AC-9

Summary

Overview

AC-9 is a “built in place” air handler serving portions of the East Wing. The system was built in 2013 and installed in 2014. Renovations of floors two through four were performed in phases. The AC-9 enclosure (manufactured by Semco and assembled in place) houses two Wood vane axial supply fans (installed in parallel), one Woods vane axial return fan, a supply plenum, return plenum, steam preheat coil, outside air, return air and relief dampers, pre-filter bank, supply fan discharge dampers, chilled water coil, and a final filter bank.

Our Assignment

The reported problem is that system air volume is significantly less than design. The fan manufacturer and various others have reported that the system is operating properly. Our assignment was to confirm or deny the existence of a reduced air volume condition, and if confirmed, provide an explanation and recommendations.

Investigative Work

1. All three fans were moved to the full flow position (both supply fans were done manually with jack screws and the return fan was done pneumatically).
2. The outside air and return air dampers were pneumatically locked at full return. The minimum outside air damper was locked at full open.
3. Air volume measurements were taken at the inlet to each fan and a static pressure profile was taken across the air handler.
4. Research and analysis work were performed to arrive at explainable conclusions.

Conclusion

1. The return fan and both supply fans are physically unable to deliver the required air volume when applied to this system. Replacement of all three fans or an alternative solution is required.
2. An electrical upgrade to the penthouse will likely be required to accommodate larger fans.
3. It is feasible that a corrective solution can be accomplished without significant downtime.

AC-9

Testing and analysis

Additional Overview

The East Wing of the hospital was renovated beginning in 2014. The building is four occupied stories above grade with a basement and a penthouse. Floors 2, 3, and 4 are occupied patient rooms.

Since the original penthouse renovation/installation (approx. 2014), floors 2, 3, and 4 have received HVAC renovation in phases. At the completion of the last phase, the air volume shortage was detected. Of the 50,000 CFM specified on drawing #M1 (dated 12/19/13), we were advised by the hospital's engineering department that partial system balance reports provided as each phase of renovation was accomplished, indicate that the system is providing approximately 60% of the 50,000 CFM design air volume. Our assignment and the subject of this report was to confirm or disprove the low volume condition, determine the cause and provide recommendations for corrective measures. The depth of our assignment was limited in scope by time and cost constraints.

Testing Overview

All ductwork entering or leaving the two duct shafts and the penthouse floor (4th floor is supplied from the floor) was examined to located traverse points on each floor for flow measurements in the vicinity of the duct shafts. Suitable traverse points were not available (due to congestion within the ceiling spaces), which required that we perform our measurements in the penthouse.

Sample readings were taken inside the air handler while in operation and it was determined that reasonably accurate readings could not be obtained with the system "as is". Although not ideal (and due to space limitations), 18" long round inlet ducts were fabricated and temporarily installed on the outside diameter of the inlet bell on each fan to facilitate testing, recognizing the existence of a vena-contracta at each inlet. Each supply fan was tested independently and the together operating in parallel.

During each test of the supply fans, the return fan was operated at ful commanded flow (commanded pneumatically and locked at full flow). Heavy plastic sheeting was used to seal the discharge damper of the idle fan to prevent backflow. The Johnson air handler controller was placed in manual to lock the dampers in a fixed position. Control of the terminal CAV boxes and reheat coils was left in automatic as our assignment scope did not allow time for examination of the terminal units.

Chilled water coil flow and air stream enthalpy analysis would have been preferable to corroborate our air side volume readings, however, the unavailability of chilled water and the lack of a reasonable coil load (both due to the time of year) made this impractical. Consequently, only volume flow readings were obtainable and were not corroborated by alternate methods. In addition, due to space and time constraints, testing in strict compliance with AMCA publication 203-90 (Field Performance Measurements), was not possible. Given the reasonable stability of the air stream at each fan's inlet (when operated independently) even with the vena-contracta resulting from our test ducts, we were able to obtain test data of adequate accuracy to enable us to identify the cause(s) of deficient air volume.

Our task was to explain the difference between the test data and the field measured data. Since field readings are inherently less accurate than laboratory readings, the goal therefore is to narrow the differences between the lab data and the field data and then arrive at reasonable conclusions that are directionally correct.

Test #1: Supply Fan F-3 and Return Fan F-1 Running, Supply Fan F-2 Off

1. Taped off the discharge damper for supply fan F-2.
2. Performed a 20 point, two diameter traverse on the inlet duct of fan F-3 and recorded the results.
3. Recorded the following readings:
   a. Return fan inlet pressure.
   b. Return fan discharge pressure.
   c. Pre-filter inlet pressure.
   d. Supply fan suction pressure.
   e. Supply fan discharge pressure (after the damper)/cooling coil inlet pressure.
   f. Cooling coil discharge pressure.
   g. Final filter discharge/supply plenum pressure.
   h. Duct pressure on Johnson controller (2^{nd} floor).
   i. Voltage and amperage.

Test #2: Supply Fan F-2 and Return Fan F-1 Running, Supply Fan F-3 Off

1. Taped off the discharge damper for supply fan F-3.
2. Performed a 20 point traverse on the inlet duct of fan F-2.
3. Recorded the readings outlined in Test #1 above.

Test #3: Supply Fans F-2 and F-3 and Return Fan F-1 Running

1. Performed a traverse of return fan F-1 inlet.
2. Performed a traverse of supply fan F-2 inlet.
3. Performed a traverse of supply fan F-3 inlet.
4. Recorded the readings outlined in Test #1 above.

Blade Pitch Measurement

Using measurements from the front of the fan casing flange, we measured and recorded the maximum blade pitch for each fan. In addition, using a protractor and the manufacturer's hand written instructions, we measured the blade pitch on supply fan F-2 and compared it to our calculated pitch to validate our method.

Duct Inspection

During our search for suitable traverse locations, the supply and return ducts were inspected at each floor where they entered or exited the shafts. Duct takeoffs on each floor were generally as shown on the plans with the exception of frequent slight reductions in size below those called for on the design drawings. The exact reduction can be determined if necessary from the as built drawings.

RELEVANT BACKGROUND INFORMATION

"Fan Static Pressure" vs. Fan Static Pressure Rise

The term "fan static pressure" is a term that applies specifically to fans and their curves and is not to be confused with the field measured difference is static pressure across a fan. Fan static pressure is defined as the outlet static pressure minus the inlet total pressure. This is important when evaluating fan's field performance against the stated curve performance in applications where the inlet total pressure is greater than zero or in a process application where elevated temperatures and pressures may be experienced. In the case of AC-9 however, the inlet velocity pressure is considered to be 0.0" making the "fan static pressure" equal to the measured static pressure across the fan. Fan static pressure generally appears as the ordinate (y-axis) on a fan volume-pressure curve.

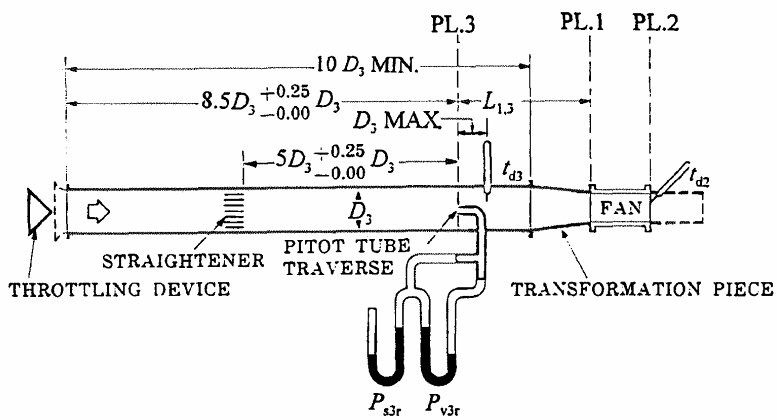
System Effect

From Table 1 (appears later in the analysis section), the static pressure rise across supply F-2 is 2.34" w.c. As measured in the field at a volume flow rate of 30,995 cfm. From the Woods fan curve for the 90KG56A4-9 (Chart #28) the advertised fan static pressure at our measured flow rte of 30,995 cfm is approximately 4.5" at a 40 degree blade pitch. The difference between our field measured fan static pressure and the catalog static pressure (.216" w.c. in this case) can be explained by the application of "system effect" factors.

System effect factors were developed by AMCA (Air Movement and Control Association) to be used by designers and diagnosticians to account for the difference between the stated performance of a given fan (under a specific test configuration) and the performance fo that same fan as installed in the field under alternate field connection configurations. AMCA Standard 210/ASHRAE Standard 51 defines the test configurations that can be used by manufacturers when certifying their fans.

Woods fans are tested in accordance with British Standard 848, Part 1, 1963, Method 1, which according to the catalog, is equivalent to AMCA Standard 210-74, Figure 16. This test configuration employs an inlet duct of at least 10 fan diameters in length (approximately 30 feet for the 90KG fan) and an outlet duct of 2-3 fan diameters (6'-9') in length. This arrangement ensures a uniform inlet velocity profile at the test point (8.5 fan diameters from the duct inlet) and an efficient conversion of velocity pressure to static pressure at the outlet. Typically one test fan of the "type" being tested is tested in the laboratory and the performance of the remaining fans (of different sizes, speeds, etc.) in the family is calculated from the test results for the single fan.

Fig 1: AMCA 210 equivalent test configuration taken from AMCA 210:



NOTES

1. Dotted lines on inlet indicate an inlet bell which may be used to approach more nearly free delivery.
2. Dotted lines on fan outlet indicate a uniform duct 2 to 3 equivalent diameters long and of an area within ± 1.0% of the fan outlet area and a shape to fit the fan outlet. This may be used to simulate an outlet duct. The outlet duct friction shall not be considered.

Any deviation from the test configuration described above (which is typical in most installations) requires the application of one or more "system effect" factors.

"System effect factors" provide correction factors in the form of additional static pressure losses, which when added to the design system static pressure (frequently referred to as external static pressure or "ESP"), normally result in an increase in the static pressure used to select the fan. The design static pressure (for a given fan family from a specific manufacturer) then becomes the sum of the actual system losses plus the correction factors (in inches w.c.) caused by installing the fan in other than its tested configurations.

System effect is not measurable and generally results in the requirement of a larger fan within a given fan family type, or in the selection of a different type of fan. When the "larger" fan is then applied to the system in the alternate configuration (and against the actual external system static pressure losses), it produces the volume flow required by the designer. The larger fan will also not plot on its fan curve and the difference will also be "system effect", but the system will operate at the design air volume.

Fan "Stall"

Single Fan

The opportunity for "fan stall" exists with most fan types and is most noticeable in fans with a characteristic "stall dip" to the left of the peak pressure point on the fan curve. Vane axial fans fall into this category. Fan stall occurs when air volume is reduced and the operating point of the fna moves up the curve and then to the left of the peak pressure point. Air volume can be reduced intentionally by various methods (blade pitch adjustment, speed changes, etc.) without creating stall, or unintentionally by the fan or fans not being able to overcome the system static pressure. Fans will normally move up and down along their curve until they reach equilibrium with they system (where the fan curve and the system curve intersect) which is known as the operating point. There is only one operating point possible when a fan and system curve intersect in the stable portion of the fan curve.

Fig. 2 : Typical vaneaxial fan curve showing "stall dip"

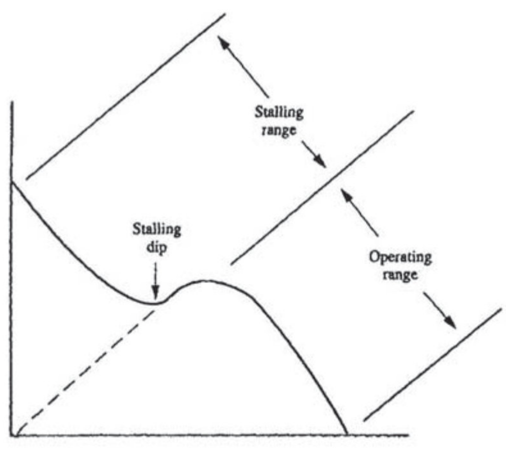


FIGURE 4.38 Static pressure versus air volume for a vaneaxial fan with a large hub-tip ratio and with large blade angles.

In the “stall range” or “unstable range” (See Fig. 2 above), the mechanics of what happens internally to the fan is the subject of much discussion, even today. Common terms such as stall, rotating stall, surge, paralleling, hunting, bi-stable flow, etc., all have different meaning, but all describe an undesirable condition that exists when a fan operates in the unstable range on its curve. How far into the stall range a fan operates determines the severity of the condition which can range fro hardly noticeable to catastrophic failure.

The mechanical of “fan stall” in vane axial fans

Most vane axial fans have blades that are shaped like an airplane wing and are known as “airfoil” blades. Vane axial fans move air and build pressure (static pressure) by “deflection”. Air travels into the fan where the airfoil blade imparts energy and “deflects” the air stream toward the rear of the fan (axial flow). Practically speaking, centrifugal force is not a factor in axial fans. Centrifugal fans on the other hand (backward inclined fans, forward curved fans, etc.), also known as radial flow fans, impart energy to the air stream by airflow deflection and centrifugal force and the air travels “radially” at approximately 90 degrees to the fan shaft.

In vane axial fans, rotation of the airfoil blade on the impeller creates a low pressure area along the top of the “wing” which draws air into the fan. The low pressure area is created by a reduction in static pressure as the velocity (and therefore velocity pressure) increase as the air moves over the top of the wing (greater distance than the back side of the wing). The air maintains contact with the inlet side (top) of the wing as it moves along the surface through what is known as a coanda effect. Air captured by the back side of the wing moves along the rear surface at a lower velocity (the travel distance is shorter) until the two streams join at the trailing edge of the wing which is directionally aimed to the side and rear of the fan. Guide vanes installed along the motor barrel gently straighten the air stream until it ts truly traveling parallel with the shaft and has achieved axial flow.

When stall occurs in a vane axial fan, the air moving along the top of the “wing” can no longer maintain surface contact through coanda effect, and the particular wing design has reached its maximum lift coefficient (aka ability to produce pressure). The air on the face of the wing then breaks away (partially or totally depending on the degree of stall) and moves outward (radially) toward the casing wall. During stall, the air stream on the face of the wing splits into two streams; one which follows the normal path and joins with the stream on the back of the blade at the trailing edge and moves out into the system, and another which moved radially outward toward the casing wall where it encounters the leading edge of the next blade on the impeller. The severity of the stall determines how much air moves in which direction. Stall creates turbulence, reduces capacity, increases power consumption and creates mechanical wear and fatigue on fan components. As stall worsens, noise and vibration increase and air moves back out the inlet bell or duct in what may appear to be reverse flow. In reality, this is more likely to be recirculation of inlet air. As this continues the static pressure drops and then rises as the fan moves through the “stall dip” and up the curve until it reaches the shutoff pressure at zero flow. Earlier it was stated that axial flow fans do not produce static pressure by centrifugal force; however, in severe stall to the left of the saddle (stall dip), centrifugal force does actually play a role in building pressure. This is not however a desirable condition. Figure 3 below illustrates normal and stall flow over an airfoil blade or “wing”. Airplane pilots strive to prevent stall because when stall occurs in an airplane, the plane tends to drop like a rock.

Fig. 3: Normal airfoil flow vs. stall flow, from Bleir Fan Handbook
(Note: the angle of attack is <u>not</u> blade pitch)

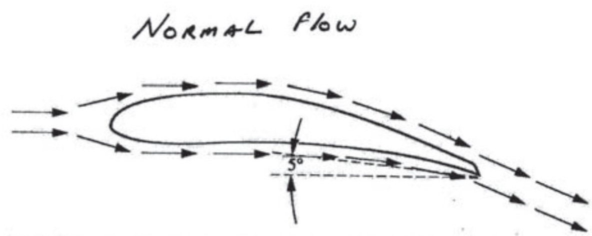


FIGURE 2.6 At a 5° angle of attack, the airflow is smooth and follows the contours of the airfoil. The direction of the airflow is deflected by 13.5°.

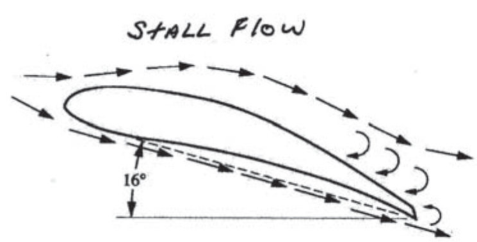


FIGURE 2.7 At a 16° angle of attack, the airfoil stalls, and separation of airflow takes place at the trailing edge and at the suction side of the airfoil, with small eddies filling the suction zones. The deflection of the airflow past the trailing edge is close to zero.

Fig. 4: Simple illustration of vaneaxial fan stall

AXIAL-FLOW FANS 4.33

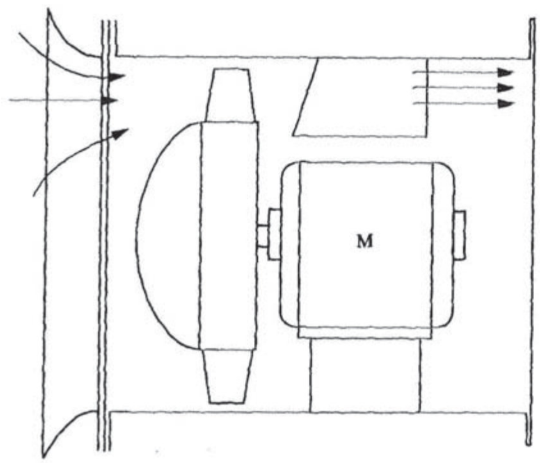


FIGURE 4.40 In the good operating range, the airflow passing through a vaneaxial fan is smooth and has no radial components.

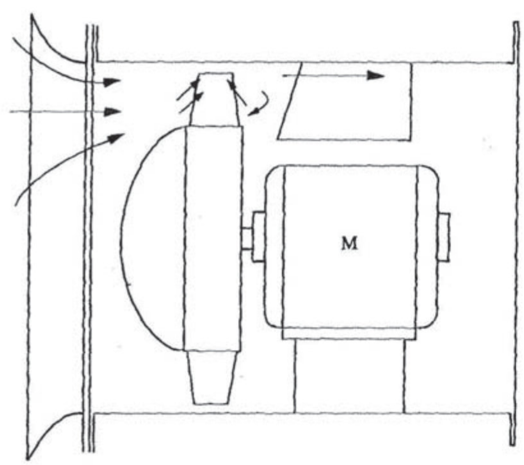


FIGURE 4.41 In the stalling range, the airflow passing through a vaneaxial fan is turbulent, with eddies and with radially outward components.

Vane Axial Fans in Parallel (General)

Applying two or more fans in parallel is common practice in many installations and is an effective way of providing limited redundancy. Vane axial fans are commonly installed in parallel however due to their characteristics, care must be taken to ensure the following:

1. The fans are selected with sufficient static pressure safety margin to prevent them from operating in the “stall” range. This applies to single fans as well. Some studies suggest that the selection point be below (on the y-axis) the lowest point in the stall dip.
2. Capacity control is provided by one common signal to ensure that the blade angles are within a few degrees of each other.
3. Large blade angles (above 24 degrees) are avoided to prevent pressure derating as presented by Woods in a technical article dating to 1954. Woods claims that at blade angles above 24 degrees for two fans in parallel, the maximum pressure development for the two fans is reduced below that of a single fan. According to Woods, for blade angles of 28 degrees and 32 degrees, expected pressure development throughout the curve must be reduced by a multiplier of .95 and .89 respectively. Forty degree pitch angle was not covered in their technical paper. Due to the 1954 date of the article, further research into its validity is advisable to avoid compounding safety factors.

Stall with Vane Axial Fans in Parallel

When two vane axial fans operate in parallel, the new combined fan curve results in a “longer” (from left to right) unstable range. The slope of the combined pressure-volume curve in the stable range remains unchanged as generally does the static pressure (see Wood’s comment on derating parallel fans due to blade angle), but the volume at the various operating points is double that of a single fan.

Assuming that fan size, blade pitch, speed, and the physical installation are identical, fans in parallel will move up and down the stable portion of their curve as if they were one fan until they reach the peak pressure point where the unstable region begins. The exact unstable condition that exists in any situation (stall, surge, paralleling, etc.) depends on the system and the application problem. Moving slightly to the left of the peak pressure point, subtle differences in the installation (slight differences in blade angle, obstructions in front of one fan’s outlet, which fan starts first, etc.) may cause the fans to move away from one another (volume wise) causing each fan to operate at a different volume point on its individual curve but at the same static pressure point. This is the stall condition believed to exist in the case of AC-9. We can confirm this by alternating which fan starts first in the sequence and then observing that the more severe stall occurs in the fan started last in the sequence. If time had permitted and if the pneumatic positioners were in working order, we could have, for information purposes, started both fans at 5 degree blade pitch and then gradually increased the blade pitch on both fans to determine where exactly the stall point exists for the two fans in parallel.

Fig 5: Illustration showing instability with two fans in parallel (AMCA)

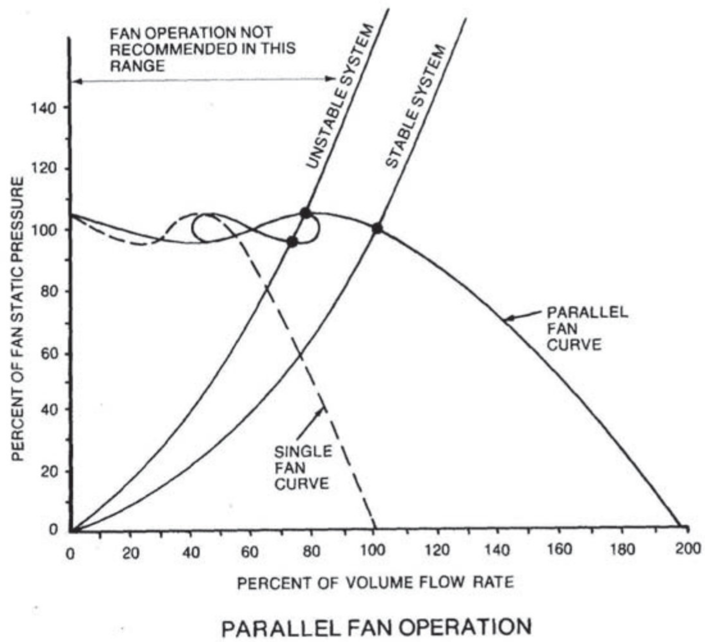


Figure 39

Analysis

Test Notes

1. All filters were in place and in visually clean condition.
2. When observing the operation of the outside air/return air dampers, we noted that the dampers hunted from full open to full closed repeatedly when in automatic.
3. We were informed by the maintenance staff that the fan manufacturer's service personnel adjusted the maximum pitch stops to achieve full load for each fan. We did not disturb the stop settings; however, measurements indicate that F-3 is adjusted to 45 degree pitch angle which is past the maximum recommended angle of 40 degrees.
4. Upon arrival, we found the pneumatic swivel joint for fan F-2 broken. It had recently been replaced by others and had failed within a few days. The cause of failure is likely the vibration caused by the stall condition. For our testing we reinstalled the manual jack screws on both supply fans to achieve maximum pitch. The jack screws were left installed and at maximum pitch upon completion of our work.
5. With either single supply fan running, operation was smooth, however erratic pressures existed in the motor barrel of fan F-3 likely due to the previously adjusted 45 degree blade.

6. When both supply fans operate in parallel, fan F-3 experiences significant “stall” indicating that it is operating in the unstable range (stall range) on its curve. Significant “rumbling” was present and the air pattern observed in the inlet had axial and radial components confirming severe stall.
7. When operated in parallel, F-2 operates at reduced flow and to the left of the peak static pressure point in the stall range although to a significantly lesser degree than F-3.
8. During parallel operation we were able to move the greater stall condition from one fan to the other by alternating which start switch was turned on first.
9. Our velocity measurements were taken using a pitot tube above 5,000 ft/min and an airfoil probe below 5,000 ft/min. Our experience indicates that greater accuracy is achieved using the airfoil probe in areas where low or negative readings may exist due to the presence of “eddies”. Given the existence of a vena-contracta at our traverse point, the likelihood existed for some negative readings along the outer circumference of the test area which proved to be true, although to a lesser degree than expected.
10. Our velocity readings were corrected (automatically by the instrument) for local density and temperature at the traverse point.
11. To improve accuracy, three velocity readings (with each reading being an average of several seconds automatically averaged by the instrument) were taken at each point. The average of the three “instrument averaged” readings was recorded for each point of the traverse.
12. Each supply fan delivers approximately 31,000 CFM when operated independently at a maximum blade pitch of 40 degrees (as specified by the manufacturer) and with the idle fan discharge opening sealed.
13. The total volume with both fans operating in parallel increases only by approximately 13% to 35,000 cfm under unstable conditions.

Measured and Calculated Data vs. Estimated Data at Design Volume

The table below compares the data obtained through field measurements and estimates the fan static pressure requirements at the design airflow. Supply fan F-2 and return fan F-1 were chosen as the basis for the estimated performance at design airflow since F-2 was at, and not past the maximum blade pitch.

Table 1 Measured vs. estimated static pressure requirements:

Item	Measured resistance (inches w.c.) at 30,995 cfm	Estimated resistance at 50,000 cfm

mechanicalservicecorporation

Outdoor air damper allowance (test had full return with the minimum outside air damper open)	0	0.0
Return pressure at the pre-filters (Pre-heat open)	-0.01	.03
Pre-filter clean pressure drop	0.13	0.34
Pre-filter allowance for loading (2 times clean loss for economy on pre-filters)	0	0.68
Dry cooling coil pressure drop	0.25	0.65
Allowance for wet coil (15%)	0	0.1
Final filter pressure drop	0.5	1.30
Final filter allowance for loading (1 times clean loss)	0	1.30
Supply side plenum pressure	1.45	3.78
Residual pressure required for proper operation of CAV terminal units	0	0
Sub total of losses less "system effect" (2.34" was the actual ∆P across the fan)	2.34" w.c.	8.18" w.c
System effect #1: Discharge damper	.83	.54
System effect #2: direct discharge to a chamber	1.6	.95
Sub total of "system effect"	2.43	1.49
Total supply side system resistance (inches W.C.)	4.77" w.c.	10.29" w.c.

Table Notes:

1. In the estimated column, a value of 0 is shown for the outside air dampers for economizer operation. Because the 0.03" w.c. required for the minimum outside air damper/pre-heat coil is greater than the pressure drop of the outside air damper at 40,000 cfm during full economizer, nothing was added for the economizer to avoid redundancy.
2. The application of system effect is specific to the installation and is velocity dependent. If the fans are replaced, the anticipated system effect for that design must be used.
3. Wet coil pressure drop must be confirmed prior to selecting alternate fans.
4. A line item for residual pressure at the terminal units is included for consideration. This should be taken into account if required when selecting new equipment.
5. System effect losses for the estimated column are for the existing fan type for comparison purposes only and assumes two fans at 25,000 cfm each to be consistent with the original design.
6. The velocity pressures for the fan annulus area and the fan outlet area on the 90KG fan curve (P_{da} and P_{df} respectively on Chart 28) correspond to those for a fan with a constant diameter motor barrel (approx. 4.19ft^2) and an outlet area equal to the inlet area (approx. 6.83 ft^2). These dimensions correspond to the

drawings in the Woods catalog but not to the drawings provided with the submittal (which agree with the installed fans). This issue creates some difficulty when attempting to match the actual field performance of the fan to the laboratory performance shown on the curve (Woods Chart 28 for a 90KG56A-4-9 varofoil fan). This is because system effect is velocity dependent and because the actual fan static pressure performance (due to static regain) is likely to be slightly better than the curve indicates. Exactly how much better would require retesting the fan in a lab or an opinion from someone with more experience in laboratory fan testing.

7. For the system effect of the discharge damper, the Woods catalog suggests 0.5 x the velocity pressure. AMCA publication 201 suggests 3.3 x the manufacturer's published pressure loss. We used the AMCA method due to the conflict with the velocity data on the Woods fan curve.
8. The table above does not include any loss that may result from the 18" long test duct installed to allow us to traverse the fan inlets as we had difficulty locating reliable system effect information for this configuration. Since adding a system effect value for this would serve to add a little more static pressure to our test, we chose not to include it since we used the airfoil probe for our perimeter measurements which took into account the vena contracta at the traverse location.

System Curve

Actual vs. Design

The actual system curve vs. the design system curve (with filter loading added) is plotted below with a typical vane axial curve showing the stall region (complete curves were not available from Woods). The plot in Figure 6 below generally illistrates what is occurring with AC-9.

Fig 6: System curve plot on a typical vaneaxial fan curve for illustration purposes

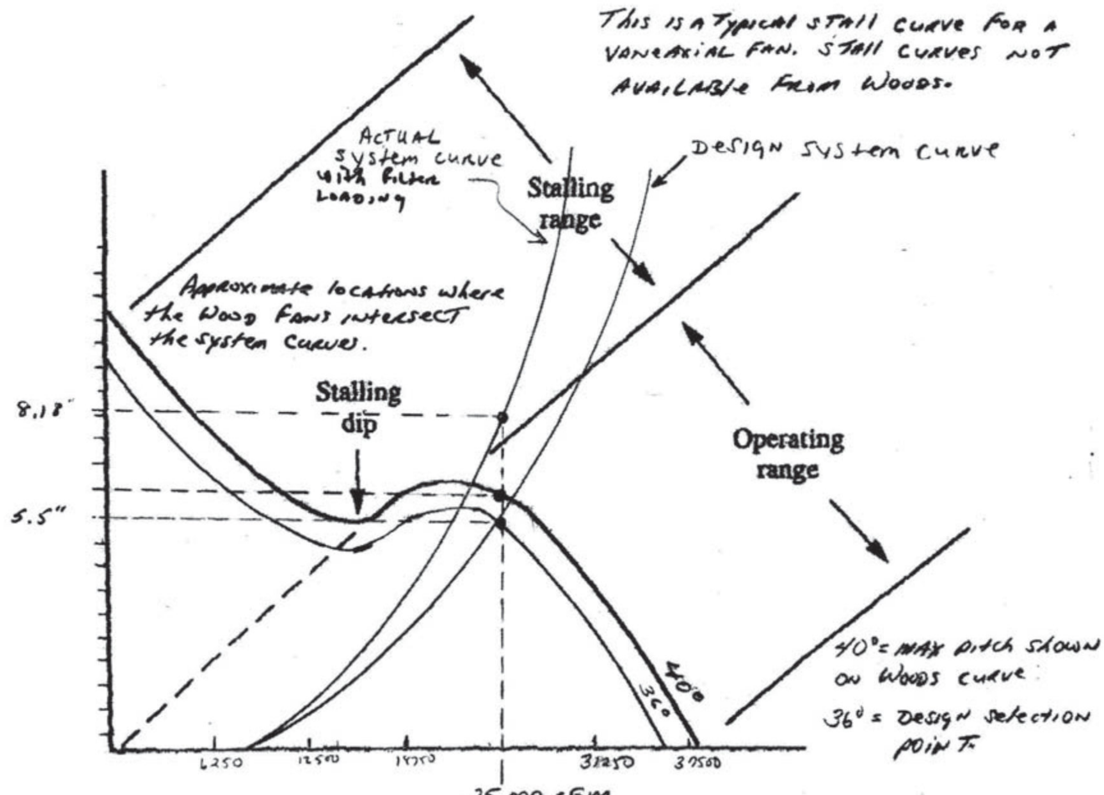
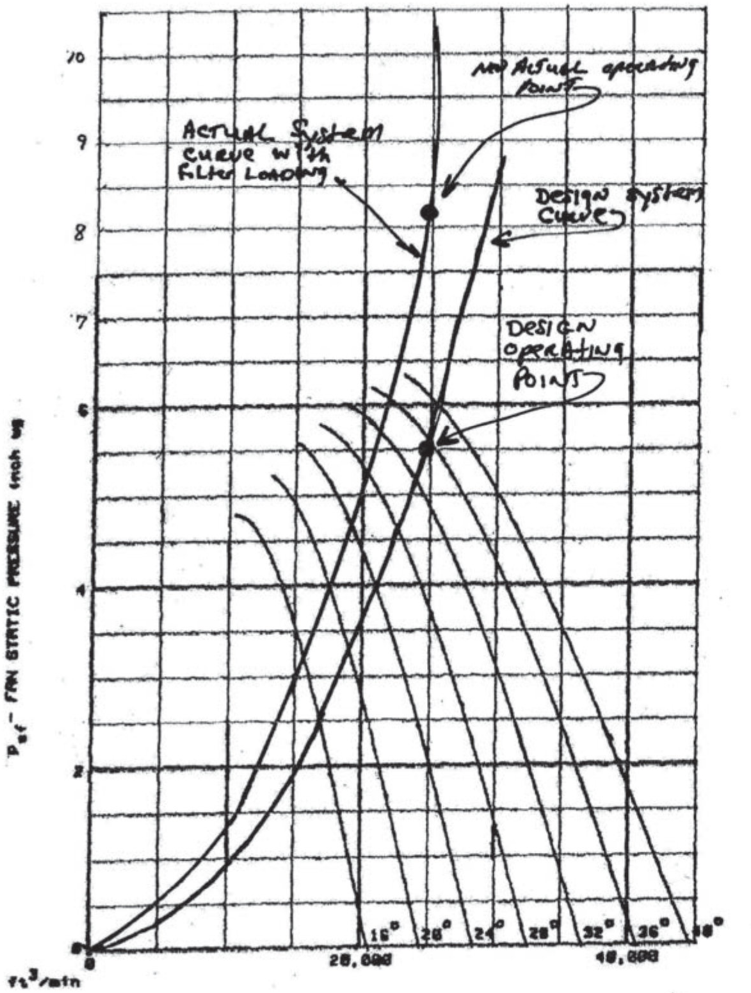


FIGURE 4.38 Static pressure versus air volume for a vaneaxial fan with a large hub-tip ratio and with large blade angles.

In Figure 7 below, we extended the graph for the pressure-volume curve for the Woods 90KG56A-4-9 supply fan and then added the design and actual system curves for illustration purposes.

Fig 7: Design and actual system curves overlaid on existing fan curve

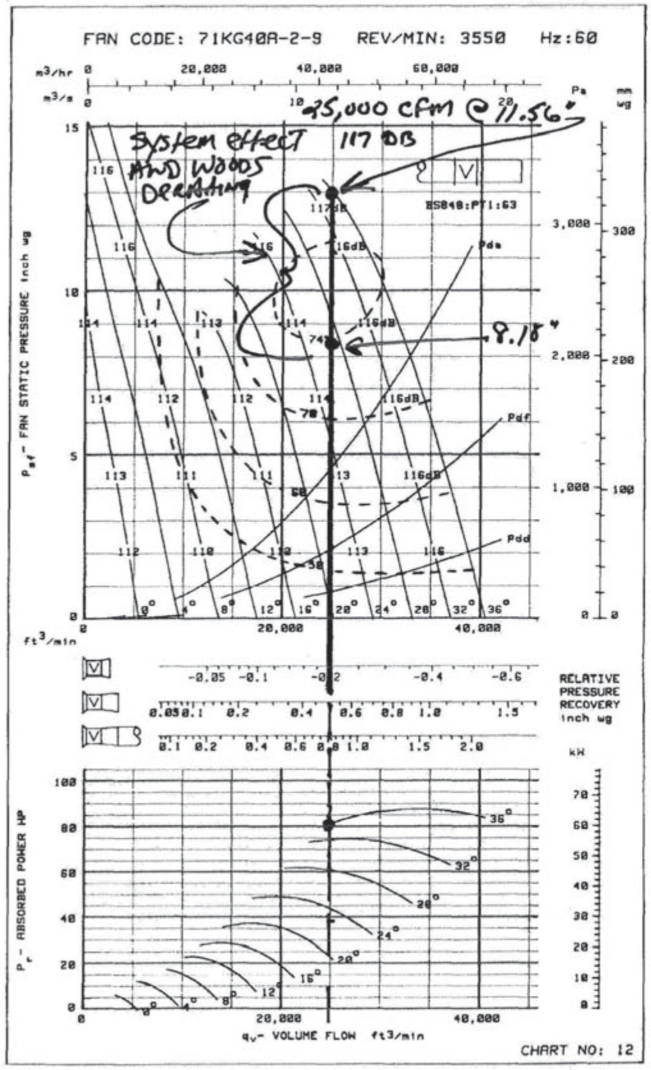


Valley AC-9 System curves overlaid on a Woods 90KG56A4-9 fan.

Alternate Fan

For illustration purposes, Figure 8 below shows an alternate Woods fan that approximately satisfies the flow and pressure requirements (although the selection within 5% of the peak pressure point and is not recommended by Woods). Do not use this as the basis for corrective measures.

Fig. 8: 3500 RPM vaneaxial fan as an alternate sample selection



Notes

1. Research from Woods (although old) indicates that vane axial fans perform differently than centrifugal fans when applied in parallel with a blade pitch above approximately 24 degrees (two fans in parallel). In situations such as this, Woods states that a given set of fans will produce less static pressure than would be expected by approximately 10% -15% depending on pitch. This requires further research due to the age of the material.

2. When selecting vane axial fans, a static pressure safety margin of 30%-50% is recommended as a balance between efficiency and guarding against forced operation in the stall range (Bleir, Fan Handbook).
3. System effect factor for any given situation is velocity dependent.

Analysis Conclusions

1. Based on the explanations in the preceding sections and the results listed in Table 1, it is our opinion that the Woods supply fans are operating as advertised and are not deficient in performance. The supply fans as applied, however, operate in stall and at greatly reduced capacity against the actual system resistance. “System effect” further exacerbates this situation.
2. The return fan is not capable of delivering the design flow at the actual return system system resistance. Failure to increase the return fan volume will likely result in greater than design outside air.
3. The resistance of the system as installed is greater than the design static pressure indicated on the drawings.

Recommendations

Replace all three fans with fans capable of operating at the static pressure required by the system as installed taking into account “system effect”.

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- Various engineering articles from Twin City Fan & Blower

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