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FAN AND SYSTEM CURVES

INTRODUCTION

Optimum matching of a fan to a system requires an understanding of both fan and system curves. While much is known and understood regarding fan curves, the same cannot always be said for system curves

Fans are selected for a specified duty point representing the fan airflow rate and pressure. Systems are specified for a design point of operation representing the system airflow rate and pressure. Together, the fan duty point and the system design point of operation represent the intended point of operation at the intersection of the fan and system curves. The actual point of operation is the point of intersection of the fan and system curves at which the system operates when energized. It is not uncommon for a system to have more than one design point of operation for intersection at corresponding fan duty points on either the same fan curve or a new fan curve.

An example of a system having more than one design point of operation at corresponding duty points on the same fan curve would be the "clean" and "dirty" conditions represented by a dust collection system where the pressure varies between these conditions. In this case, the actual point of operation starts at the airflow rate and pressure representing the clean condition. As the dust collector media loads with contaminants, the pressure increases and the airflow rate decreases until the dirty condition is reached. The dust collector cleaning cycle is then energized, returning the actual point of operation back to the clean condition. The slope of the system curve is changing in response to the change in system pressure, while the fan curve remains unchanged. See Figure 1 where A represents the airflow and pressure at clean conditions and B represents the airflow and pressure at dirty conditions.

An example of a system having more than one design point of operation at corresponding duty points on a new fan curve is when the fan is operated with an inlet vane damper or variable frequency drive (VFD) to maintain a constant airflow rate while pressure varies. This occurs in a dust collection system when the airflow rate is held constant while the pressure varies with the contaminant loading (resistance) on the filter media. For fans using an inlet vane damper, the system design point of operation for the clean condition would be at a point on the dampened fan curve. As the pressure increases the damper is opened to produce a new fan curve at the same airflow rate but at the higher pressure. See Figure 2 where A represents the airflow and pressure at clean conditions and B represents the airflow and pressure at dirty conditions.

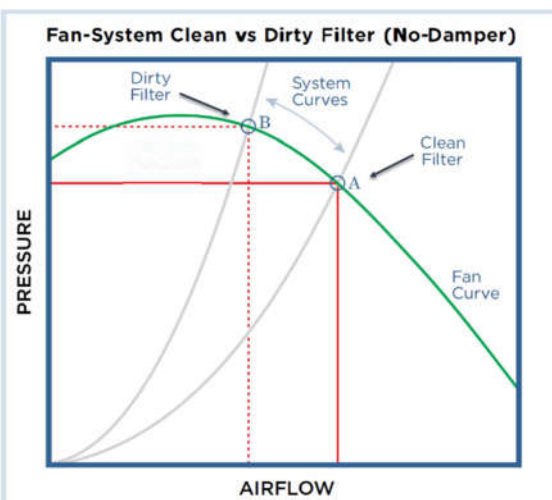

Figure 1

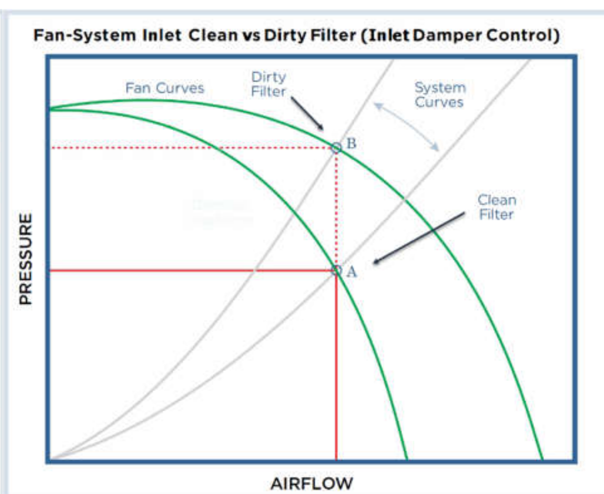

Figure 2

For fans using a VFD, the fan speed is increased or decreased for the same purpose. In both cases, more than one control device may sometimes be necessary (inlet vane damper, VFD or outlet damper). See Figure 3 where A represents the airflow and pressure at clean conditions and B represents the airflow and pressure at dirty conditions.

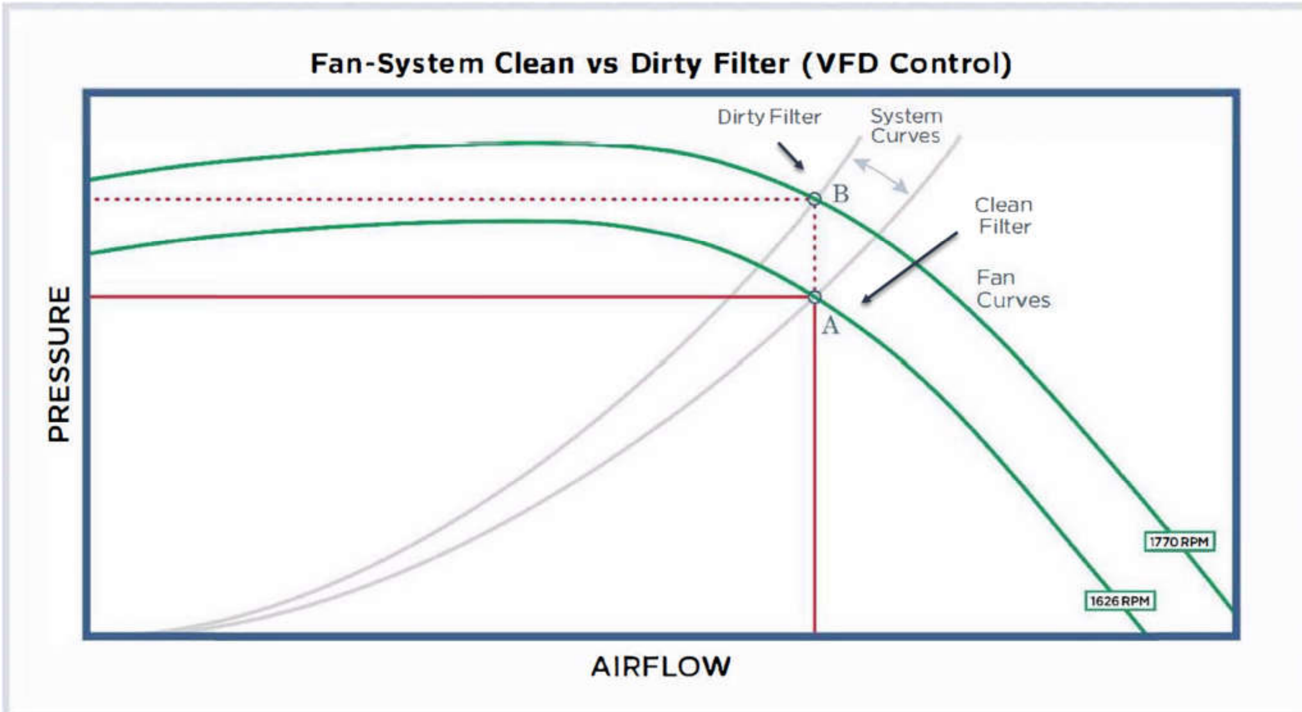

Figure 3

Notice that in Figure 1, only the system curve changes in response to the change in pressure while in Figures 2 and 3 both the fan and system curves are changing. For this reason, a clear understanding of fan and system curves is necessary to optimize fan and system performance.

Airflow Regions

A common assumption is that fans and systems always operate in wholly turbulent flow conditions. While this is certainly true of fans, it is not always true of systems. This is important because the method for calculating both the predicted fan speed, airflow, pressure and power and the system airflow and pressure is different when operating in turbulent and non-turbulent flow regions. For the purpose of this discussion, a non-turbulent flow region is defined as any region outside the turbulent flow region, i.e., transitional or laminar flow regions.

Figure 4 depicts a simple graph illustrating laminar, transitional and turbulent flow regions based on the Reynolds number for ducted airflow. The Reynolds number (Re) is a value representing the ratio of momentum to viscous forces of the airstream. As momentum forces decrease, the airstream approaches laminar flow conditions and as the momentum forces increase, the airstream approaches turbulent flow conditions.

In laminar flow conditions, the airstream flow vectors are stable against all disturbances and the airflow streamlines move in parallel. This region is defined by a Re up to 1160 (≈ 0-30 fpm), and normally occurs in stable, low velocity system components.

In transitional flow conditions, the airstream flow vectors fluctuate between laminar and turbulent flow conditions. This region is defined by a Re between 1160 and 3000 (≈ 30-100 fpm), and can occur in any unstable, low velocity system component.

In turbulent flow conditions, the airstream flow vectors are chaotic, rotational, angular, and move in the general direction of flow. This region is defined by a Re of 3000 (≈ 100 fpm) and higher and occurs in fans and throughout an air moving system. Turbulent flow conditions are also responsible for the dynamic losses associated with system components such as elbows, branch entries, contractions and expansions, etc.

Certainly, industrial fan and air moving systems operate well above 100 fpm and in the turbulent flow region – except when they do not. Non-turbulent flow conditions are found in system components such as filtration media, mist eliminators, wet scrubbers, fluidized beds, etc. In these cases, the system is referred to as a *Hybrid Flow System* instead of a *Turbulent Flow System*, since some portions of the system operate in turbulent flow conditions while other portions operate in non-turbulent flow conditions.

It is important to identify the flow regions existing in each system so that the points along the system curve can be accurately calculated and predicted in order for the fan and system curves to intersect as intended.

Airflow Regions: Ducted Airflow

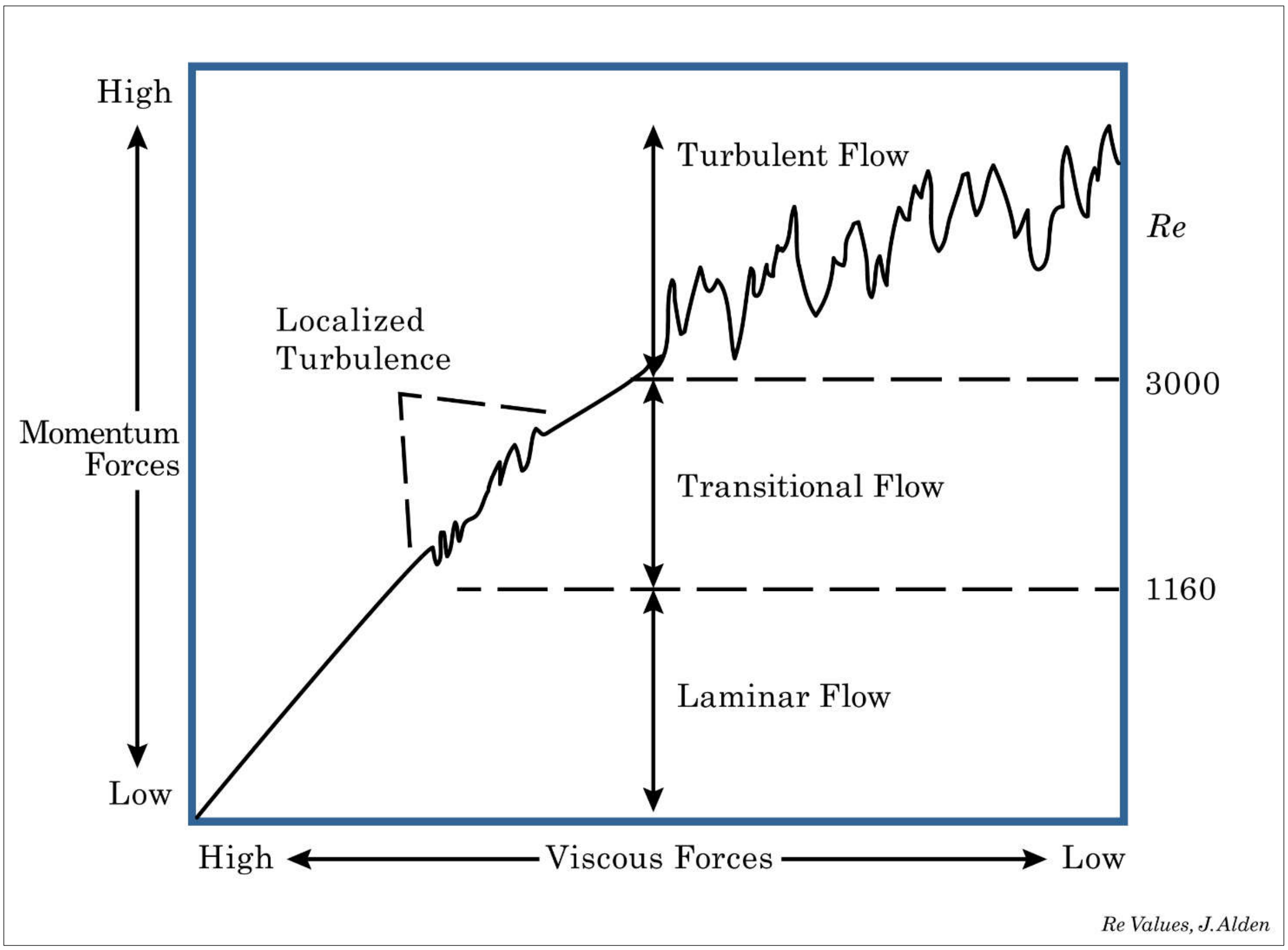

Figure 4

Turbulent Flow Systems

Because fans always operate in wholly turbulent flow, fan curves are calculated for turbulent flow conditions. When systems are operating in wholly turbulent flow, the points along the system curve are also calculated for turbulent flow conditions and the system is referred to as a *Turbulent Flow System*.

Since the Affinity Laws apply specifically to turbulent flow conditions, they can be directly applied to both fans and turbulent flow systems[1]. When applied to fans, they are commonly referred to as the Fan Affinity Laws or simply, the Fan Laws. With the density factor df, representing the change in airstream density (ρ_{act}/ρ_{std}) between points, then for a specific fan:

Flow (Q) is proportional to the change in speed (RPM) and conversely, speed is proportional to the change in flow:

$Q_2 = Q_1 (RPM_2/RPM_1)$, or
$RPM_2 = RPM_1 (Q_2/Q_1)$

Pressure (P) varies with the square of the change in flow or speed:

$P_2 = P_1 (Q_2/Q_1)^2 (df_2/df_1)$, or
$P_2 = P_1 (RPM_2/RPM_1)^2 (df_2/df_1)$

Power (HP) varies with the cube of the change in flow or speed

$HP_2 = HP_1 (Q_2/Q_1)^3 (df_2/df_1)$, or
$HP_2 = HP_1 (RPM_2/RPM_1)^3 (df_2/df_1)$

When the Affinity Laws are applied to systems for calculating points along the system curve:

To determine new system static pressures corresponding to known or target airflow rates, the corresponding system static pressures are calculated as

$P_2 = P_1 (Q_2/Q_1)^2 (df_2/df_1)$

To determine new system airflow rates corresponding to known or target system pressures, the corresponding airflow rates are calculated as

$Q_2 = Q_1 (P_2/P_1)^{0.5} (df_2/df_1)^{-0.5}$

Example 1 – Figure 5: Turbulent Flow System (TF)

In Figure 5, Fan Curve-1 operates at 1500 rpm and intersects the system curve at Q_1, 10,000 cfm and P_1, 16 "wg static pressure. When the system is operating in wholly turbulent flow conditions as depicted in system curve "TF", the Affinity Laws can be used to accurately predict additional points for both the fan and system curves. In this example, selecting Q_2 for 20,000 cfm then:

$RPM_2 = RPM_1 (Q_2/Q_1) (df_2/df_1)$
$RPM_2 = 1500 (20{,}000/10{,}000) (1)$
$RPM_2 = 3000$

$P_2 = P_1 (Q_2/Q_1)^2 (df_2/df_1)$
$P_2 = 16 (20{,}000/10{,}000)^2 (1)$
$P_2 = 64$ "wg

In this case, the fan speed increases to 3000 rpm and the fan curve intersects the system curve as predicted at Q_2, 20,000 cfm and P_2, 64 "wg.

Hybrid Flow Systems (HF)

When systems are operating in a combination of turbulent and non-turbulent flow, the points along the system curve are a hybrid calculation of the turbulent and non-turbulent flow conditions and the system is referred to as a *Hybrid Flow System*.

There are portions of some hybrid flow systems which in theory may appear to be operating in turbulent flow, but are not. Examples where the Re is above 3000 (≈ 100 fpm) but the airflow responds as if in non-turbulent flow includes a) passage through filter media where the flow characteristics are altered by the matrix of the filtration media, the influence of contaminants, or both, b) wet scrubbers where the flow characteristics are altered by the effect of water in the airstream, operation at a constant pressure, or both, and c) flow across non-ducted surfaces.

Filtration media may have laminar or non-laminar flow characteristics. In laminar flow, the media loss varies directly with the change in flow rate, while in non-laminar flow the media loss varies exponentially with the type of media and contaminant. Consult the media supplier for exact characteristics and values.

Fan speed and power values for hybrid flow systems cannot be projected from the Affinity Laws and must be derived from new fan curves. Correction to the airstream density, if applicable, only applies to the turbulent flow portion of the hybrid flow system.

System curves for four (4) common types of hybrid flow systems can be developed as follows:

Example 2 – Figure 5: Hybrid Flow System With Non-Laminar Flow Media (HF_1):

This is a type of hybrid flow system with operation in non-turbulent flow conditions where the loss across a system component varies exponentially with the change in airflow rate, but does not follow the square relationship of a turbulent flow system. This exponent normally ranges from 1.4 to 1.8. The exponent $^{1.6}$ is selected as a reasonable approximation since as it approaches lower or higher values, the difference is insignificant or immeasurable. Examples include wet or dry filtration media, catalyst beds, scrubber packings, mist eliminators, etc.

The system curve for this type of hybrid flow system is shown in Figure 5, system curve HF_1, is calculated as:

$P_2 = P_{1\text{-media}}\,(Q_2/Q_1)^{\approx 1.6} + [P_{1\text{-tf}}\,(Q_2/Q_1)^2\,(df_2/df_1)]$
$P_2 = 6\ (20{,}000/10{,}000)^{\approx 1.6} + (10)\ (20{,}000/10{,}000)^2\ (1)$
$P_2 = 18 + 40$
$P_2 = 58$ "wg sp

The new predicted system design point of operation is Q_2, 20,000 cfm at P_2, 58 "wg sp. The new fan speed and power cannot be predicted using the Affinity Laws and can only be determined from a new fan curve using the new fan duty point of 20,000 cfm at 58 "wg sp.

Example 3 – Figure 5: Hybrid Flow Systems With Laminar Flow Media (HF_2):

This is a type of hybrid flow system with operation in laminar flow conditions where the loss across a system component varies directly with the change in airflow rate. Examples include safety filters or other low velocity system components having stable airstream flow vectors.

The system curve for this type of hybrid flow system shown in Figure 5, system curve HF_2, is calculated as:

$P_2 = P_{1\text{-media}}\,(Q_2/Q_1) + [P_{1\text{-tf}}\,(Q_2/Q_1)^2\,(df_2/df_1)]$
$P_2 = 6\ (20{,}000/10{,}000) + (10)\ (20{,}000/10{,}000)^2\ (1)$
$P_2 = 12 + 40$
$P_2 = 52$ "wg sp

The new predicted system design point of operation is Q_2, 20,000 cfm at P_2, 52 "wg sp. The new fan speed and power cannot be predicted using the Affinity Laws and can only be determined from a new fan curve using the new fan duty point of 20,000 cfm at 52 "wg sp.

Example 4 - Hybrid Flow Systems With a Constant Pressure Component (HF_3):

This is a type of hybrid flow system with operation in either turbulent or non-turbulent flow conditions where the loss across a system component or a portion of the system is constant over a range of airflow rates. Examples include pressurized plenums, combustion chambers, wet venturi or impingement scrubbers, fluidized beds, etc.

The system curve for this type of hybrid flow system shown in Figure 5, system curve HF_3, is calculated as:

$P_2 = P_c + [P_{1\text{-}tf}\,(Q_2/Q_1)^2\,(df_2/df_1)]$
$P_2 = 12 + (4)\,(20{,}000/10{,}000)^2\,(1)$
$P_2 = 12 + 16$
$P_2 = 28$ "wg sp

The new predicted system design point of operation is Q_2, 20,000 cfm at P_2, 28 "wg sp. The new fan speed and power cannot be predicted using the Affinity Laws and can only be determined from a new fan curve using the new fan duty point of 20,000 cfm at 28 "wg sp

Hybrid Flow Systems With Dissimilar, Non-Turbulent Flow Conditions:

In cases where a hybrid flow system has dissimilar non-turbulent flow conditions, the system curve is calculated as a composite of the above formulas as follows:

$$P_2 = [P_{1\text{-media}}\,(Q_2/Q_1)^{\approx 1.6} + P_{1\text{-media}}\,(Q_2/Q_1) + P_c] + P_{1\text{-tf}}\,(Q_2/Q_1)^2\,(df_2/df_1)$$

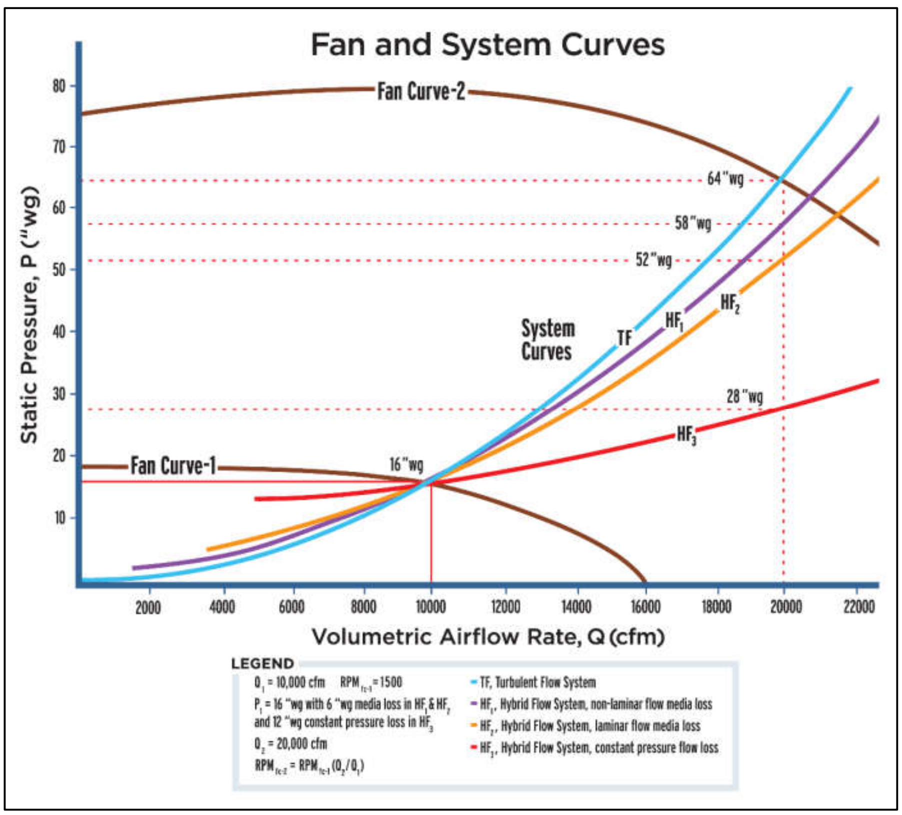

Figure 5

CONCLUSIONS

Understanding fan and system curves to accurately predict successful operation at the desired intersection of the fan and system curves requires:

a) Knowing and understanding how to apply the Affinity Laws to fan and turbulent flow system curves

b) Recognizing when a system is operating as a hybrid flow system, and

c) Understanding how to determine the system curve characteristics of a hybrid flow system to predict the system design point(s) of operation for matching with the corresponding duty point(s) of the fan curve.

Density factor corrections, if applicable, apply to both fans and turbulent flow systems but only apply to the turbulent flow portion of a hybrid flow system.

From Figure 5 it is seen that:

a) The Affinity Laws can be used to calculate new fan curves for turbulent flow systems, but not for hybrid flow systems.

b) For hybrid flow systems, new fan curves must be derived from the new design point(s) of operation in order to ensure that the fan curve intersects the hybrid flow system curve at the intended point of intersection.

In hybrid flow systems in which the non-turbulent flow pressure value is small or insignificant in proportion to the overall system pressure, the hybrid flow system can be calculated as a turbulent flow system with only an immeasurable or insignificant deviation from the actual values.

REFERENCES

[1]The New York Blower Company; Engineering Letter 2 “Fan Laws and System Curves”

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