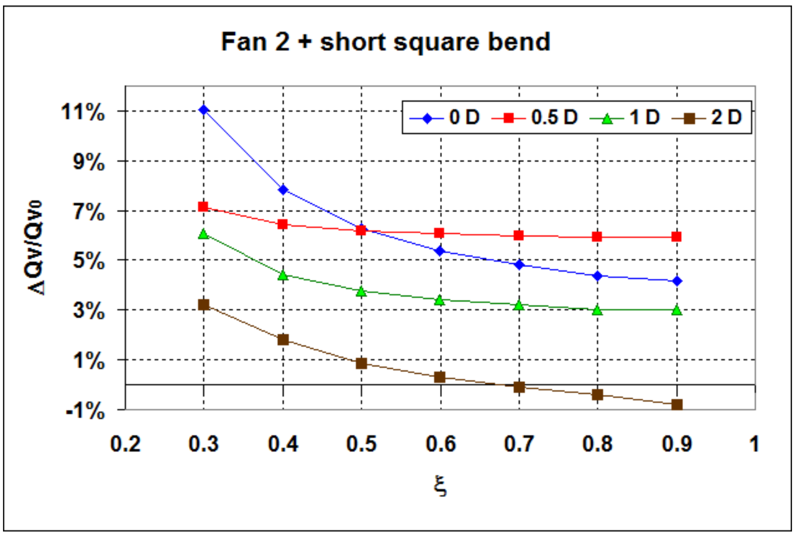

Figure 9: Curves $\Delta Qv / Qv_0\,(\xi)$ *Fan 2 with short square bend*

Table 2 presents the system effect factors (SEF) calculated according to the procedure described in 2.2 for all the fans and inlet fittings tested, with various distances L/D between the inlet component and the fan. The blank cells in the Table indicate that no measurements have been made for those configurations. Figures in black in Table 2 mean that SEF is lower than 3%, those in red indicate a SEF between 3 and 10% and when SEF is larger than 10% the figures are blue coloured. Most of the figures are black, which means that the installation effect is generally moderate. An exception is noticed with Fan 9 (BC centrifugal) and banjo connection, where SEF reaches 16.8 for L/D = 0 and even 19.6 for L/D = 2. A measurement error is not excluded for this test configuration.

Table 2: System effect factors at inlet
(black: SEF < 3%, red: 3 ≤ SEF < 10%, blue: SEF ≥ 10%)

		AXIAL AND MIXED FLOW							CENTRIFUGAL	
	L/D	1	2	3	4	5	6	7	8	9
Rect/Circ transition (a)	0	-0.4%	-0.9%	0.0%	-0.1%	-0.3%	-1.2%	0.5%	-0.4%	0.1%
	0.5						-0.5%			
	1	0.5%	0.1%	0.3%	0.0%	-0.4%	0.3%	0.9%	-0.3%	0.4%
	2	-1.0%	0.3%	-0.4%	-0.4%	-0.6%	0.7%	0.7%	-0.1%	0.7%
Short square bend (b)	0	3.7%	6.3%	0.9%	2.7%	0.4%	-0.2%	1.0%	1.3%	2.0%
	0.5	2.6%	6.2%	-0.8%	2.0%					
	1	0.3%	3.8%	1.0%	-0.6%	-0.3%	0.6%	0.7%	2.6%	1.0%
	2	0.5%	0.7%	-0.6%	-0.4%	0.3%	-1.2%	1.2%	0.0%	1.0%
Square mitred bend (c)	0	0.2%	6.4%	2.5%	0.0%	2.0%	0.5%	1.2%	2.5%	1.0%
	0.5		5.6%	2.2%	-0.5%					
	1	0.1%	5.4%	1.9%	-0.1%	0.8%	0.5%	1.2%	2.7%	1.0%
	2	-0.2%	2.5%	-0.8%	0.0%	-0.3%	-0.2%	1.6%	1.8%	1.0%
Segmented bend (d)	0	-1.0%	-1.0%	-1.9%	-0.3%	-1.6%	-0.3%	1.1%	-0.2%	-0.1%
	0.5	1.5%	1.9%	-0.2%	-0.3%					0.3%
	1	3.6%	0.6%	0.5%	0.0%	-1.4%	0.9%	1.4%	1.2%	0.4%
	2	1.8%	1.3%	1.0%	-0.9%	-1.3%	-0.7%	1.7%	-0.1%	0.8%
Rect Splitter Silencer (e)	0							2.1%		
	0.5									
	1							1.8%		
	2							1.5%		
Banjo Connection (f)	0									16.8
	0.5									
	1									
	2								0.2%	19.6

SEF is significant with Fan 2 and Fittings b (short square bend) and c (square mitred bend), with logical results case since the effect decreases when L increases from 0 to 2D. These two inlet components do not induce a noticeable installation effect on the other fans, which means that the effect depends both on the fitting type and fan geometry in a way that appears difficult to explain and anticipate.

ANALYSIS OF AMCA TEST DATA

ASHRAE Research Project 1216-RP

Test setup

The objective of this research project is to obtain a body of measured aerodynamic and acoustic inlet system effects for a single inlet backward inclined/airfoil centrifugal fan of 762 mm impeller diameter tested according to installation type B (free inlet, ducted outlet). The inlet fittings of this project are five bearings of different types with their supports (see example in Figure 10) and two cabinets of heights 2D and 3D and width L varying from 2D to 0.25D, where D is the impeller diameter (Figure 11). All the tests were performed at three fan speeds 796, 1327 and 1731 rpm. Details on the experimental setup and test methods are given in [6].

Figure 10: BI centrifugal fan with inlet bearing obstruction (from [6])

Unlike the NEL study the pressure losses of the fittings have not been measured in this work. Therefore, the "fan + fitting" curves used for the determination of SEF are not corrected to account for the fitting pressure losses, at least in a first stage.

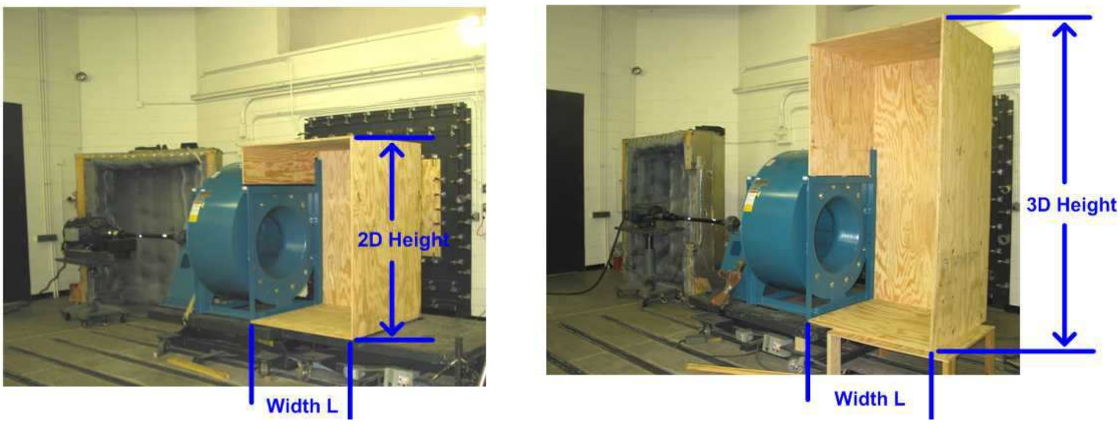

Figure 11: BI centrifugal fan with inlet cabinets 1 (left) and 2 (right) (from [6])

Results

Figure 12 shows a set of curves $\Delta Qv/Qv_0$ function of ξ obtained at 1327 rpm with inlet cabinet 1 (height 2D) for the different widths L. The flowrate drop is much higher with L = 0.25D, except in the low flow range $\xi \leq 2000$. Even if the curve for L = 0.25D is not flat the SEF is always calculated from the average of $\Delta Qv/Qv_0$ over the whole ξ range.

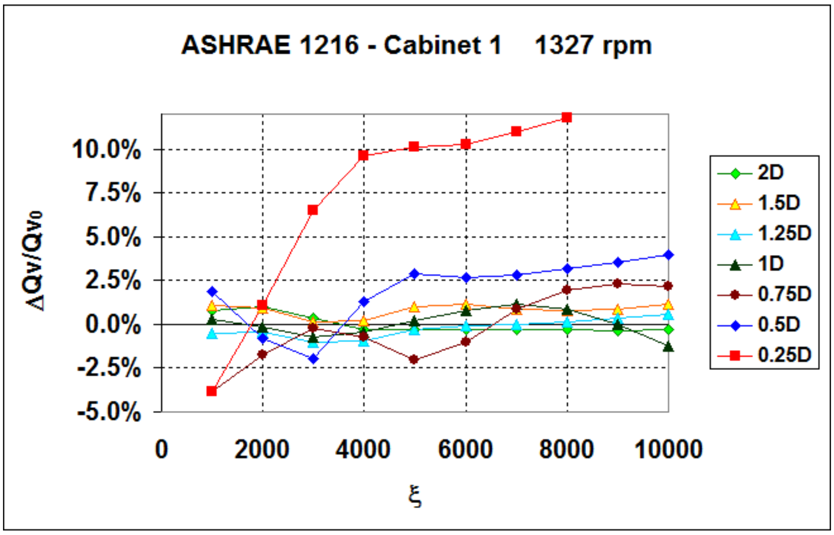

Figure 12: Curves $\Delta Qv / Qv_0\,(\xi)$ *Cabinet 1 N = 1327 rpm*

Figure 13 compares the SEF obtained on the two inlet cabinets. The SEF was averaged over the three fan speeds. It decreases when L/D increases, which was expected, and it is slightly smaller on cabinet 1 than on cabinet 2 for L/D between 0.5 and 1.25. The SEF induced by the inlet bearings, not shown in Figure 13, does not exceed 2% whatever the bearing configuration.

As indicated previously the pressure losses of the inlet obstacles have not been measured in this project, which appears justified for the bearings but not for the inlet cabinets.

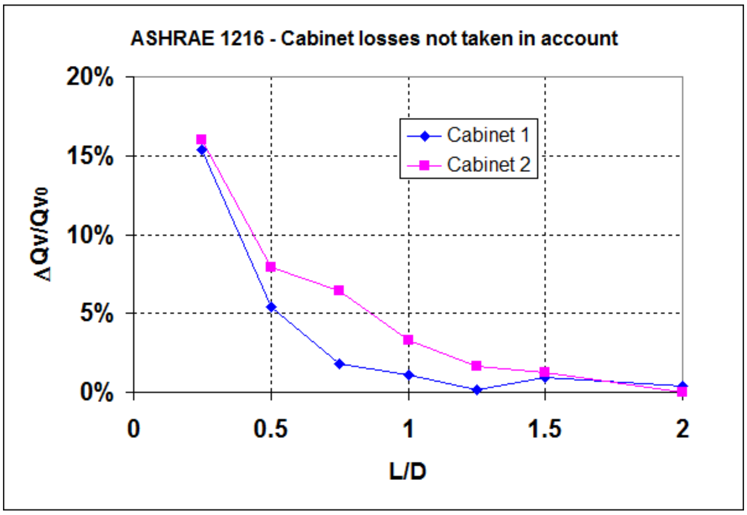

Figure 13: Comparison of SEF of cabinets 1 and 2 (pressure losses not included)

Indeed, the sharp edge entrance of the cabinet and the 90° deviation imposed to the flow by the box before entering the fan may be considered as pressure losses that could be non negligible when the width L is reduced and the flow velocity in the plenum increased accordingly.

To assess the installation effect of the cabinets with accounting for these pressure losses, a rough estimate of the losses have been made. The pressure loss coefficient ζ of the sharp edge entrance of the cabinet has been estimated to 0.9 and the loss coefficient of the bend in the plenum estimated to 1.3. The total pressure loss of the cabinet is therefore:

$$\mathbf{\Delta p = 0.5\rho\zeta V^2} \qquad \text{with } \zeta = 2.2 \text{ and } V = Qv/(L.H)$$

L varies from 0.25D to 2D, H = 2D for cabinet 1 and 3D for cabinet 2.

The SEF induced by the two cabinets when their pressure losses are accounted is shown in Figure 14. This figure clearly illustrates that cabinet 1 induces a smaller installation effect than cabinet 2 when L/D < 1.5, probably because the flow at the fan entrance is more axisymmetric with cabinet 1 due to the near symmetry of its bottom and top walls with respect to the impeller axis.

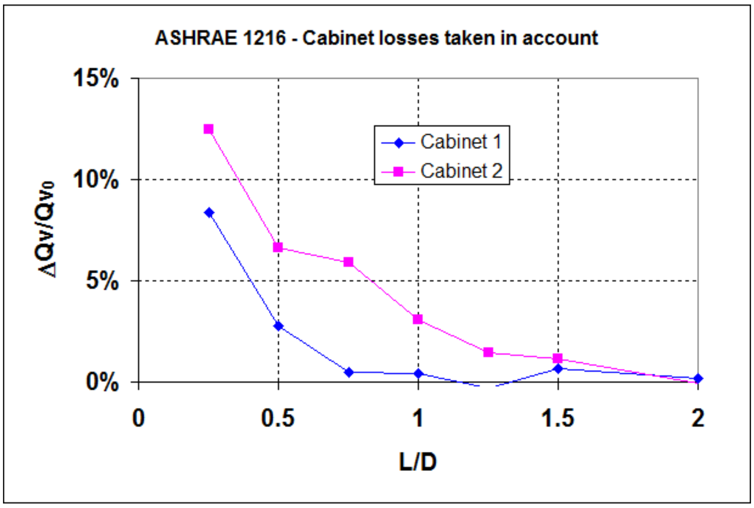

Figure 14: Comparison of SEF of cabinets 1 and 2 (pressure losses included)

Example of acoustic installation effect

From the AMCA sound test data measured in a reverberant chamber at the fan inlet it is possible to quantify a system effect on the sound power level due to the inlet disturbance. The procedure is similar to that adopted for defining SEF except that the relative flowrate drop $\Delta Qv/Qv_0$ is replaced by the difference in A-weighted inlet sound power levels $\Delta L_{WA} = L_{WA\text{ free inlet}} - L_{WA\text{ with inlet obstacle}}$ averaged over the ξ range.

Figure 15 shows the evolution of ΔL_{WA} with ξ for cabinet 1, N = 1327 rpm, and the different widths L. ΔL_{WA} is negative when the sound level of the fan with inlet cabinet is higher than the level of the fan with free inlet. As expected the noise level increases when L is reduced because of the highly turbulent flow at the fan entrance.

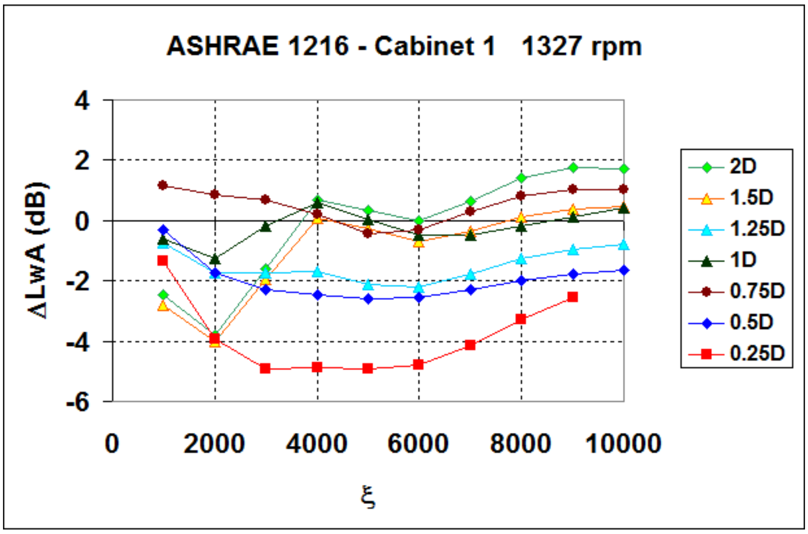

Figure 15: Curves ΔL_{WA} (ξ) Cabinet 1 N = 1327 rpm

Figure 16 compares the acoustic system effect (ΔL_{WA} averaged over ξ and then on the three fan speeds) obtained with the two cabinets.

The magnitude of the system effect decreases when L/D increases from 0.25 to 1, then it is nearly constant and close to zero. Below L/D = 1 the effect is more important on cabinet 2 than on cabinet 1. This result may be explained by the same argument as in 4.1.2 concerning the symmetry of the bottom wall and top wall of the box with respect to the fan axis. Cabinet 2 is more dissymmetric than cabinet 1 and should induce a more inhomogeneous and turbulent flow at the fan entrance than cabinet 1.

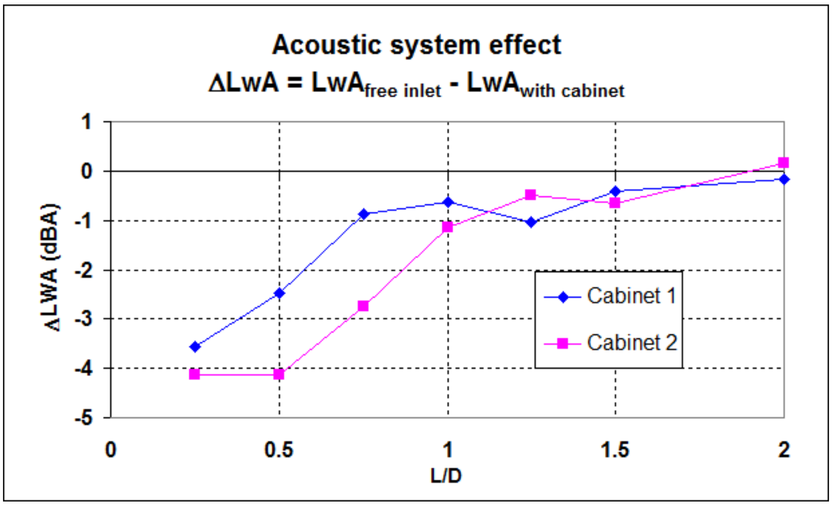

Figure 16: Comparison of acoustic system effects of cabinets 1 and 2

ASHRAE Research Project 1272-RP

Test setup

The objective and the test procedure of this research project are similar to those of 1216-RP. The test fan is a forward curved centrifugal fan of 321 mm impeller diameter and the inlet obstacles are four bearings of various types and two cabinets of heights 2D and 3D and different widths L from 0.25D to 2D. The test speeds are 1000 rpm, 1500 rpm and 2000 rpm. The details of the experimental setup are presented in [7]. Figure 17 shows a view of the fan with inlet cabinet 1 of 2D height.

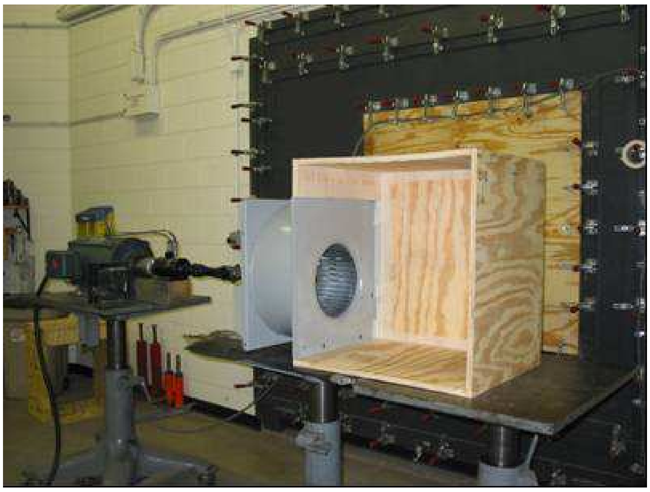

Figure 17: FC centrifugal fan with inlet cabinet 1 (from [7])

Results

Like in 4.1.2 the pressure losses of the cabinets, which have not been determined in the tests, are estimated with a loss coefficient $\zeta = 2.2$. Figure 18 compares the SEF of the two cabinets without accounting for the box pressure losses, while Figure 19 makes the same comparison with the estimated pressure losses taken in consideration.

The SEF considerably increases when L/D is reduced from 1 to 0.25. Furthermore, Figure 19 shows that the SEF is slightly larger with cabinet 2, probably for the same reason as that given in 4.1.2 for the BI centrifugal fan.

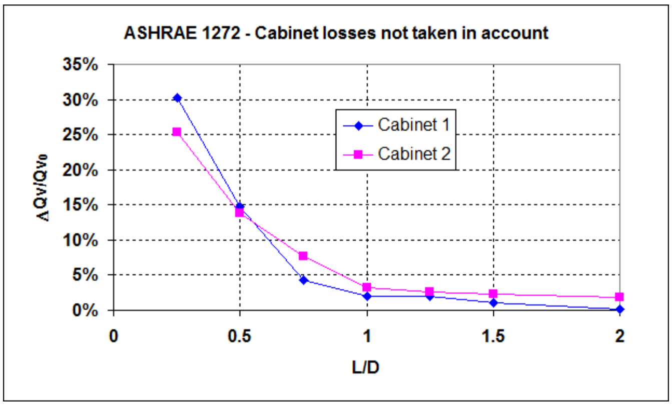

Figure 18 Comparison of SEF of cabinets 1 and 2 (pressure losses not included)

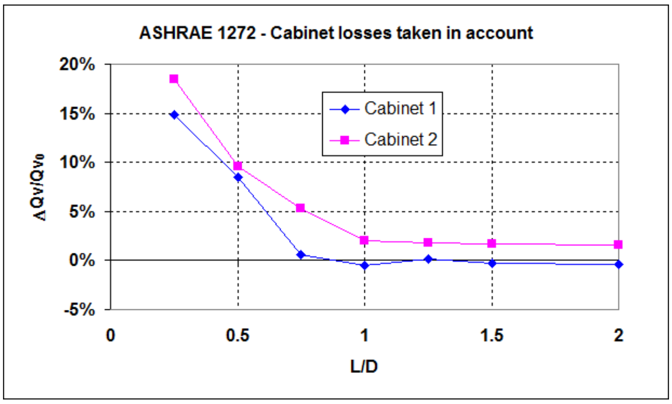

Figure 19 Comparison of SEF of cabinets 1 and 2 (pressure losses included)

CONCLUSIONS

The objective of this paper was to quantify the installation effect due to fittings or obstructions at the fan inlet. A new definition of system effect factor is proposed, based on the average percentage of flowrate drop over the fan curve due to the inlet disturbance. The analysis of test results obtained by NEL with this new SEF definition shows that the installation effect is nearly negligible (less than 3%) in most of the configurations of fans and fittings tested. This analysis also confirms that the amplitude of SEF depends on the geometry of both the fan and the fitting in a way that is difficult to explain and predict.

The analysis of the AMCA test results on two centrifugal fans of backward inclined and forward curved type is easier to understand qualitatively. It simply shows that the SEF increases when the width L of the inlet cabinet decreases, with a considerable jump between 0.5D and 0.25D. This degradation of the performance curve is accompanied by an increase of the noise level. The comparison of the results obtained on the two inlet cabinets of height 2D and 3D shows that the SEF and the acoustic system effect are slightly worse with the 3D height cabinet because of the strong dissymmetry of the bottom and top walls of the box with respect to the fan axis.

More work is needed to go further on in the interpretation of aerodynamic installation effects and the use of CFD simulations should constitute a valuable tool to analyze and predict those effects in a very near future.

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