FLORIDA SOLAR ENERGY CENTER®

CONTRACT REPORT

Impacts of Varying Filter Pressure Drop on Energy Consumption in Typical AC Systems in Residential and Small Commercial Buildings

Part-1 High Efficiency Variable Speed System

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Prepared for:
Etatech Inc

Prepared by:

Nabil Nassif, Ph.D.

Florida Solar Energy Center
1679 Clearlake Road
Cocoa, Florida 32922

1679 Clearlake Road, Cocoa, FL 32922-5703 • Phone: 321-638-1000 • Fax: 321-638-1010
www.fsec.ucf.edu

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Energy impacts of varying filter pressure drop in typical AC systems in residential and small commercial buildings.

Part-1: high efficiency variable speed system

Abstract

Filters are used in HVAC systems for both commercial and residential buildings to protect the equipment and improve indoor air quality in conditioned spaces. Although there are benefits of using an air filter in an air conditioning system, the resistance associated with it can increase fan energy use and may adversely affect air conditioning system performance and efficiency. A whole building simulation model using EnergyGauge® along with a detailed fan model were used to simulate the annual fan energy consumption for various air conditioning system capacities under different levels of filter cleanliness and for various filter minimum efficiency reporting values (MERV).
The simulation results showed that the energy consumption in fan increases considerably as the filters get dirty over time. The annual fan energy consumption and associated energy cost with very dirty filter of MERV4 rating can increase in the range of 8 - 20% of that for a clean MERV4 filter. It becomes 15-40% higher for MERV 7 filter, 20-40% for MERV 10, and 30-70% for MERV 13. The increase in fan energy cost due to a dirty filter depends mainly on air system capacities, filter MERV ratings, and the degree of the filter cleanliness. Considering the manufacture's data of a typical 5-ton system and MERV 7, the annual fan energy consumption increases 440 kWh with dirty filter and 873 kWh with very dirty filter, an increase of 14.8% and 29.5 % of fan energy use, respectively. This represents extra annual energy cost of $52.75 and $104.73 for dirty and very dirty filters based on 12 cent per kWh. The cost can reach to $127.94 and $249.94 if MERV 13 is used instead. These simulations indicated that the air filter can be an important source of energy waste if the dirty filter is not replaced optimally over time.

Introduction

Filters are typically used in both commercial and residential building systems. They are located in the main airstream to protect the equipment and improve indoor air quality in conditioned spaces [1-2]. Without filters, particles may accumulate on fans and heat exchanger coils adversely affecting heat transfer [3-6]. Although there are benefits of using an air filter in an air conditioning system, the resistance associated with it can increase fan energy use and may adversely affect air conditioning system performance and efficiency [7]. As the filter gets dirty over time, the pressure drop across it rises and causes not only an increase in fan power but a reduction in air conditioning system performance and air distribution system efficiency. With variable speed blowers, as the resistance increases due to a dirty filter, the fan speed increases, and thereby the fan power, in order to maintain the same airflow rate and meet space sensible load requirements. With a standard constant speed fan motor, as the filter gets dirty the external static pressure increases and airflow rate drops. The reduced airflow rate makes the system run longer to meet the same load requirement. In addition, the low air flow rate leads to reduce the system capacity and sensible heat ratio [8-10]. Thus, this study will investigate the effect of varying air filter pressure drop on whole system performance that includes fan and air conditioning system performance. ANSI/ASHRAE Standard 52.2-2007 [11] uses minimum reporting value MERV to rate the effectiveness of air filters, ranging from 1 to 16. The study will also consider various minimum reporting value MERVs, typically used in residential and small commercial buildings.

To estimate fan electric demand, a detailed fan model based on the dimensionless coefficients of flow, pressure head, and shaft power is used [12-14]. To evaluate the annual fan energy consumption in air conditioning systems using different air filter MERV ratings and levels of cleanliness, EnergyGauge® simulation model [15] is used together with the fan model.

The study is divided into two main parts (1) Part-I: high efficiency variable speed system and (2) Part-II: standard constant speed system. This report presents only the first part related to the variable speed system and a companion report Part-II will provide a detailed study on a standard constant speed system. As mentioned previously, a dirty filter used with a variable speed fan causes an increase in pressure drop and an increase in fan speed to maintain the same airflow rate. Thus, this report will consider the effect of the air filter on fan power only and the report is divided as follow. First, we will provide insight into the simulation model used for the fan power and annual energy use calculations. Second we will describe the impact of the air filter on the performance of a typical variable speed air system. Then, we will follow by providing detailed information on air conditioning systems and air filters most used in residential and small commercial building applications. In the last section, we will discuss the results and provide recommendations.

Impact of Filters on Variable Speed Fan Performance

The pressure drop associated with the air filter can increase fan energy use leading to an increase in total energy consumption. The pressure drop through the filter depends mainly on certain factors such as airstream velocity, filter type, MERV rating, area, and cleanliness. Figure 1 shows the static pressure distribution for a typical system with assumptions of three different levels of cleanliness (e.g. clean, dirty, and very dirty). As the filter becomes dirty or very dirty, the variable speed fan will react to operate at an increased speed because of higher through-filter resistance in order to maintain the same airflow rate. Figure 2 shows fan and system curves for clean, dirty, and very dirty filters. The point 1 represents the conditions when the filter is clean and the fan operates at a speed of n_1 and uses power of hp_1 to maintain for instance, an airflow rate of 60% of the design value. However, the point 2 represents the conditions when the filter is somewhat dirty and the fan operates at a higher speed of n_2 and consumes more power (hp_2) to maintain the same airflow rate. Similarly, the point 3 repents the condition when the filter is very dirty. As the filter becomes dirty (moving from point 1 to point 2 then to point 3), the fan speed increases from n_1 to n_2 and then to n_3 and the fan power from hp_1 to hp_2 and then to hp_3 to keep the same airflow rate. As the airflow rate varies in variable speed systems, the potential operating points are then located on system curves shown in Figure 2.

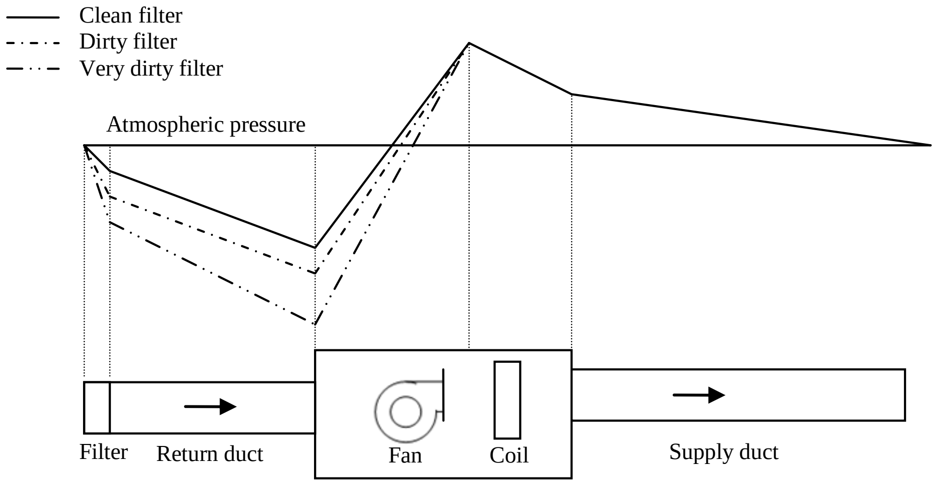

Figure 1. Static pressure distribution for a typical system with different levels of cleanliness

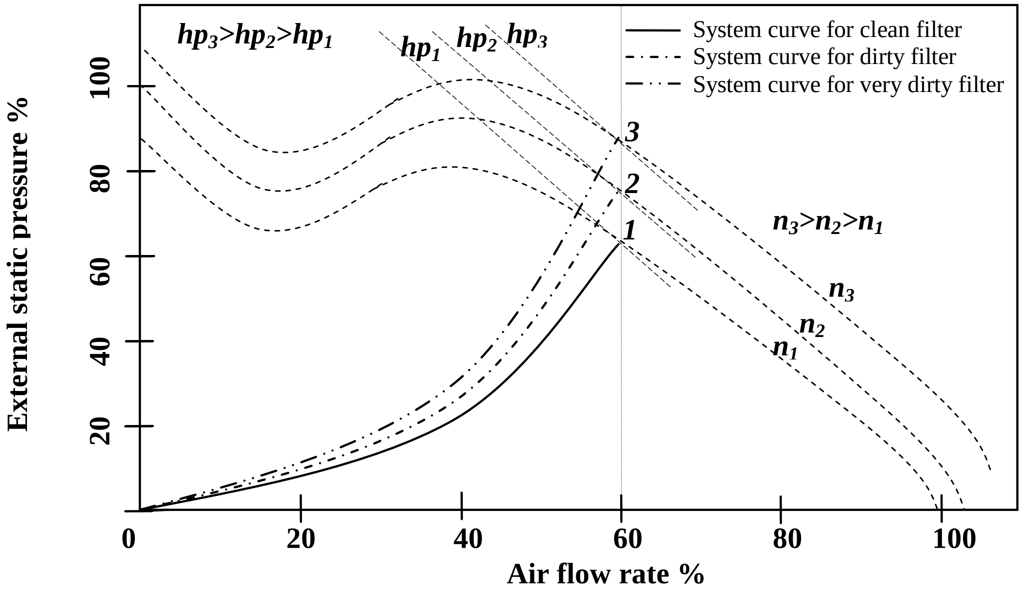

Figure 2. Fan and system curves for different levels of cleanliness

AC and Filters Description

Data for gas heating/electric cooling single packages collected from two different manufactures are used with indoor air blowers designed for low heat units, as typically in Florida. Various air conditioning system capacities, ranging from 2 to 5 tons, are used as shown in Table 1. Figure 3 and Figure 4 show manufacture's data for the 2-ton system fan used in this study. The data includes fan speed, external static pressure (without filter installed), break horsepower, and airflow rate. The fan speed contours are also illustrated in Figure 3. The design data for external static pressure (not including the filter) and horsepower are selected based on the wide field data collected by Parker et. al. [11]. Although three design options of flow rate per ton were examined (350 cfm/ton, 400 cfm/ton, and 450 cfm/ton), with only the one option of 400 cfm/ton shown and discussed in this report as there is not much difference in the results using an assumption of fixed air velocity through the filter under those different design options.

Table 1. Design information of considered air conditioning systems

	Design option	flow	rpm	hp	External static pressure in.wg. (no filter)
2 ton	400 cfm/ton	800	570	0.14	0.4
3 ton	400 cfm/ton	1200	724	0.33	0.6
4 ton	400 cfm/ton	1600	854	0.6	0.8
5 ton	400 cfm/ton	2000	1009	1.05	1

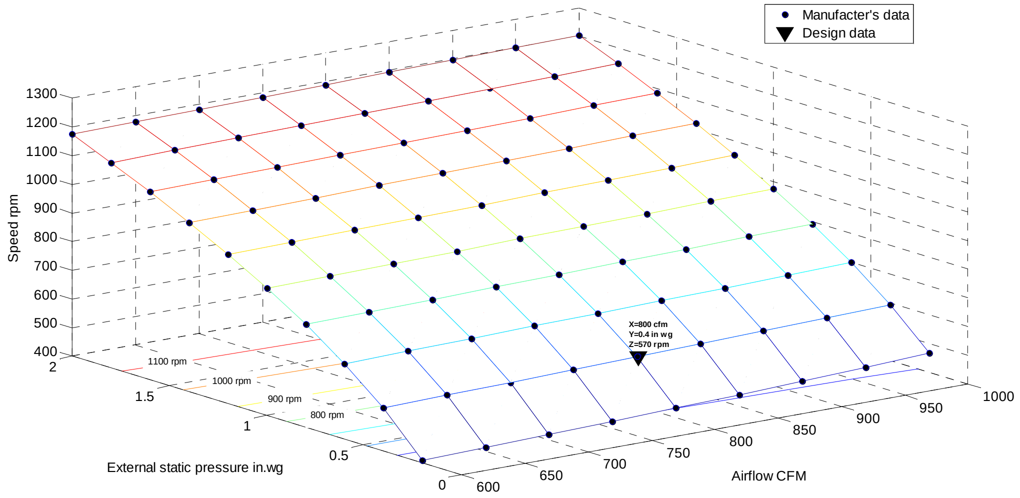

Figure 3. Fan speed contour plot (beneath the mesh)

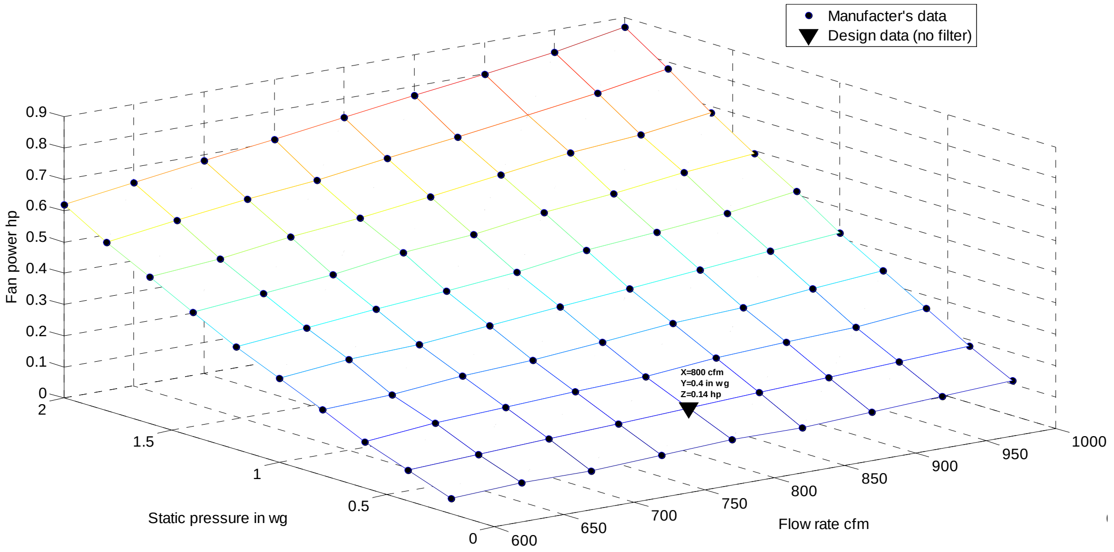

Figure 4. Fan power, static pressure, and airflow rate data

ANSI/ASHRAE Standard 52.2-2007 [11] uses the minimum reporting value MERV to rate the effectiveness of air filters. The scale is designed to represent the worst case performance of a filter when dealing with particles in the range of 0.3 to 10 micrometers. The MERV rating is from 1 to 16. Higher MERV ratings correspond to a greater percentage of particles captured on each pass, with a MERV 16 filter capturing more than 95% of particles over the full range. According to Table 12-1 of the Standard 52-2-2007, the Standard recommends a minimal change in resistance that should be at least twice the initial resistance through filters. The filter is assumed to be clean when the filter resistance is equal to the initial value in Table 12-1 of Standard 52 and to be very dirty when the filter resistance is equal to the final resistance value. The filter is then assumed to be dirty in the middle of that range. Table 2 shows the filter resistance (pressure drop across the filter) used in this study.

Table 2. Filter resistance for different levels of cleanliness and MERV ratings.

	Filter pressure drop (in. wg)		
	Clean	**Dirty**	**Very dirty**
MERV 4	0.1	0.2	0.3
MERV 7	0.2	0.4	0.6
MERV 10	0.3	0.65	1
MERV 13	0.4	0.9	1.4

Several field testing have been done by air filter manufactures on several types and size of filters (http://texairfilters.com/news/testsonpressuredrop.htm). The test showed that the pressure drop through clean filters of MERV 7, MERV10, and MERV13 and with different air velocities and filter depths is in the range of 0.17 - 0.42 in. wg. These results are consistent with the data for the clean filter provided in Table 2. Table 3 shows the external static pressures considered in this study. Those external static pressures based on design airflow rate conditions are obtained by adding the pressure drop through the filter assumed in Table 2 with external static pressure without filter shown in Table 1. It is worth noting that the airflow rate varies with loads and the corresponding external static pressures with low airflow rates are determined based on parabolic system curves shown in Figure 2.

Table 3. External static pressures considered in this study

	External Static Pressure (in. wg)												
	No filter	**MERV 4**			**MERV 7**			**MERV 10**			**MERV 13**		
	in. wg	**clean**	**dirty**	**very dirty**	**clean**	**dirty**	**very dirty**	**clean**	**dirty**	**very dirty**	**clean**	**dirty**	**very dirty**
2 ton	0.4	0.5	0.6	0.7	0.6	0.8	1	0.7	1.05	1.4	0.8	1.3	1.8
3 ton	0.6	0.7	0.8	0.9	0.8	1	1.2	0.9	1.25	1.6	1	1.5	2
4 ton	0.8	0.9	1	1.1	1	1.2	1.4	1.1	1.45	1.8	1.2	1.7	2.2
5 ton	1	1.1	1.2	1.3	1.2	1.4	1.6	1.3	1.65	2	1.4	1.9	2.4

Simulation Model

To estimate fan electric demand, a detailed steady state fan model introduced by Clark [12] and also described in HVAC toolkit [13] is used. The model is based on the dimensionless coefficients of flow (Φ), pressure head (Ψ), and shaft power (η_f) as the following.

$$\Phi = \frac{\dot{Q}}{N \cdot d^3} \tag{1}$$

$$\Psi = \frac{P_S}{\rho \cdot N^2 \cdot d^2} \tag{2}$$

$$\eta_f = \frac{\dot{Q} \cdot P_S}{\dot{W}_s} \tag{3}$$

Where the term (d) is fan diameter and (ρ) is air density. The Q, Ps, and N are fan airflow rate, total static pressure, and speed, respectively. The performance of a fan is represented by a polynomial regression to the manufacturer's data using these dimensionless coefficients.

$$\Psi = a_0 + a_1\Phi + a_2\Phi^2 + a_3\Phi^3 + a_4\Phi^4 \tag{4}$$

$$\eta_f = b_0 + b_1\Phi + b_2\Phi^2 + b_3\Phi^3 + b_4\Phi^4 \tag{5}$$

The coefficients a_i , b_i are determined from the manufacturer's data (such as in Figures 3 and 4) . The fan power ($\dot{W}_t$) is determined using the shaft power ($\dot{W}_s$) and the motor efficiency (η_m):

$$\dot{W}_t = \frac{\dot{W}_s}{\eta_m} \tag{6}$$

The fan model has been used and validated in several studies [16-18].

To evaluate the annual energy consumption in air conditioning systems with different levels of cleanliness and filter MERV ratings, a whole building energy simulation model is used together with the fan model. Figure 5 shows the flow chart of the fan energy calculations used in this study. A simulation model for a residential building, using EnergyGauge® USA, is used to generate the number of hours within each part-load (i.e., ratio of airflow rate to design airflow) fraction range as a function of different system capacities, building characteristics, and locations. The study is limited to Orlando weather conditions and typical home characteristics as shown in Appendix A. Appendix B shows the number of hours within each part-load for 2-ton system. EnergyGauge® USA is a user-friendly home energy simulation software tool designed specifically for the easy and accurate evaluation of home energy-efficiency. The software uses DOE 2.1-E hourly building energy simulation software to simulate energy use.

The airflow rate data from EnergyGauge® is then fed to the fan model to estimate the fan power. Annual energy use is then calculated by the sum of the product of the fan power and number of hours within each airflow rate fraction range as a follow:

$$FanEnergy = \sum \dot{W}_t \cdot Nh \qquad (7)$$

Where the *Nh* is the number of hours within each airflow rate fraction range.

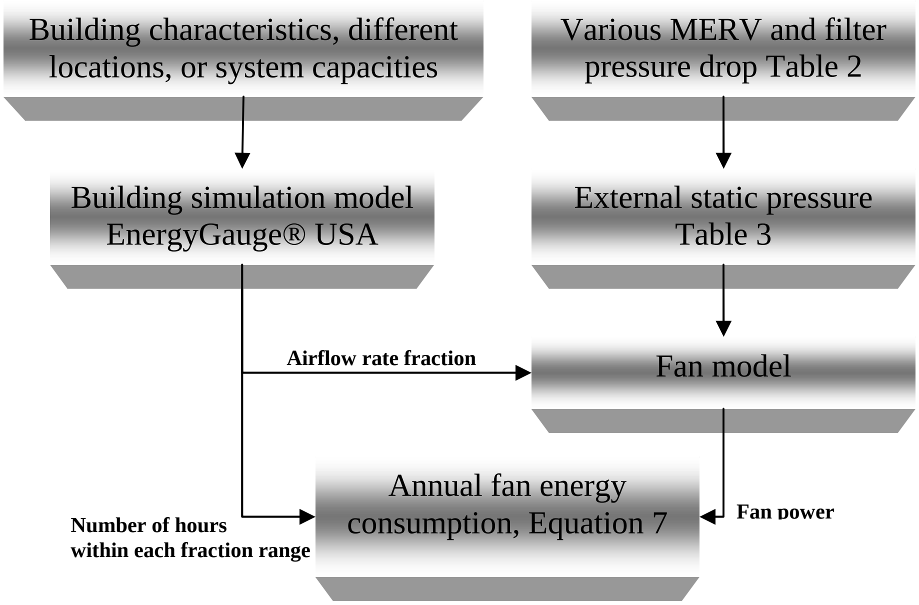

Figure 5. Flowchart of the fan energy calculations

Results and Discussion

The simulation model was used to investigate the impacts of varying air filter resistance on energy consumption of different air conditioning system capacities and various MERV ratings of filters. It is worth noting that MERV 4 and MERV 8 filters are usually used in standard and better residential systems as shown in Table E-1 of ASHRAE Standard 52.2-2007 [11]. The fan performance data is collected from two different manufacturers and for four system capacities of 2, 3, 4, and 5 tons. The data includes fan speed, external static pressure (without filter installed), break horsepower, and airflow rate. Figure 6 shows the fan and system performance curves with no filter, clean, dirty, and very dirty filters for the 2-ton system and MERV 10. To provide the airflow rate required for a specific load or airflow rate (e.g. 800 cfm), the fan needs to operate at an elevated speed as the filter gets dirty. For instant, when the MERV 10 filter becomes very dirty, the external pressure increases from 0.7 in. wg to 1.4 in. wg and consequently the fan speed rises from 727 rpm to 1004 rpm and associated fan power from 0.24 hp to 0.49 hp in order to provide the same airflow rate of 800 cfm. With variable speed fans, the airflow rate will be varied depending on the space load. The operating points will be then located on the system curves shown in Figure 6 such as “system curve-clean filter” for clean filter or “system curve-dirty filter” for dirty filter.

Figures 7 and 8 show the external static pressure and associated fan power for different system capacities and filter ratings and cleanliness. The external static pressures shown in those figures are based on design airflow rates. The fan power is calculated by the fan model described above using the external pressure and airflow rate as inputs. It is clear that the external static pressure and associated fan power increases from no filter to clean filter then to very dirty filter and from MERV 4 to MERV 7 and then to MERV 13. As an example, for MERV 13 and 5-ton system, the external static pressures are 1, 1.4, 1.9, and 2.4 in.wg., the associated fan powers are 0.79, 0.98, 1.25, and 1.54 kW, for no filter, clean, dirty, and very dirty filters, respectively. For a 5-ton system using a clean filter, the external static pressures are 1.1, 1.2, 1.3, and 1.4 in.wg., the associated fan power are 0.84, 0.89, 0.93, and 0.98 kW for MERV 4, MERV 7, MERV 10, MERV 13, respectively.

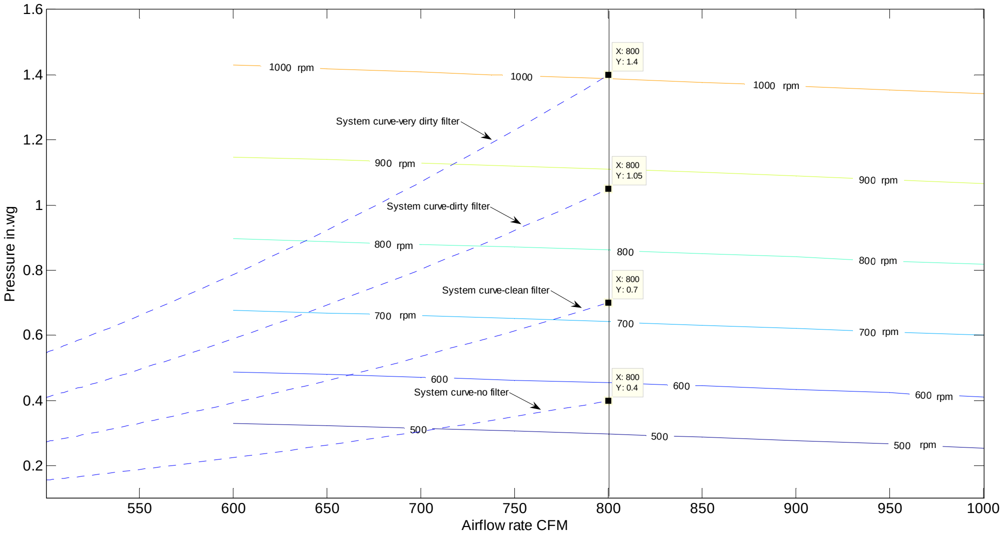

Figure 6. Fan and system performance curves with no filter, clean, dirty, and very dirty filters for the 2-ton system and MERV 10

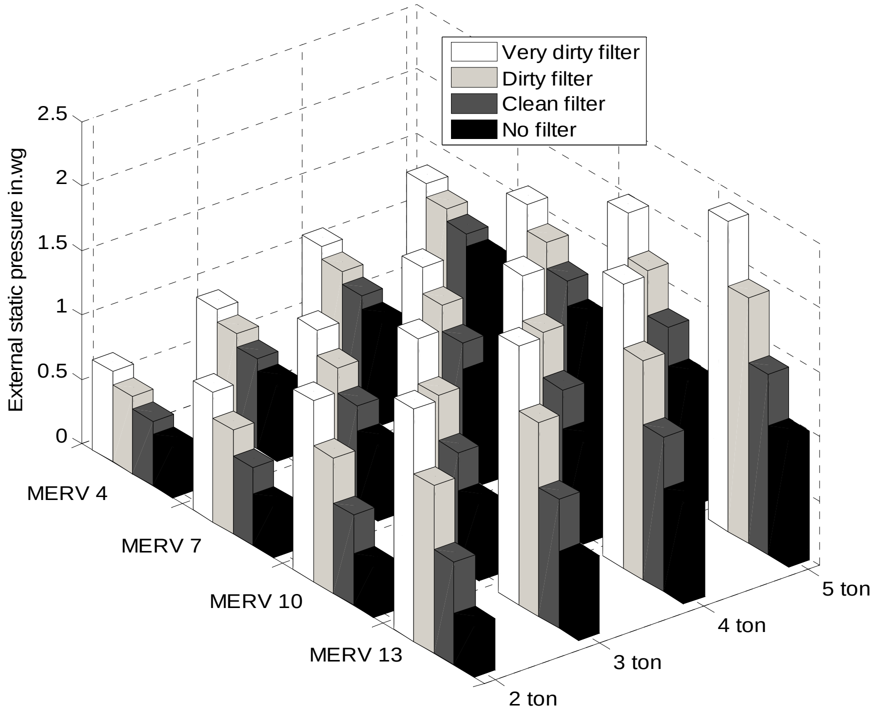

Figure 7. External static pressures for different system capacities and filter ratings and cleanliness

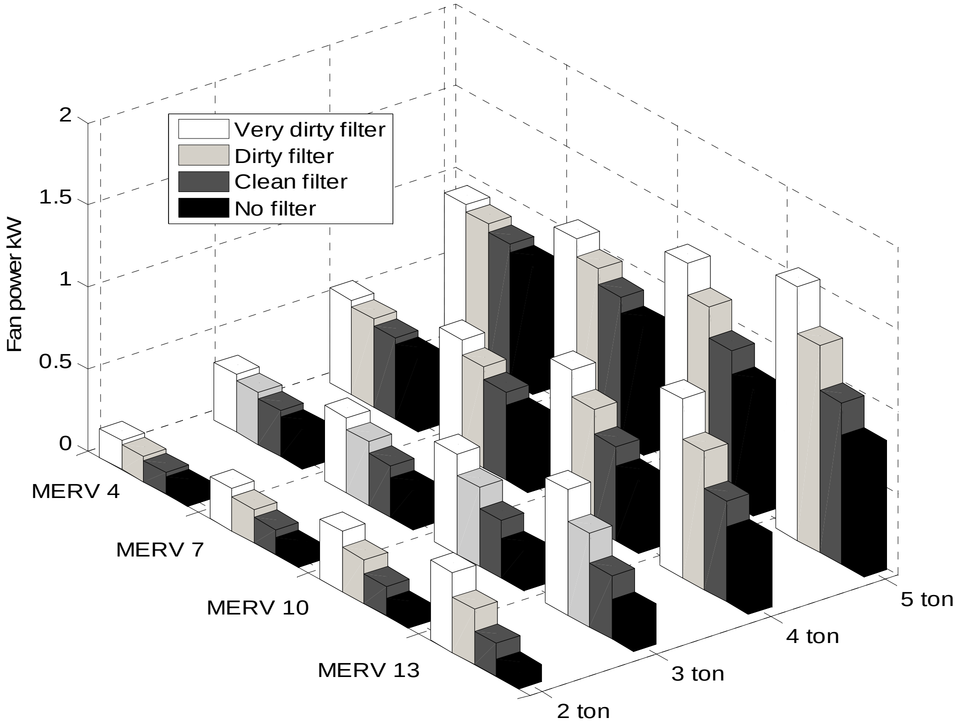

Figure 8. Fan power for different system capacities and filter ratings and cleanliness

The results above are based on design airflow rates. However, in variable speed system, airflow varies with loads. The EnergyGauge® simulation model is then used to determine the number of hours within each part-load range (airflow rate fraction range). The fan model calculates the fan power at various values of flow rate and pressure and then the outputs are multiplied by the number of hours to get annual fan energy use. Figure 9 shows the annual fan energy consumption for different system capacities, filter ratings and cleanliness. The annual fan energy consumption increases as the filter gets dirty and with higher filter MERV ratings. For MERV 7 and 5-ton system, the annual fan energy consumptions are 2510, 2961, 3401, and 3834 kWh for no filter, clean, dirty, and very dirty filters, respectively. This result indicates that the annual fan energy consumption increases 440 kWh with dirty filter and 873 kWh with very dirty filter as compared to that for a clean filter, an increase of 14.8% and 29.5 %, respectively. The extra fan energy consumptions associated with employing dirty or very dirty filters instead of using a clean filter are illustrated in Figure 10. Based on 12 cent per kWh, the extra fan energy cost associated with employing dirty or very dirty filters are shown in Table 4 and Table 5. As a result, the annual fan energy consumption and associated energy cost with very dirty MERV4 filter (typically used in residential building) can be 8 - 20% greater than that for clean MERV4 filter. It becomes 15-40%, 20-40%, and 30-70% higher in the case of MERV 7, MERV 10, and MERV 13 filters, respectively. Based on the investigated system characteristics and data used in this report, the annual fan energy cost is $327.80 when clean MERV 4 filter of 5-ton system is used and it becomes $382.37 when the MERV 4 filter gets very dirty, an increase of $54.57 (see table below). The results indicate that the dirty filter can cause a significant waste in fan energy and show the necessity of replacing the filter frequently over time.

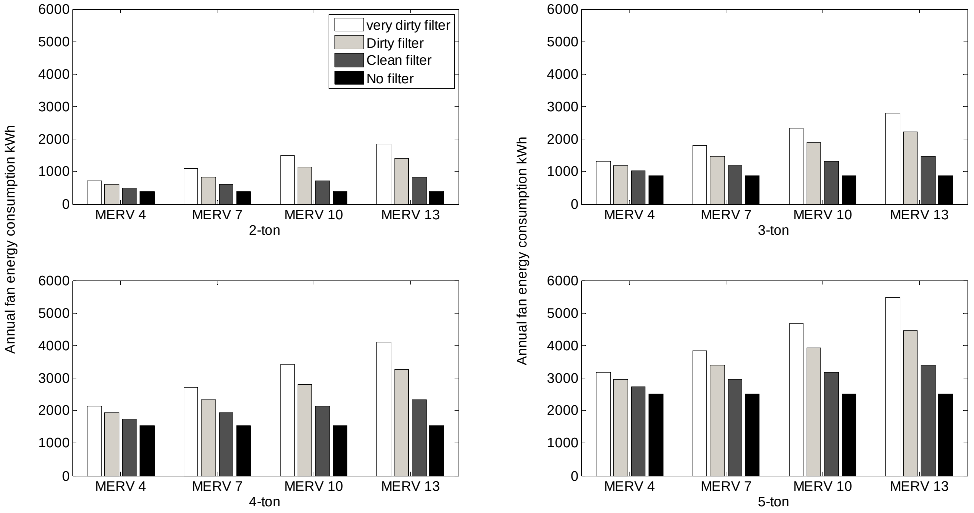

Figure 9. Annual fan energy consumptions for different system capacities and filter ratings and cleanliness

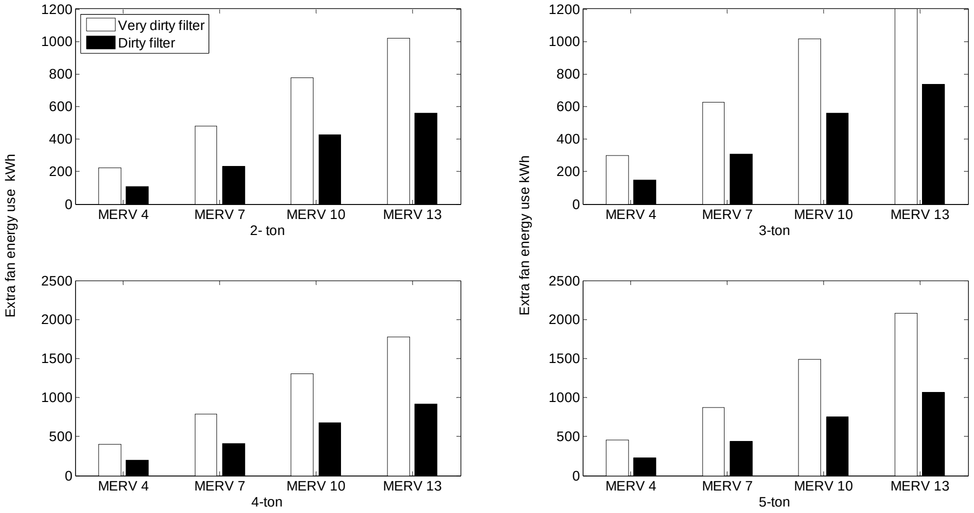

Figure 10. The extra fan energy consumptions associated with employing dirty or very dirty filters (instead of clean filters)

Table 4. Annual fan energy cost and additional costs associated with employing dirty or very dirty filters instead of clean filters

		Annual cost $	**Annual cost $**	**Annual cost $**	**Additional Cost $**	**Additional Cost $**	**Additional Cost %**	**Additional Cost %**
		Clear	Dirty	Very Dirty	Dirty	Very Dirty	Dirty	Very Dirty
2 ton	MERV4	59.60	72.60	86.23	13.00	26.63	21.8%	44.7%
	MERV7	72.60	100.42	130.39	27.82	57.79	38.3%	79.6%
	MERV10	86.23	137.29	179.56	51.06	93.32	59.2%	108.2%
	MERV12	100.42	167.60	222.84	67.19	122.42	66.9%	121.9%
3 ton	MERV4	123.17	140.92	159.08	17.75	35.92	14.4%	29.2%
	MERV7	140.92	177.74	216.32	36.83	75.41	26.1%	53.5%
	MERV10	159.08	226.22	281.14	67.14	122.05	42.2%	76.7%
	MERV12	177.74	265.97	336.66	88.22	158.92	49.6%	89.4%
4 ton	MERV4	207.82	231.17	255.46	23.35	47.64	11.2%	22.9%
	MERV7	231.17	280.06	325.94	48.89	94.78	21.1%	41.0%
	MERV10	255.46	336.84	411.76	81.38	156.30	31.9%	61.2%
	MERV12	280.06	390.73	492.71	110.68	212.65	39.5%	75.9%
5 ton	MERV4	327.79	355.33	382.37	27.54	54.58	8.4%	16.6%
	MERV7	355.33	408.08	460.07	52.75	104.74	14.8%	29.5%
	MERV10	382.37	472.91	560.95	90.54	178.58	23.7%	46.7%
	MERV12	408.08	536.03	658.02	127.94	249.94	31.4%	61.2%

Table 5. Annual energy cost (HVAC) and additional costs associated with employing dirty or very dirty filters instead of clean filters

		Annual cost $	**Annual cost $**	**Annual cost $**	**Additional Cost $**	**Additional Cost $**	**Additional Cost %**	**Additional Cost %**
		Clear	Dirty	Very Dirty	Dirty	Very Dirty	Dirty	Very Dirty
2 ton	MERV4	539.60	552.60	566.23	13.00	26.63	2.4%	4.9%
	MERV7	552.60	580.42	610.39	27.82	57.79	5.0%	10.5%
	MERV10	566.23	617.29	659.56	51.06	93.32	9.0%	16.5%
	MERV12	580.42	647.60	702.84	67.19	122.42	11.6%	21.1%
3 ton	MERV4	603.17	620.92	639.08	17.75	35.92	2.9%	6.0%
	MERV7	620.92	657.74	696.32	36.83	75.41	5.9%	12.1%
	MERV10	639.08	706.22	761.14	67.14	122.05	10.5%	19.1%
	MERV12	657.74	745.97	816.66	88.22	158.92	13.4%	24.2%
4 ton	MERV4	687.82	711.17	735.46	23.35	47.64	3.4%	6.9%
	MERV7	711.17	760.06	805.94	48.89	94.78	6.9%	13.3%
	MERV10	735.46	816.84	891.76	81.38	156.30	11.1%	21.3%
	MERV12	760.06	870.73	972.71	110.68	212.65	14.6%	28.0%
5 ton	MERV4	807.79	835.33	862.37	27.54	54.58	3.4%	6.8%
	MERV7	835.33	888.08	940.07	52.75	104.74	6.3%	12.5%
	MERV10	862.37	952.91	1,040.95	90.54	178.58	10.5%	20.7%
	MERV12	888.08	1,016.03	1,138.02	127.94	249.94	14.4%	28.1%

Conclusion

The whole building energy simulation model used EnergyGauge® in conjunction with a detailed fan model to investigate the impacts of filter cleanliness on fan energy consumption in high efficiency variable-speed air conditioning systems. The study takes into consideration the different MERV filters and system capacities, ranging from 2 to 5 tons and filter ratings from 4 to 13. The simulation results showed that the fan energy consumption increases considerably as the filters get dirty over time. The annual fan energy consumption and associated energy cost increases with very dirty filter by approximately 8 - 20% for MERV 4 filter, 15-40% for MERV 7 filter, 20-40% for MERV 10, and 30-70% for MERV 13. The increase of fan energy cost due to a dirty filter depends mainly on filter MERV ratings, air system capacities, as well as on the degree of the filter cleanliness. Considering the manufacture’s data of a typical 5-ton system and MERV 7, the annual fan energy consumption increases 440 kWh with dirty filter and 873 kWh with very dirty filter, an increase of 14.8% and 29.5 %, respectively. This represents an annual energy cost increase of $52.75 and $104.73 for dirty and very dirty filters based on 12 cent per kWh. The cost can reach to $127.94 and $249.94 if MERV 13 is used instead. These simulations indicated that the air filter can be important source of energy waste if the dirty filter is not replaced optimally over time.

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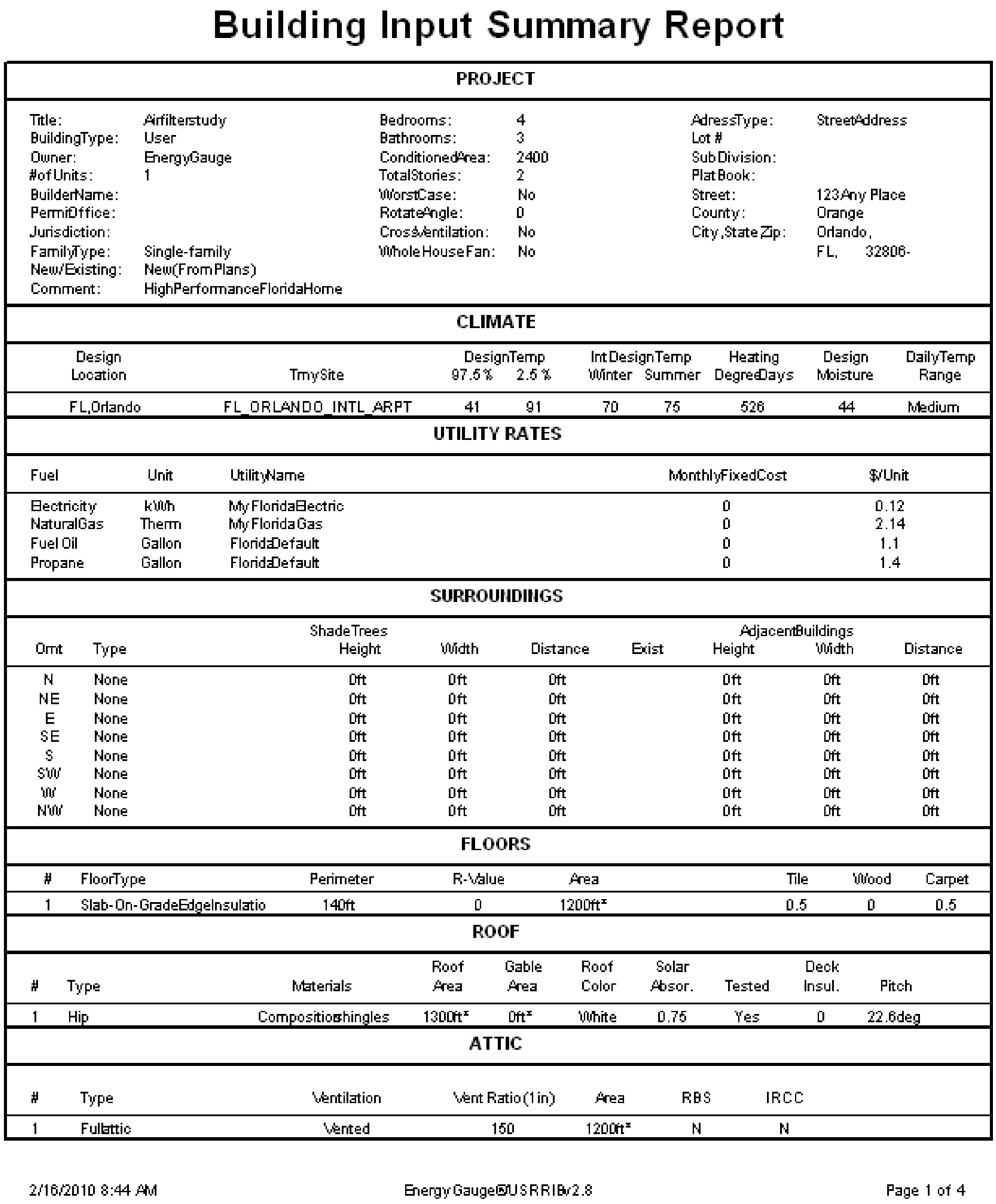
APPENDIX A



Building Input Summary Report

PROJECT

Title:	Airfilterstudy	Bedrooms:	4	AdressType:	StreetAddress
BuildingType:	User	Bathrooms:	3	Lot #	
Owner:	EnergyGauge	ConditionedArea:	2400	SubDivision:	
#ofUnits:	1	TotalStories:	2	PlatBook:	
BuilderName:		WorstCase:	No	Street:	123Any Place
PermitOffice:		RotateAngle:	0	County:	Orange
Jurisdiction:		CrossVentilation:	No	City,State,Zip:	Orlando,
FamilyType:	Single-family	WholeHouseFan:	No		FL, 32806-
New/Existing:	New(FromPlans)				
Comment:	HighPerformanceFloridaHome				

CLIMATE

Design Location	TmySite	DesignTemp 97.5%	2.5%	IntDesignTemp Winter	Summer	Heating DegreeDays	Design Moisture	DailyTemp Range
FL,Orlando	FL_ORLANDO_INTL_ARPT	41	91	70	75	526	44	Medium

UTILITY RATES

Fuel	Unit	UtilityName	MonthlyFixedCost	$/Unit
Electricity	kWh	MyFloridaElectric	0	0.12
NaturalGas	Therm	MyFloridaGas	0	2.14
Fuel Oil	Gallon	FloridaDefault	0	1.1
Propane	Gallon	FloridaDefault	0	1.4

SURROUNDINGS

		ShadeTrees				AdjacentBuildings		
Ornt	Type	Height	Width	Distance	Exist	Height	Width	Distance
N	None	0ft	0ft	0ft		0ft	0ft	0ft
NE	None	0ft	0ft	0ft		0ft	0ft	0ft
E	None	0ft	0ft	0ft		0ft	0ft	0ft
SE	None	0ft	0ft	0ft		0ft	0ft	0ft
S	None	0ft	0ft	0ft		0ft	0ft	0ft
SW	None	0ft	0ft	0ft		0ft	0ft	0ft
W	None	0ft	0ft	0ft		0ft	0ft	0ft
NW	None	0ft	0ft	0ft		0ft	0ft	0ft

FLOORS

#	FloorType	Perimeter	R-Value	Area	Tile	Wood	Carpet
1	Slab-On-GradeEdgeInsulatio	140ft	0	1200ft^2	0.5	0	0.5

ROOF

#	Type	Materials	Roof Area	Gable Area	Roof Color	Solar Absor.	Tested	Deck Insul.	Pitch
1	Hip	Compositionshingles	1300ft^2	0ft^2	White	0.75	Yes	0	22.6deg

ATTIC

#	Type	Ventilation	Vent Ratio(1in)	Area	RBS	IRCC
1	Fullattic	Vented	150	1200ft^2	N	N

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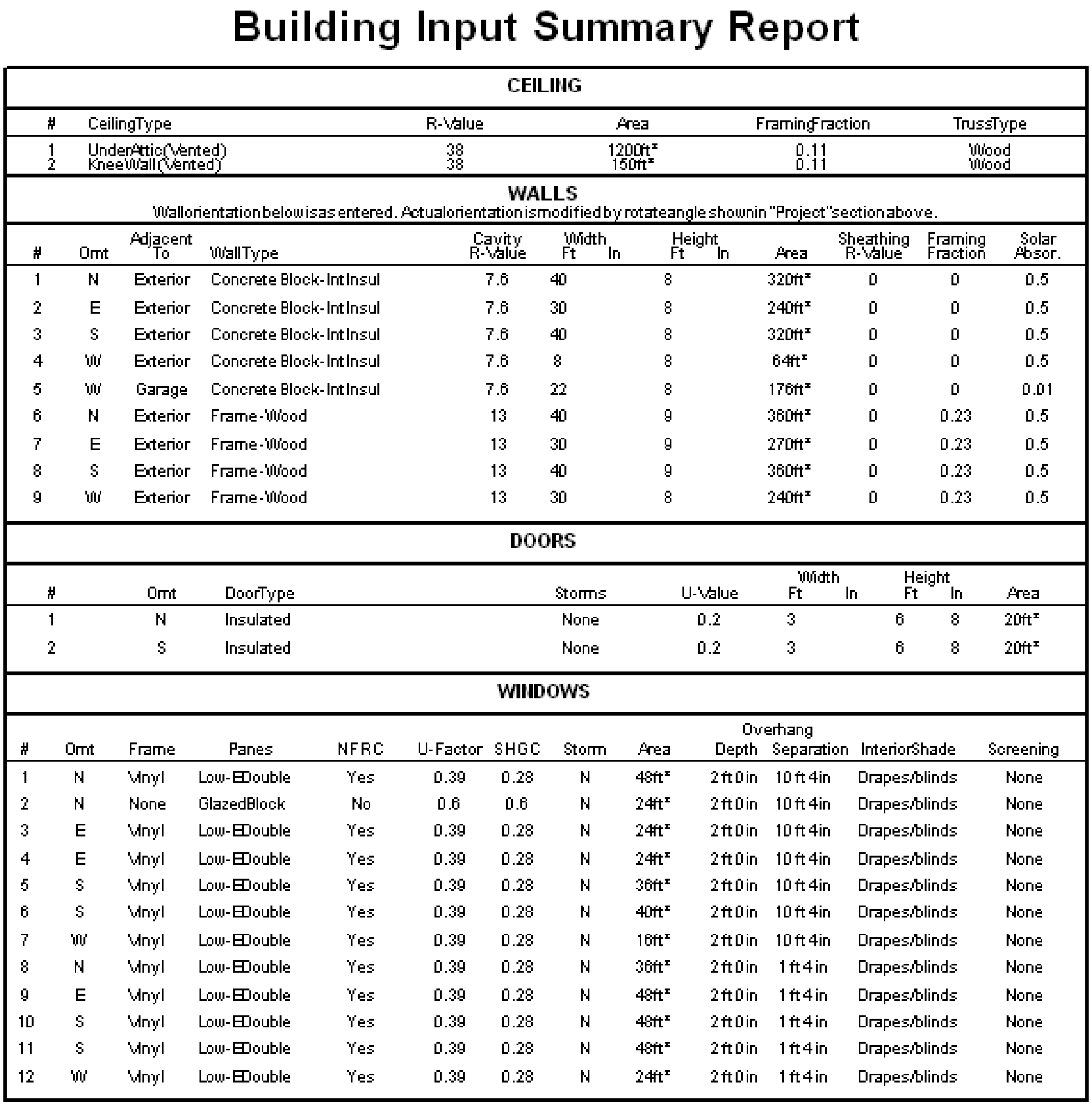


Building Input Summary Report

CEILING

#	CeilingType	R-Value	Area	FramingFraction	TrussType
1	UnderAttic(Vented)	38	1200ft²	0.11	Wood
2	KneeWall(Vented)	38	150ft²	0.11	Wood

WALLS

Wallorientation belowisas entered. Actualorientation ismodified by rotateangle shownin "Project" section above.

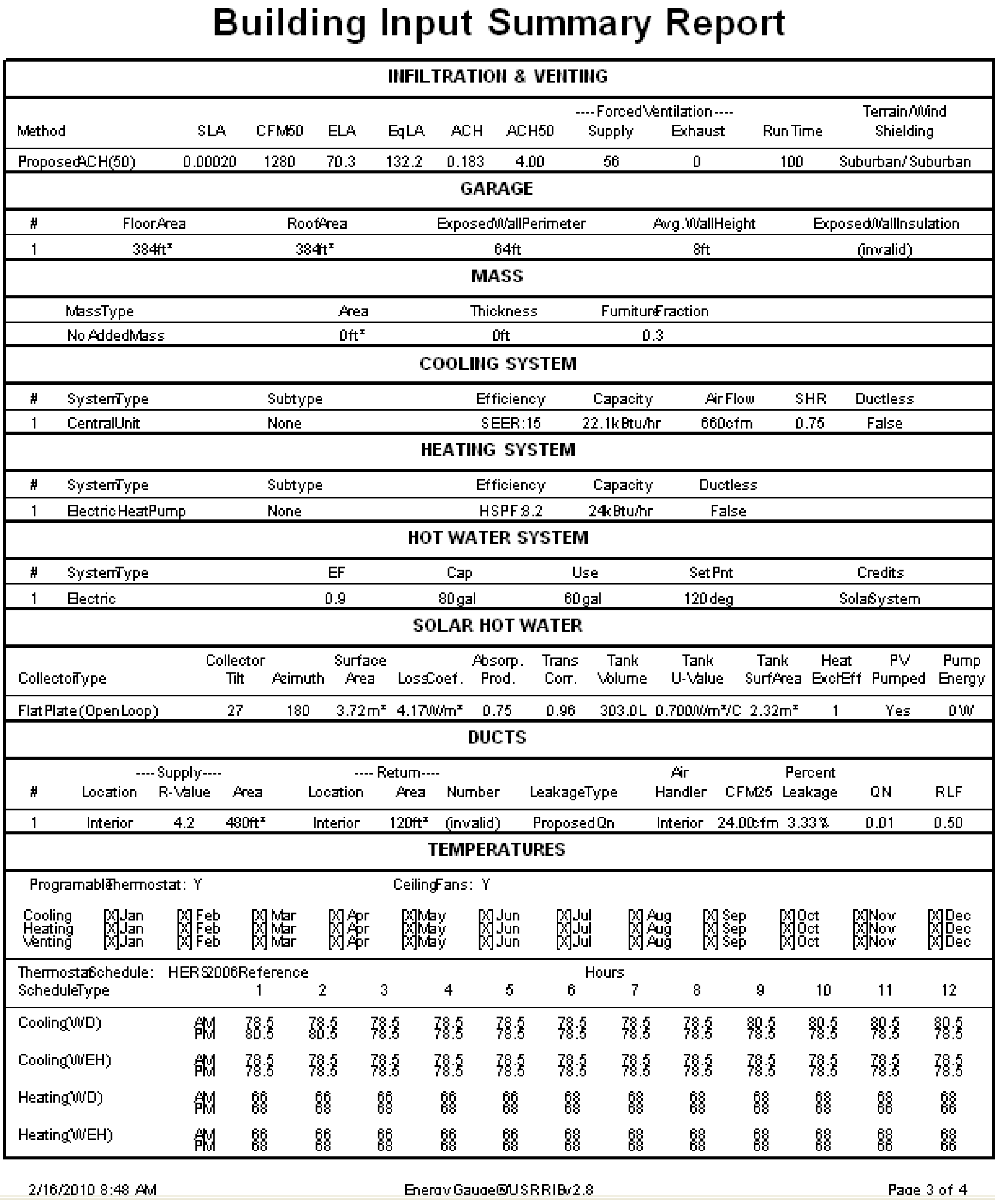
#	Ornt	Adjacent To	WallType	Cavity R-Value	Width Ft	Width In	Height Ft	Height In	Area	Sheathing R-Value	Framing Fraction	Solar Absor.
1	N	Exterior	Concrete Block-Int Insul	7.6	40		8		320ft²	0	0	0.5
2	E	Exterior	Concrete Block-Int Insul	7.6	30		8		240ft²	0	0	0.5
3	S	Exterior	Concrete Block-Int Insul	7.6	40		8		320ft²	0	0	0.5
4	W	Exterior	Concrete Block-Int Insul	7.6	8		8		64ft²	0	0	0.5
5	W	Garage	Concrete Block-Int Insul	7.6	22		8		176ft²	0	0	0.01
6	N	Exterior	Frame-Wood	13	40		9		360ft²	0	0.23	0.5
7	E	Exterior	Frame-Wood	13	30		9		270ft²	0	0.23	0.5
8	S	Exterior	Frame-Wood	13	40		9		360ft²	0	0.23	0.5
9	W	Exterior	Frame-Wood	13	30		8		240ft²	0	0.23	0.5

DOORS

#	Ornt	DoorType	Storms	U-Value	Width Ft	Width In	Height Ft	Height In	Area
1	N	Insulated	None	0.2	3		6	8	20ft²
2	S	Insulated	None	0.2	3		6	8	20ft²

WINDOWS

#	Ornt	Frame	Panes	NFRC	U-Factor	SHGC	Storm	Area	Overhang Depth	Overhang Separation	InteriorShade	Screening
1	N	Vinyl	Low-E Double	Yes	0.39	0.28	N	48ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
2	N	None	GlazedBlock	No	0.6	0.6	N	24ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
3	E	Vinyl	Low-E Double	Yes	0.39	0.28	N	24ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
4	E	Vinyl	Low-E Double	Yes	0.39	0.28	N	24ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
5	S	Vinyl	Low-E Double	Yes	0.39	0.28	N	36ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
6	S	Vinyl	Low-E Double	Yes	0.39	0.28	N	40ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
7	W	Vinyl	Low-E Double	Yes	0.39	0.28	N	16ft²	2 ft 0 in	10 ft 4 in	Drapes/blinds	None
8	N	Vinyl	Low-E Double	Yes	0.39	0.28	N	36ft²	2 ft 0 in	1 ft 4 in	Drapes/blinds	None
9	E	Vinyl	Low-E Double	Yes	0.39	0.28	N	48ft²	2 ft 0 in	1 ft 4 in	Drapes/blinds	None
10	S	Vinyl	Low-E Double	Yes	0.39	0.28	N	48ft²	2 ft 0 in	1 ft 4 in	Drapes/blinds	None
11	S	Vinyl	Low-E Double	Yes	0.39	0.28	N	48ft²	2 ft 0 in	1 ft 4 in	Drapes/blinds	None
12	W	Vinyl	Low-E Double	Yes	0.39	0.28	N	24ft²	2 ft 0 in	1 ft 4 in	Drapes/blinds	None



Building Input Summary Report

INFILTRATION & VENTING

Method	SLA	CFM50	ELA	EqLA	ACH	ACH50	---- Forced Ventilation ---- Supply	Exhaust	Run Time	Terrain/Wind Shielding
Proposed ACH(50)	0.00020	1280	70.3	132.2	0.183	4.00	56	0	100	Suburban/ Suburban

GARAGE

#	Floor Area	Roof Area	Exposed Wall Perimeter	Avg. Wall Height	Exposed Wall Insulation
1	384 ft²	384 ft²	64 ft	8 ft	(invalid)

MASS

Mass Type	Area	Thickness	Furniture Fraction
No Added Mass	0 ft²	0 ft	0.3

COOLING SYSTEM

#	System Type	Subtype	Efficiency	Capacity	Air Flow	SHR	Ductless
1	Central Unit	None	SEER: 15	22.1 kBtu/hr	660 cfm	0.75	False

HEATING SYSTEM

#	System Type	Subtype	Efficiency	Capacity	Ductless
1	Electric Heat Pump	None	HSPF: 8.2	24 kBtu/hr	False

HOT WATER SYSTEM

#	System Type	EF	Cap	Use	SetPnt	Credits
1	Electric	0.9	80 gal	60 gal	120 deg	Solar System

SOLAR HOT WATER

Collector Type	Collector Tilt	Azimuth	Surface Area	Loss Coef.	Absorp. Prod.	Trans Corr.	Tank Volume	Tank U-Value	Tank Surf Area	Heat Exch Eff	PV Pumped	Pump Energy
Flat Plate (Open Loop)	27	180	3.72 m²	4.17 W/m²	0.75	0.96	303.0 L	0.700 W/m²/C	2.32 m²	1	Yes	0 W

DUCTS

#	---- Supply ---- Location	R-Value	Area	---- Return ---- Location	Area	Number	Leakage Type	Air Handler	CFM25	Percent Leakage	QN	RLF
1	Interior	4.2	480 ft²	Interior	120 ft²	(invalid)	Proposed Qn	Interior	24.0 cfm	3.33 %	0.01	0.50

TEMPERATURES

Programable Thermostat: Y　　Ceiling Fans: Y

Cooling	[X] Jan	[X] Feb	[X] Mar	[X] Apr	[X] May	[X] Jun	[X] Jul	[X] Aug	[X] Sep	[X] Oct	[X] Nov	[X] Dec
Heating	[X] Jan	[X] Feb	[X] Mar	[X] Apr	[X] May	[X] Jun	[X] Jul	[X] Aug	[X] Sep	[X] Oct	[X] Nov	[X] Dec
Venting	[X] Jan	[X] Feb	[X] Mar	[X] Apr	[X] May	[X] Jun	[X] Jul	[X] Aug	[X] Sep	[X] Oct	[X] Nov	[X] Dec

Thermostat Schedule: HERS 2006 Reference

Schedule Type		Hours 1	2	3	4	5	6	7	8	9	10	11	12
Cooling (WD)	AM	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	80.5	80.5	80.5	80.5
	PM	80.5	80.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5
Cooling (WEH)	AM	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5
	PM	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5	78.5
Heating (WD)	AM	66	66	66	66	66	68	68	68	68	68	68	68
	PM	68	68	68	68	68	68	68	68	68	68	66	66
Heating (WEH)	AM	66	66	66	66	66	68	68	68	68	68	68	68
	PM	68	68	68	68	68	68	68	68	68	68	66	66

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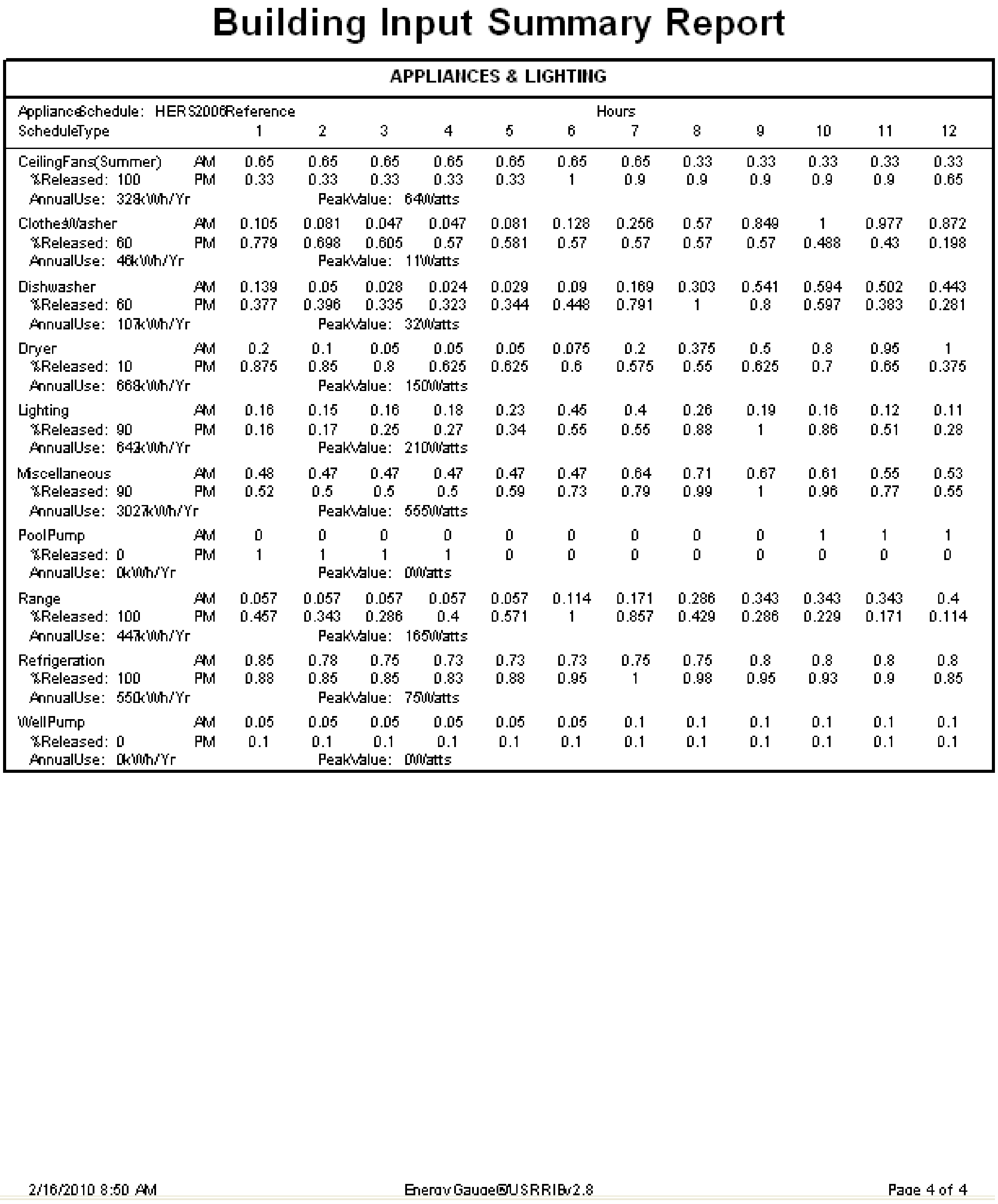

Building Input Summary Report

APPLIANCES & LIGHTING

Appliance Schedule: HERS 2006 Reference		Hours											
Schedule Type		1	2	3	4	5	6	7	8	9	10	11	12
Ceiling Fans (Summer)	AM	0.65	0.65	0.65	0.65	0.65	0.65	0.65	0.33	0.33	0.33	0.33	0.33
%Released: 100	PM	0.33	0.33	0.33	0.33	0.33	1	0.9	0.9	0.9	0.9	0.9	0.65
AnnualUse: 328 kWh/Yr		PeakValue: 64 Watts											
Clothes Washer	AM	0.105	0.081	0.047	0.047	0.081	0.128	0.256	0.57	0.849	1	0.977	0.872
%Released: 60	PM	0.779	0.698	0.605	0.57	0.581	0.57	0.57	0.57	0.57	0.488	0.43	0.198
AnnualUse: 46 kWh/Yr		PeakValue: 11 Watts											
Dishwasher	AM	0.139	0.05	0.028	0.024	0.029	0.09	0.169	0.303	0.541	0.594	0.502	0.443
%Released: 60	PM	0.377	0.396	0.335	0.323	0.344	0.448	0.791	1	0.8	0.597	0.383	0.281
AnnualUse: 107 kWh/Yr		PeakValue: 32 Watts											
Dryer	AM	0.2	0.1	0.05	0.05	0.05	0.075	0.2	0.375	0.5	0.8	0.95	1
%Released: 10	PM	0.875	0.85	0.8	0.625	0.625	0.6	0.575	0.55	0.625	0.7	0.65	0.375
AnnualUse: 668 kWh/Yr		PeakValue: 150 Watts											
Lighting	AM	0.16	0.15	0.16	0.18	0.23	0.45	0.4	0.26	0.19	0.16	0.12	0.11
%Released: 90	PM	0.16	0.17	0.25	0.27	0.34	0.55	0.55	0.88	1	0.86	0.51	0.28
AnnualUse: 642 kWh/Yr		PeakValue: 210 Watts											
Miscellaneous	AM	0.48	0.47	0.47	0.47	0.47	0.47	0.64	0.71	0.67	0.61	0.55	0.53
%Released: 90	PM	0.52	0.5	0.5	0.5	0.59	0.73	0.79	0.99	1	0.96	0.77	0.55
AnnualUse: 3027 kWh/Yr		PeakValue: 555 Watts											
Pool Pump	AM	0	0	0	0	0	0	0	0	0	1	1	1
%Released: 0	PM	1	1	1	1	0	0	0	0	0	0	0	0
AnnualUse: 0 kWh/Yr		PeakValue: 0 Watts											
Range	AM	0.057	0.057	0.057	0.057	0.057	0.114	0.171	0.286	0.343	0.343	0.343	0.4
%Released: 100	PM	0.457	0.343	0.286	0.4	0.571	1	0.857	0.429	0.286	0.229	0.171	0.114
AnnualUse: 447 kWh/Yr		PeakValue: 165 Watts											
Refrigeration	AM	0.85	0.78	0.75	0.73	0.73	0.73	0.75	0.75	0.8	0.8	0.8	0.8
%Released: 100	PM	0.88	0.85	0.85	0.83	0.88	0.95	1	0.98	0.95	0.93	0.9	0.85
AnnualUse: 550 kWh/Yr		PeakValue: 75 Watts											
Well Pump	AM	0.05	0.05	0.05	0.05	0.05	0.05	0.1	0.1	0.1	0.1	0.1	0.1
%Released: 0	PM	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
AnnualUse: 0 kWh/Yr		PeakValue: 0 Watts											

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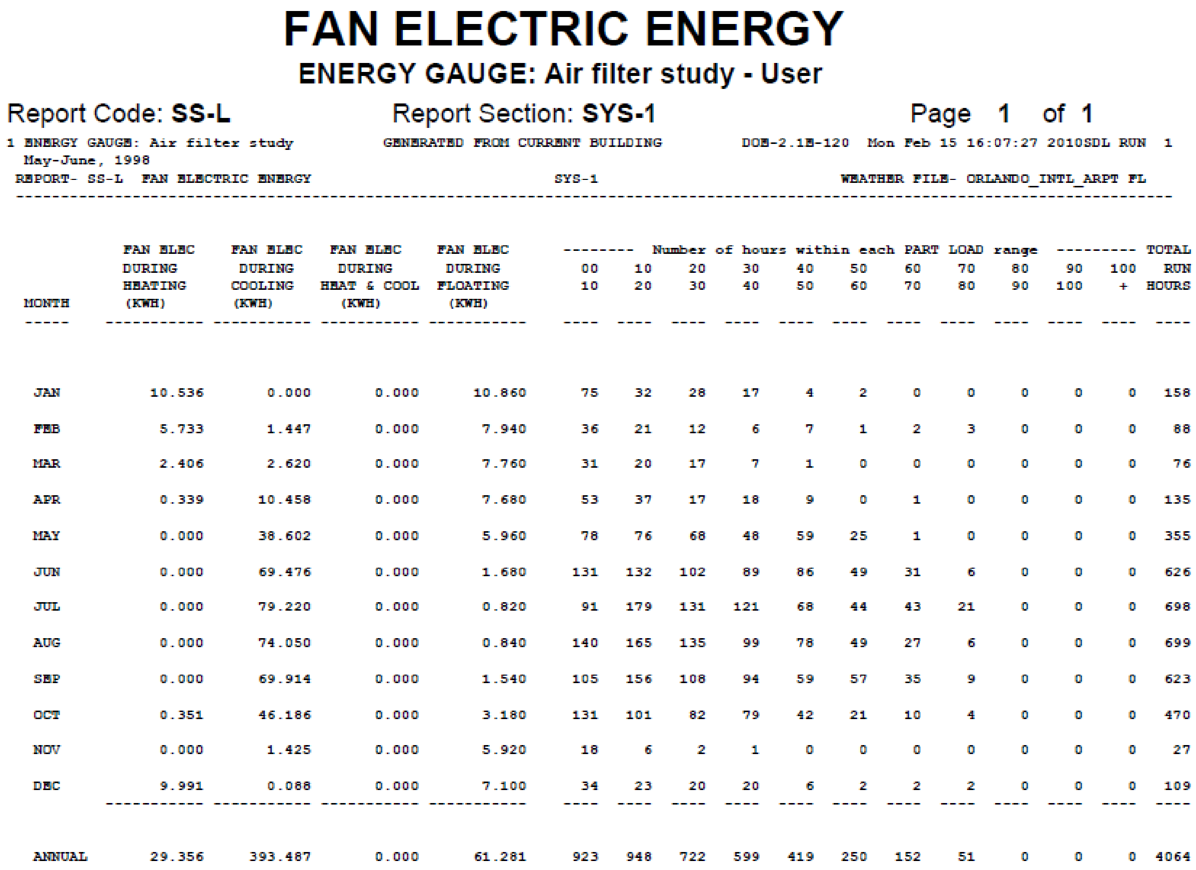
APPENDIX B



FAN ELECTRIC ENERGY

ENERGY GAUGE: Air filter study - User

Report Code: **SS-L** Report Section: **SYS-1** Page **1** of **1**

```
1 ENERGY GAUGE: Air filter study          GENERATED FROM CURRENT BUILDING         DOE-2.1E-120  Mon Feb 15 16:07:27 2010SDL RUN  1
  May-June, 1998
 REPORT- SS-L  FAN ELECTRIC ENERGY                            SYS-1                          WEATHER FILE- ORLANDO_INTL_ARPT FL
 -----------------------------------------------------------------------------------------------------------------------------------


              FAN ELEC    FAN ELEC    FAN ELEC    FAN ELEC     --------  Number of hours within each PART LOAD range  --------- TOTAL
              DURING      DURING      DURING      DURING          00    10    20    30    40    50    60    70    80    90   100   RUN
              HEATING     COOLING   HEAT & COOL  FLOATING         10    20    30    40    50    60    70    80    90   100     + HOURS
  MONTH       (KWH)       (KWH)       (KWH)       (KWH)
  -----    ----------- ----------- ----------- -----------      ----  ----  ----  ----  ----  ----  ----  ----  ----  ----  ----  ----



  JAN           10.536       0.000       0.000      10.860        75    32    28    17     4     2     0     0     0     0     0   158

  FEB            5.733       1.447       0.000       7.940        36    21    12     6     7     1     2     3     0     0     0    88

  MAR            2.406       2.620       0.000       7.760        31    20    17     7     1     0     0     0     0     0     0    76

  APR            0.339      10.458       0.000       7.680        53    37    17    18     9     0     1     0     0     0     0   135

  MAY            0.000      38.602       0.000       5.960        78    76    68    48    59    25     1     0     0     0     0   355

  JUN            0.000      69.476       0.000       1.680       131   132   102    89    86    49    31     6     0     0     0   626

  JUL            0.000      79.220       0.000       0.820        91   179   131   121    68    44    43    21     0     0     0   698

  AUG            0.000      74.050       0.000       0.840       140   165   135    99    78    49    27     6     0     0     0   699

  SEP            0.000      69.914       0.000       1.540       105   156   108    94    59    57    35     9     0     0     0   623

  OCT            0.351      46.186       0.000       3.180       131   101    82    79    42    21    10     4     0     0     0   470

  NOV            0.000       1.425       0.000       5.920        18     6     2     1     0     0     0     0     0     0     0    27

  DEC            9.991       0.088       0.000       7.100        34    23    20    20     6     2     2     2     0     0     0   109
           ----------- ----------- ----------- -----------      ----  ----  ----  ----  ----  ----  ----  ----  ----  ----  ----  ----


  ANNUAL        29.356     393.487       0.000      61.281       923   948   722   599   419   250   152    51     0     0     0  4064
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