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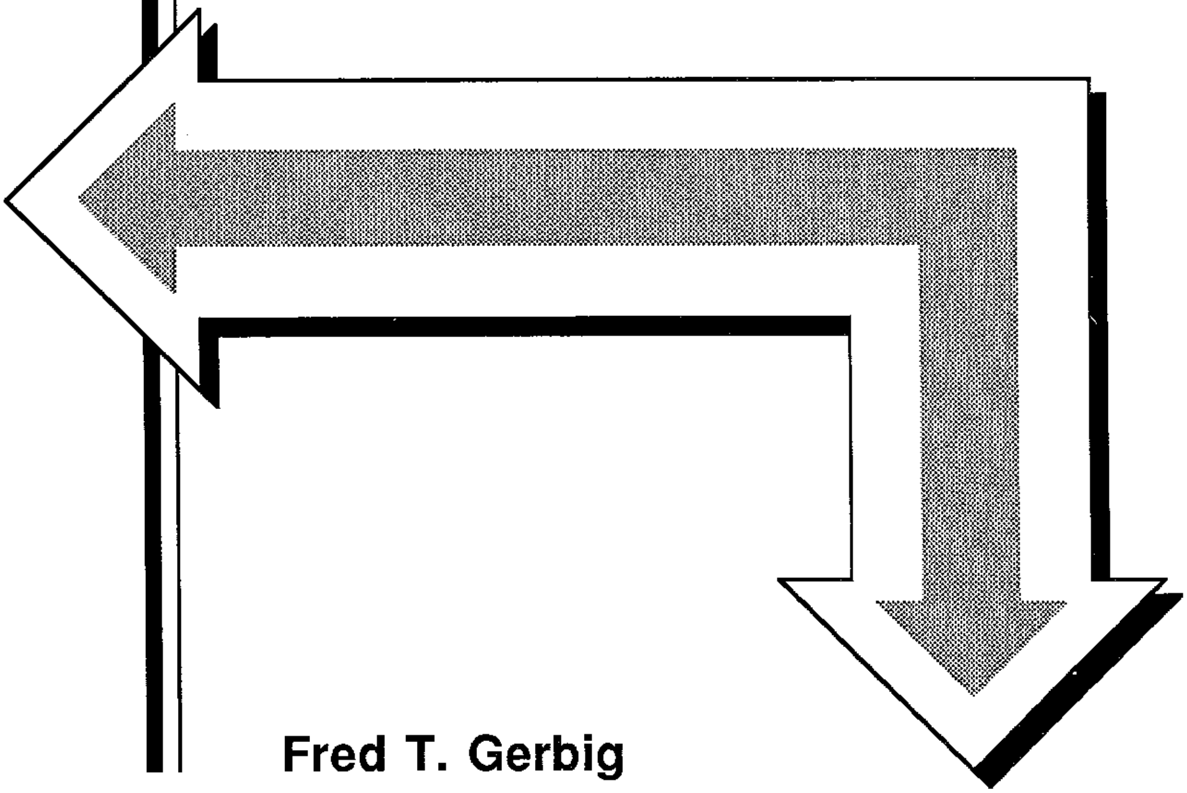
VERTICAL LAMINAR-FLOW CLEANROOMS



Fred T. Gerbig

ENERGY CONSUMPTION IN VERTICAL LAMINAR-FLOW CLEANROOMS

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> ***"What is usually overlooked or ignored is that VLF energy costs vary widely among system designs."***

Industrial users of vertical laminar-flow (VLF) clean-rooms have recognized both the need for and the energy- intensive nature of these systems. Cleaner air, resulting in higher product yields, can quickly pay for the large expenditure required for VLF. What is usually overlooked or ignored is that VLF energy costs vary widely among system designs. Where one user pays $55/sq ft or more for mechanical system operating costs, another user is paying only $13/sq ft. This wide disparity indicates a general lack of knowledge on the part of designers, as well as a lack of discipline on the part of users and owners.

To account for the wide variation in operating costs, this article examines the components of VLF energy use and shows how they relate to system design. The article also shows how to optimize system efficiency.

ENERGY COMPONENTS

If an electric bill for a cleanroom were subdivided into its component energy costs, it might look like the list presented in the following box. If electric bills were itemized in this way, designers would be under greater pressure to find solutions to the energy problems in VLF systems. The reason for the lack of attention to energy costs is that most designers have their background in the heating, ventilating, and air conditioning (HVAC) industry. In an ordinary HVAC design, lighting is a major heat gain consideration, and the cost of operating an air-handling system is low owing to the small volumes of air handled. An extra inch of pressure on an HVAC system is not significant. But when air-handling systems are applied to VLF, the flow rates are increased by roughly 100 times.

ITEMIZED UTILITY BILL FOR A TYPICAL VLF FACILITY

Direct Charges	Indirect Cooling Charges
Air pressure generation	Heat gain of lights
Fan efficiency	Heat gain of air pressure
Motor efficiency	Heat gain of fan inefficiency
Drive losses of fans	Heat gain of motor inefficiency
Lighting	Heat gain of drive losses
Process load	Heat gain of process load
Delivery of makeup air	People and miscellaneous heat gains
Reflected power	Humidity control

Small errors in system design become very costly at these high flow rates. Now, lighting becomes a small fraction of the total energy and heat gain.

If we ignore lighting, process load, makeup air, and exhaust air, we can then itemize the electric charges in the utility bill. These charges are directly controlled by the system designer. For simplicity, energy rates in watts per square foot VLF will be utilized. It will be assumed that the flow rate is 90 cu ft/min/sq ft, or 90 ft/min.

FAN SYSTEM ENERGY

Any handbook on fans[1] will give the following relationship between pressure, horsepower (bhp), flow, and efficiency:

$$\text{bhp} = \frac{\text{flow} \times \text{pressure}}{K \times \text{efficiency}} \qquad [1]$$

If flow (F) is expressed in cubic feet per minute and pressure (P) is expressed in inches wg, then

$$\text{bhp} = \frac{F \times P}{6356 \times E_f \times E_d} \qquad [2]$$

where E_f is fan efficiency and E_d is drive efficiency (belts, frequency inverters, etc.). Pressure and efficiency must both be based on either total or static measurements. (Vane-axial fan manufacturers use total pressure and total efficiency, whereas centrifugal fan manufacturers use static pressure and static efficiency.)

To convert bhp into watts required by the motor, we use the following expression:

$$\text{watts} = \frac{\text{bhp} \times 746}{E_m} \qquad [3]$$

where E_m is the motor efficiency.

"Now it can be seen that the energy used in the fan system is paid for again by the cooling required."

Combining equations 2 and 3,

$$\text{watts} = \frac{F \times P \times 746}{6356 \times E_f \times E_m \times E_o} \qquad [4]$$

For 1 sq ft of VLF at 90 cu ft/min, the equation becomes

$$\frac{\text{watts}}{\text{sq ft}} = \frac{10.56 \times P}{E_f \times E_m \times E_o} \qquad [5]$$

This equation now represents the energy rate of the fan system over which the designer can exercise some control.

This equation does not, however, include the energy required to cool the heat of the fan system. For each watt input by the fan system, 3.41 Btu/h of heat will be realized in the cleanroom enclosure. The heat of the motors may or may not contribute to the heat in the space, depending on motor location. Some air-handling systems have the motors outside the airstream so that heat from motor inefficiency is added to the mechanical room and not to the cleanroom. Most typically, however, cleanrooms have the motors in the airstream.

The cooling load (Btu/h) of the fan system is then 3.41 times the wattage. The energy rate to cool is equal to the heat divided by the energy efficiency ratio (EER) of the air conditioning system. EER is the cooling effect in Btu/h of the air conditioning divided by the electrical input in watts.

The total energy rate, including the cooling is then

$$\frac{\text{watts}}{\text{sq ft}} = \frac{10.56 \times P}{E_f \times E_m \times E_o} \left[1 + \frac{3.41}{\text{EER}} \right] \qquad [6]$$

Now it can be seen that the energy used in the fan system is paid for again by the cooling required. In fact, if EER is 10, then the cooling energy is approximately one-third of the fan system energy. An EER of 10 is typical for many air conditioning systems. Most industrial users know the EER value for their system. Of course, all the motors used to deliver the conditioned air must be taken into account when the energy efficiency ratio is being calculated.

"A number of methods are available that reduce flow during unoccupied conditions."

MINIMIZING ENERGY CONSUMPTION

The energy rate described by Equation 6 must be multiplied by the hours of use to obtain total energy use in watt-hours. A prime consideration for reducing operating energy is therefore the hours at peak pressure. For unoccupied hours, flow rates may be safely turned down to one-third to one-half of design flow. A number of methods are available that reduce flow during unoccupied conditions. These include using two-speed motors, using frequency inverters, reducing the number of operating fans, changing the pitch angles on vane-axial fans, and using inlet vanes on centrifugal fans. Of these, the most attractive is the use of two-speed motors. At half speed, the flow is reduced by 50% and the wattage by more than 60%, even though motor efficiency drops about 10% on the lower speed. Frequency inverters will reduce amperage draw by two-thirds at half flow, but will add considerable cost in comparison to two-speed motors. (Fan-shaft horsepower will be reduced by a cube function of fan speed, but the motor and drive characteristics change at the reduced load, so that at half speed the motor wattage will not reduce to one-eighth of full load.)

Inlet vanes should not be employed, because of the penalty in the full-load condition. Inlet vanes will add 8-20% to the bhp at design load as a result of air obstruction.

REDUCING PRESSURE

Reducing design flow energy is largely dependent on reducing pressure. The author has observed very large variations in system pressure of VLF rooms. Where a well-designed system is operating at 1.2 in. wg, another is operating at 7 in. wg. No other parameter in the system can vary this much! One reason for the wide variation in pressure is the type of cleanroom chosen.

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> ***"Once a system has been selected, the designer must study each component in the airflow loop that contributes to system pressure."***

VLF rooms may be categorized into three distinct types: pressurized plenum system, ducted terminal filter systems, and modular fan-filter systems. Of these, the pressurized plenum designs have the greatest advantage in reducing system pressure. This is because the fans may be close-coupled to the plenum, opportunities exist to recover fan velocity pressure in the plenum, and ductwork can be eliminated.

The ducted terminal filter system suffers additional losses owing to the overhead duct, branch takeoff, flexible duct, and sudden expansion of the air entering the filter shroud. George Cadwell of Flanders Filter, Inc., tested a pressurized plenum system and a ducted system on a project where all other parameters were equal (i.e., similar design, same fans and attenuators, etc.). The only difference in the two rooms was in the type of ceiling. Cadwell found that the operating pressure in the ducted system was 1.0 in. wg higher than that found in the pressurized plenum system. He found a loss of 0.3 in. wg at the collar entering the filter module and a loss of 0.3 in. wg at the fitting takeoff from the trunk duct. The other 0.4 in. wg loss was attributed to the ductwork and to loss of velocity pressure.

The modular fan filter approach can also minimize system pressure. However, most modular systems use small, forwardly curved fans, smaller motors, and belt drives, which can offset the advantage gained through the reduced pressure.

Once a system has been selected, the designer must study each component in the airflow loop that contributes to system pressure. Life-cycle cost analysis can readily be used to determine the economic trade-offs of various components in the system. This is not the "business as usual" approach of an HVAC system design. For example, using high-efficiency prefilters to extend the life of a HEPA filter will prove very costly for the owner. Such prefilters could add $6/sq ft to the system's annual operating cost.

Probably the most overlooked component of a system is fan velocity pressure. Typically, velocity pressure at the fan outlet represent 30% of the total system pressure. This is especially true of vane-axial fans, whose nature is to generate a larger velocity head than that generated by centrifugal fans. Loss of the velocity head is referred to as the "system effect factor."[2] Wherever possible, fan discharges should be carefully examined for ways to recover the velocity pressure, converting as much energy as possible to static pressure. The final or lowest velocity pressure in the VLF system (at 90ft/min) is essentially zero. Thus, the usable pressure is static.

OPTIMIZING COMPONENT EFFICIENCIES

Each component used to deliver air has an associated energy loss. Fan efficiency can vary widely. Large centrifugal fans operate at efficiencies of 75-80%, whereas small, forwardly curved fans operate at 50-60%. Vane-axial fans can have efficiencies (based on total pressure) exceeding 80%. All of these are peak values. Selection of the operating point by the designer may not be at peak efficiency, because of size constraints, allowance for filter loading, and cost. Special attention should be given to selecting a fan that will operate close to peak efficiency for the life of the system. Figure 1 shows a comparison between a large centrifugal fan (at 75% efficiency) and a small fan at 55% efficiency.

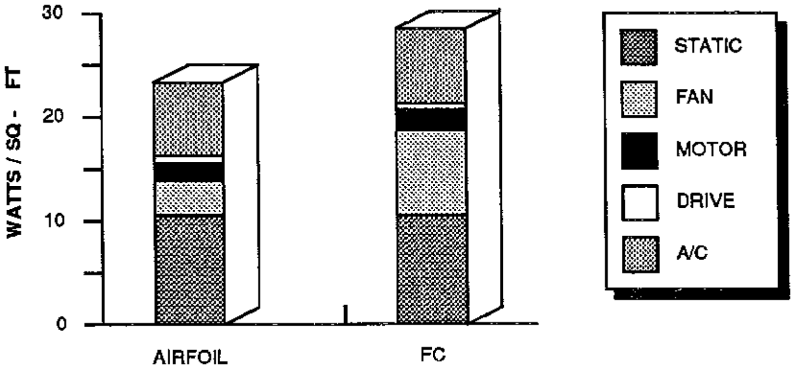


Figure 1.

ENERGY COMPARISON

FAN TYPE COMPARISON

In the conversion of motor electrical energy to mechanical energy, a portion is lost in the form of heat. Consider the following two motors: one is a standard 7- 1/2-hp motor, while the other is a high-efficiency type of the same capacity (Table 1). In a typical cleanroom at 3 in. wg, the higher-efficiency motor would reduce the energy rate by nearly 4 watts/sq ft. Usually, this savings is enough to warrant the extra initial expense involved. Since most public utilities charge on the basis of kilovolt-amperes and not kilowatts, the higher power factor saves additional operating expense. With either standard or high-efficiency motors, power factor correction to 0.90 or more should be considered. Since there is an ample supply of technical literature on power factor correction, it will not be discussed futher here.

Motor Type	Percent Load	E_m*	Power Factor	Amperage
Standard	100	85.7	78.0	10.5
	75	85.7	72.1	8.5
	50	83.6	61.2	6.9
	25	74.9	40.6	5.8
High-Efficiency	100	91.0	86.0	9.0
	75	91.7	82.6	7.0
	50	91.5	74.6	5.2
	25	87.8	53.2	3.8

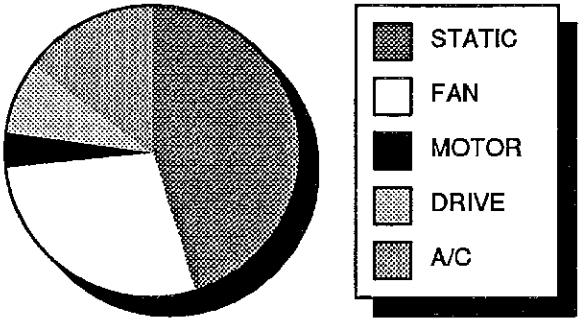
* Guaranteed minimum efficiency

Belt drives for smaller motors can be less efficient than the motor in transmitting the required power. Figure 2 shows this quite clearly.[3] The best solution to belt losses is to eliminate belts completely and use direct-drive fans.

System pressure is the variable over which the designer can exercise the most control. Using typical industry values for efficiency factors, it is possible to graphically demonstrate operating costs as a function of system pressure. Figure 2 is drawn using the following efficiencies: fans, 75%; belts, 94%; motors, 88%; and an air conditioning EER of 9. Once system pressure has been determined, belt losses and motor losses are added as a percentage of fan energy. Then the total energy input must be cooled, resulting in the final values of Figure 3.

Figure 2.

ENERGY RELATIONSHIPS



Fan Efficiency at 75%
Belt Efficiency at 94%
Motor Efficiency at 89%
A/C EER of 9

Figure 3.

ENERGY CONSUMPTION

Watts/SQ-FT versus pressure

PRESSURE	1	2	3	4
STATIC	10.56	21.12	31.68	42.44
FAN	3.52	7.04	10.56	14.08
MOTOR	1.44	2.88	4.32	5.76
DRIVE	.67	1.35	2.02	2.70
A/C	6.14	12.27	18.41	24.54
TOTAL	22.33	44.67	66.99	89.32

SYSTEM REVIEW

To illustrate the wide variation in VLF energy consumption, five systems were measured using a power factor meter and current transformer. The data were converted to watts per square foot, as shown in Figure 4. This figure excludes the energy required by the air conditioning system.

Systems A, B and F are pressurized plenum designs, which have significantly lower energy consumption than the other three systems. System B used centrifugal fans and had a sensible cooling coil included in the recirculated air loop, accounting for a slight increase in pressure.

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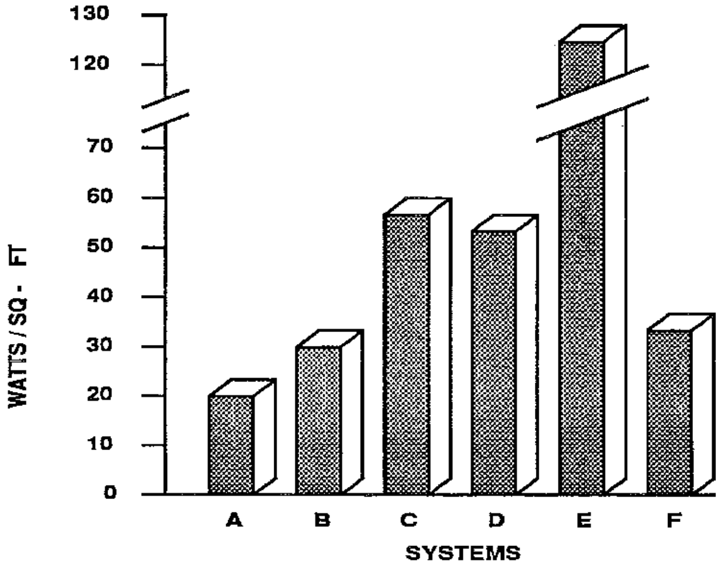
Figure 4.

ENERGY GRAPH



The fan filter installation shown as System D had no walls or return grilles and consisted only of a prefilter and HEPA filter. Fans were forwardly curved and belt-driven with three-phase integral horsepower motors. Smaller modules using single-phase motors can consume as much as 111 Watts/sq ft.[4]

The ducted module installations represented as systems C and E used similar-sized vane-axial fans. The primary causes of high energy consumption in System E were extra ductwork, several transitions from square to round, and, most importantly, the loss of the velocity head caused by a poor discharge condition.

Installation A also used van-axial fans, but used little ductwork and contained a discharge fitting designed to recover velocity pressure.

As it turns out, systems A and E engaged in the same industry, and both represent sizable installations. If the area of these installations is 10,000 sq ft, if both systems are charged 6 cents per day at a two-thirds reduction in energy, the annual operating cost paid by the owner of System E amounts to over $400,000 per year more than System A. That's significant!

CONCLUSION

Although the capital costs of designing and installing a VLF cleanroom are large, it should not be overlooked that the energy-intensive nature of these facilities can result in significant operating costs as well. A number of critical decisions during the process of designing a VLF cleanroom can pay enormous dividends over the life of the facility. A knowledgeable designer and a disciplined owner/user represent the best team to optimize system efficiency and thereby reap those dividends.

> ***"A number of critical decisions during the process of designing a VLF cleanroom can pay enormous dividends over the life of the facility."***

REFERENCES

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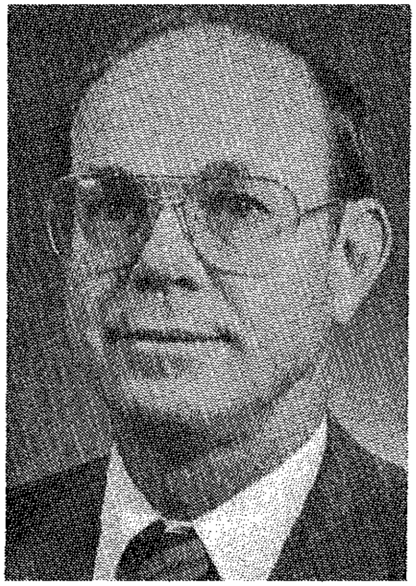

THE AUTHOR

Fred T. Gerbig, PE, is a staff member of Technology Management, Inc., the cleanroom construction management and general contracting subsidiary of Adolfson & Peterson, Inc. He also is founder of Gerbig Engineering Company and holds bachelor of science degrees in electrical and mechanical engineering.

His articles on cleanrooms and cleanroom testing have been published in trade journals and magazines. He is past secretary of the Institute of Environmental Sciences (IES) RP-006 committee dealing with standardized testing procedures. He is the former secretary of IES RP-50 for rewriting Federal Standard 209B and General Chairman of all IES committees relating to cleanrooms. In 1986, he was awarded the "James R. Milton Award" for contributions to the RP-50 committee.