Cooling Tower, Pump and Fan Component Constraints of EnergyPlus™ when performing the whole building energy simulation for LEED

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Abstract

According to Appendix G of ANSI/ASHRAE/IESNA Standard 90.1, there are certain requirements for simulating system operation and the equipment efficiencies. However, the modellers should be aware of some very important considerations to ensure compliance with manufacturer's specifications and the standards due to the default values and calculations methods in EnergyPlus™. This paper presents the observations from a case-study where an EnergyPlus™ model was developed to satisfy the requirements of EAc1 for LEED. Under LEED Energy and Atmosphere credit-1 (EAc1) Option 1, an evaluation of the building performance is required through whole building energy simulation. Several parameters related to the components of pump, cooling tower, fan were identified that need to be accurately input in the energy model. The underlying principles and the subsequent calculations involved to achieve compliance with the standards would be presented in this paper. Furthermore, suggestions to accurately model these components to perform the simulation accurately would be presented.

1 Introduction

An energy modeller may often be faced with specifications for equipment that cannot be input into simulation software, and so will find it necessary to calculate the variables that are available for input from these specifications. However, there may often be values buried within the code which can be hard for the modeller to discover and cause the equipment to perform incorrectly. The aim of this paper is to identify some of these values for the pump, fan and cooling tower components in EnergyPlus. These will be identified and this paper will provide equations that incorporate these values so that the inputs into the simulation software allow the specified equipment to work as it should in real life. This paper will also work through some examples to show that these equations are working as required by the user.

2 Methodology

The case-study used in this paper is a newly constructed commercial office building, with administration offices, dining spaces and circulation areas located in Frankfurt, Germany. The project team aims for a Gold certification under the Leadership in Energy & Environmental Design (LEED® v2009) Green Building Rating System which was developed by the U.S. Green Building Council (USGBC).

The design case HVAC system consists of a radiant heating system for space heating, radiant ceiling panels for the office spaces to meet the additional heating and cooling loads. A direct outdoor system (DOAS) feeds into the spaces to provide fresh air to the space. The hot water for radiant system and air handling unit (AHU) heating coils is providing by the boiler. The chilled water for the radiant panels and AHU cooling coil is provided by the water-cooled chiller. Additionally, three roof-top split systems are used for conference room and server room, which have significantly high cooling loads or different operation schedules.

The baseline case HVAC system is system 5, assisted with system 3 for exceptional spaces, according to ASHRAE 90.1-2007. This system was selected based on the usage, number of floors, conditioned area and the heating source. Each thermal block was modeled with its own HVAC system, according to the note in G3.1.1. The systems were oversized according to G3.1.1.2, 15% for the cooling and 25% for the heating. (ASHRAE, 2007) Table1 summarizes the baseline system information that was used for this project.

Table 1: Baseline System Summary

System	*Serving*	*Description*	*Fan Control*	*Cooling Type*	*Heating Type*
5. Packaged VAV with Reheat	Conditioned Space	Packaged rooftop VAV with reheat	VAV	Direct expansion	Hot-water fossil fuel boiler
3. PSZ-AC	Conditioned Space	Packaged rooftop air conditioner	Constant volume	Direct expansion	Fossil fuel furnace

EnergyPlus™ version 8.0 was used to perform the whole building energy simulation. Constraints of the pump, fan and cooling tower components were discovered while performing the proposed design and baseline whole building energy simulation in EnergyPlus™.

PUMP

Pump pressure is used to create pressure differences in fluids. There are three types of pump powers:

1. Pump hydraulic power (P_h) – It is the power required to increase the fluid velocity and pressure.
2. Pump shaft power (P_s) - This represents the sum of the pump hydraulic power and the power loss due to inefficiencies in power transmission from the motor to the fluid, which is typically calculated as the pump hydraulic power divided by the pump impeller efficiency.
3. Pump electric power (P_e) - It is also known as pump nominal power, is the power which finally goes to the electricity bills. It considers the power consumed by the motor to turn the pump shaft, which is typically calculated as the pump shaft power divided by the pump motor efficiency.

The equation below specifies the pump power equation that EnergyPlus™ uses to calculate all these values.

$$P_e = P_s / Eff_{motor} = (P_h / Eff_{impeller}) / Eff_{motor} = P_h / Eff_{tot} \quad (1)$$

In order to define the required power to run a pump in EnergyPlus™, the efficiency of the pump must be taken into account. This is made up of the impeller efficiency ($Eff_{impeller}$) and motor efficiency (Eff_{motor}).

$$Eff_{tot} = Eff_{motor} \cdot Eff_{impeller} \quad (2)$$

In the pump object in EnergyPlus the user can input the motor efficiency. However, there is no manual input for the impeller efficiency. If either pump power or pressure drop is autosized, the software assumes an impeller efficiency of 0.78.

In order to model water pumps according to Appendix G it is necessary use the pump motor power-per-unit-flow method. This method defines the electric power of the pump divided by the flow at design conditions. The power-per-unit-flow method specified in ASHRAE 90.1-2007 Appendix G, requires that the baseline building design pump power shall be:

- *“G3.1.3.5 The baseline building design hot-water pump power shall be 19 W/gpm. The pumping system shall be modeled as primary-only with continuous variable flow.*
- *G3.1.3.10 The baseline building design pump power shall be 22 W/gpm. Chilled-water systems shall be modeled as primary/secondary systems. (22 W/gpm is the total pump power allowance for all primary/secondary CHW pumps, and not the power of each pump, however there is no guidance from Appendix G as to how this should be split between the two pumps)*
- *G3.1.3.11 The baseline building design condenser water pump shall be 19 W/gpm. Each chiller shall be modeled with separate condenser water and chilled-water pumps interlocked to operate with the associated chiller.”*

The pump motor power-per-unit-flow and impeller efficiency are not input variables in EnergyPlus™. In fact, impeller efficiency was previously indicated to have an assumed value (0.78) by EnergyPlus™. In EnergyPlus™ the pump object has 6 user-defined variables to calculate the pump power: rated flow rate, rated pump head, rated power consumption, motor efficiency, fraction of motor inefficiencies to fluid stream, and the coefficient of the part load performance curve. So these variables need to be calculated.

Field	Units	Obj1
Name		Heat_Pump
Inlet Node Name		Heat_Pump_In
Outlet Node Name		Heat_Pump_Out
Rated Flow Rate	m3/s	autosize
Rated Pump Head	Pa	244792.39
Rated Power Consumption	W	autosize
Motor Efficiency		0.9
Fraction of Motor Inefficiencies to Fluid Stream		0.1
Coefficient 1 of the Part Load Performance Curve		0
Coefficient 2 of the Part Load Performance Curve		1
Coefficient 3 of the Part Load Performance Curve		0
Coefficient 4 of the Part Load Performance Curve		0
Minimum Flow Rate	m3/s	0
Pump Control Type		Continuous
Pump Flow Rate Schedule Name		Always On

Values must be calculated in order to ensure the correct pump power and efficiency is calculated by the software

Describes the performance of the pump at part load

Figure 1: Pump:VariableSpeed EnergyPlus™ component inputs

Calculating the values for the first five variables requires the user to know the following information:

- Maximum flow water flow rate through the pump.
- Motor efficiency of the pump; there is no baseline requirement for motor efficiency. However, adjusting pumping head or rated power capacity based on the different motor efficiency input would achieve the required performance.

Table 2: EnergyPlus Motor Efficiency – Pressure Drop Inputs

Eff_{motor}	$Eff_{impeller}$	Eff_{tot}	*19 W/gpm Pressure Drop (Pa)*	*22 W/gpm Pressure Drop (Pa)*
100%	78%	78%	234902	271992

90%	78%	70%	211412	244792
80%	78%	62%	187921	217593
70%	78%	55%	164431	190394
60%	78%	47%	140941	163195
50%	78%	39%	117451	135996
40%	78%	31%	93961	108797
30%	78%	23%	70471	81597
20%	78%	16%	46980	54398
10%	78%	8%	23490	27199

- Fraction of motor inefficiencies to the fluid stream
- Efficiency of the pump (W/gpm)
- Impeller efficiency (the default in EnergyPlus is 0.78, inputting the rated power consumption and the rated pump head will override this default)

The pumps performance at part load is described by coefficient 1 – 5 of the part load performance curve. The fraction of full load power is determined by the cubic equation (Baseline Case – Linear):

$$FractionFullLoadPower = C_1 + C_2 PLR + C_3 PLR^2 + C_4 PLR^3 \quad (3)$$

When all the coefficients are 0 except C_2 which is 1 the following curve is described.

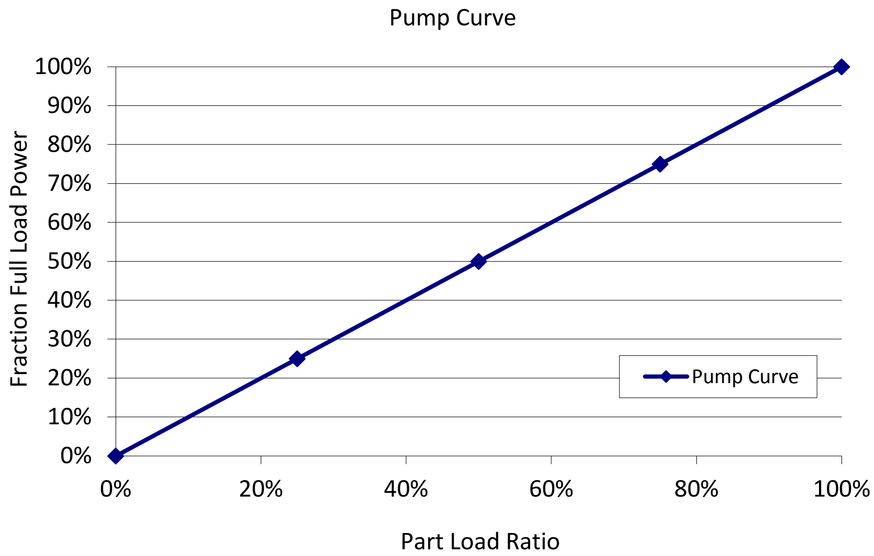

Figure 2: Baseline Pump Curve

Example:

- Maximum flow water flow rate through the pump
  autosize (EnergyPlus sizing of pump rated flow rate 0.00848622 m^3/s)
- Motor efficiency of the pump
  90%
- Fraction of motor inefficiencies to the fluid stream
  10%

- Performance of the pump
  19W/gpm (83.66 W/(m^3/h))
- Pump impeller efficiency
  78%

Results from EnergyPlus™ simulation for the hot water pump in the baseline model were plotted to verify its performance as shown in Fig.3.

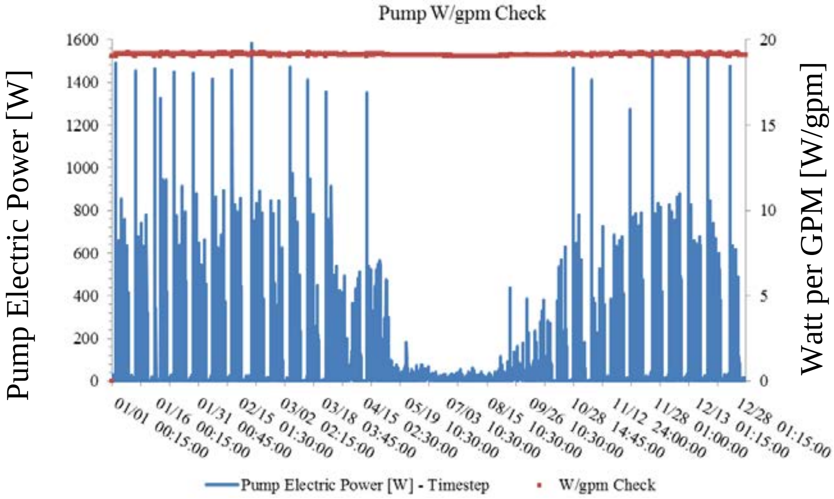

Figure 2: Baseline Pump 19 W/gpm Check

Similar approach can be used to assess the inputs for the water pumps in the design case model. Simulation experts often use a single equivalent pump in energy models instead of modeling multiple pumps. Based on the Table 3, the design case equivalent pump performance was calculated to be 6.5 W/gpm. This value was calculated by dividing the total power of all the pump motors with the total flow rate of the system (Total pump motor power/ Total flow rate).

Table 3.1 Proposed Design Case Pump details

Proposed Design Chilled Water Pump List	*Pump Power [kW]*	*Pump Power [HP]*	*Flow Rate [m3/h]*	*Flow Rate [gpm]*	*Motor Power [kW]*	*Motor Power [HP]*	*Pressure (ft)*	*Pressure (m)*
Primary Chilled Water Pump	1.88	2.52	95.0	418	1.90	2.55	16.40	5.0
AHU Cooling Coils	1.57	2.11	72.0	317	1.90	2.55	19.69	6.0
Radiant Ceilings	2.91	3.90	82.0	361	3.20	4.29	32.81	10.0
Fan Coil Units	1.17	1.57	41.0	181	1.25	1.68	16.40	5.0

Table 3.2 (Cont.) Proposed Design Case Pump details

Proposed Design Chilled Water Pump List	*Design case W/gpm*	*Pump impeller Efficiency*	*Motor Efficiency*
Primary Chilled Water Pump	4.5	68.8%	98.9%
AHU Cooling Coils	6.0	75.0%	82.6%
Radiant Ceilings	8.9	76.8%	90.9%
Fan Coil Units	6.9	47.7%	93.6%

COOLING TOWER

In EnergyPlus, “CoolingTower:TwoSpeed” component is used for the baseline cooling tower simulation. The leaving condenser water temperature is controlled by using two-speed fan motors. EnergyPlus™ has two performance input methods which allow the user to specify tower performance: “UFactorTimesAreaAndDesignWaterFlowRate” or “NominalCapacity”

The design water flow rate is assumed to be 5.382×10^{-8} m3/s per watt of nominal capacity (3 gpm/ton), the calculation is shown in formula (4). *“The thermal capability of a cooling tower used for air conditioning is often expressed in nominal cooling tower tons. A nominal cooling tower ton is defined as cooling 3 gpm of water from 95°F to 85°F at a 78°F entering air wet-bulb temperature. At these conditions, the cooling tower rejects 15,000 Btu/h per nominal cooling tower ton. The historical derivation of this 15,000 Btu/h cooling tower ton, as compared to the 12,000 Btu/h evaporator ton, is based on the assumption that at typical air-conditioning conditions, for 12,000 Btu/h of heat picked up in the evaporator, the cooling tower must dissipate and additional 3000 Btu/h of compressor heat. ”* [ASHRAE Handbook]

$$\dot{V}_{tower,w,des} = 5.382 \times 10^{-8} \cdot \dot{Q}_{tower,nom} \tag{4}$$

The nominal fan power was sized to be 0.0105 times the design load.

$$\dot{Q}_{fan,nom} = 0.0105 \cdot \dot{Q}_{tower,nom} \tag{5}$$

The unit conversion is listed in Table 4 below, thus EnergyPlus™ default calculation assumes a 59.7 gpm/hp cooling tower performance (which uses the maximum flow rating of the tower divided by the fan nameplated rated motor power).

Table 4: EnergyPlus Fan Power Calculation Factor

Unit	*Fan Power Calculation Factor*
m3/s per W, nominal Capacity	5.38E-08
gpm per W, nominal Capacity	8.40E-04
gpm per W, cooling tower fan rated motor power	8.00E-02
gpm per hp, cooling tower fan rated motor power	5.97E+01

ASHRAE 90.1-2007 Appendix G.1.3.11 requires that “The heat rejection device shall be an axial fan cooling tower with two speeds fans. Table 6.8.1G requires a minimum baseline case performance of 38.2 gpm/hp, for axial fan open cooling towers at 75/85/95°F condition (wet-bulb outdoor air / leaving water / entering water). Cooling tower performance is defined as the maximum flow rate of the tower divided by the fan nameplate rated motor power.”

Cooling tower capacity varies with fan speed, and the energy varies with the cube of the speed ratio. The overall fan energy in the energy plus results reflects cooling tower energy consumption. For baseline simulation, fan power at high fan speed and fan power at low fan speed need to be defined. The default value of fan power calculation factor is 0.0105 in EnergyPlus™, inputting the fan power at rated fan speed, and cooling tower nominal capacity will override this default value. The “fan power at high fan speed” and “fan power at low fan speed” fields need to be fulfilled with calculated values and should not be “autosize”. For example, the calculated adjusted fan power sizing factor is 0.0166 for baseline cooling tower performance of 38.2 gpm/hp.

Example:

- Design water flow rate

0.0104345 m3/s, from EnergyPlus Sizing of 3 gpm/ton

- Cooling tower performance requirement
  38.2 gpm/hp
- High speed nominal capacity
  193877 W
  Using (4):

$$\dot{Q}_{tower,nom} = \dot{V}_{tower,w,des} / 5.382\times10^{-8} = 0.0104345 / 5.382\times10^{-8} = 193877W$$

- Calculated fan power at design fan speed
  3220 W

Using (5) with adjusted factor: $\dot{Q}_{fan,nom} = 0.0166 \cdot \dot{Q}_{tower,nom} = 3220W$

OR Using $38.2gpm/hp$ *and* $0.0104345m^3/s$

Field	Units	Obj1
Name		Cond_CT
Water Inlet Node Name		Cond_CT_In
Water Outlet Node Name		Cond_CT_Out
Design Water Flow Rate	m3/s	
Air Flow Rate at High Fan Speed	m3/s	autosize
Fan Power at High Fan Speed	W	3219.5
U-Factor Times Area Value at High Fan Speed	W/K	
Air Flow Rate at Low Fan Speed	m3/s	autosize
Fan Power at Low Fan Speed	W	3220.5
U-Factor Times Area Value at Low Fan Speed	W/K	
Air Flow Rate in Free Convection Regime	m3/s	autosize
U-Factor Times Area Value at Free Convection Air Flow	W/K	
Performance Input Method		NominalCapacity
High Speed Nominal Capacity	W	193877
Low Speed Nominal Capacity	W	96938.5
Free Convection Capacity	W	
Basin Heater Capacity	W/K	
Basin Heater Setpoint Temperature	C	2

Figure 3: Screenshot of Cooling Tower EnergyPlus Inputs

Results from EnergyPlus™ simulation for the baseline performance of the cooling tower in the model were plotted to verify its performance as shown in Fig.5:

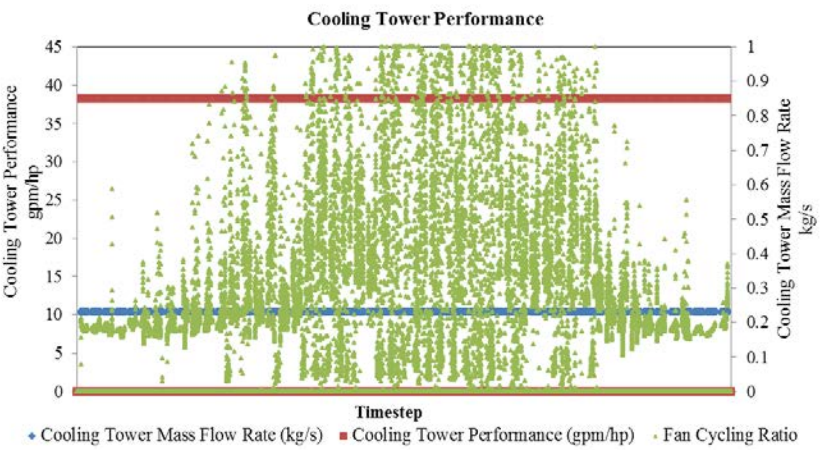

Figure 4: Cooling Tower Performance Check

The cooling tower fan is cycling on and off during each timestep; when the cooling tower cycles on, the performance is >=38.2 gpm/hp as required. The condenser side flow rate of the chiller is sized according to the concept and equation above.

FAN

The component of Fan:VariableVolume was used to model the fans in EnergyPlus™. The component energy performance was verified as a part of the case study. All the inputs required by EnergyPlus™ for this component are used in calculating the end-energy usage. Post-processing the information related to the part load ratio from the simulated results and the simulation inputs was done to calculate the energy performance. This was compared against the simulated energy performance value for validation. ASHRAE 90.1-2007 Appendix G3.1.3.15 – VAV Fan Part-Load performance requires that "VAV system supply fan shall have variable-speed drives, and their part-load performance characteristics shall be modeled using either Method 1 or Method 2 specified in table G3.1.3.15. In this case study, method 2 was used to simulate the fan performance."

$$P_{fan} = 0.0013 + 0.1470 \times PLR_{fan} + 0.9506 \times (PLR_{fan})^2 - 0.0998 \times (PLR_{fan})^3$$

Where, P_{fan} : fraction of full-load fan power and

PLR_{fan} : fan part-load ratio (current cfm / design cfm)

In EnergyPlus™, following input was used:

Field	Units	Obj1
Name		PVAV-02_Fan
Availability Schedule Name		Fan Schedule
Fan Efficiency		0.65
Pressure Rise	Pa	2124
Maximum Flow Rate	m3/s	1.52
Fan Power Minimum Flow Rate Input Method		FixedFlowRate
Fan Power Minimum Flow Fraction		
Fan Power Minimum Air Flow Rate	m3/s	0.74
Motor Efficiency		0.9
Motor In Airstream Fraction		1
Fan Power Coefficient 1		0.0013
Fan Power Coefficient 2		0.1470
Fan Power Coefficient 3		0.9506
Fan Power Coefficient 4		-0.0998
Fan Power Coefficient 5		0
Air Inlet Node Name		PVAV-02_OAM_Out
Air Outlet Node Name		PVAV-02_Fan_Out
End-Use Subcategory		General

Figure 6: Fan:VariableVolume component in EnergyPlus™

The following output variables were extracted from the simulation results.

PVAV-02_FAN_OUT: System Node Mass Flow Rate [kg/s](TimeStep)

PVAV-02_FAN: Fan Electric Power [W](TimeStep)

The fan electric power from the simulation results was then compared to the calculated value based on the following equations:

Since, $FullLoadPower = \Delta P \times Q$

ΔP : Full load fan pressure rise (Pa)

Q: Design (Maximum) flow rate (m^3/s)

So, $FullLoadPower = \Delta P \times Q = 2124 \times 1.52 = 3228.48W$

The difference of both these fan power values was plotted for each of the partial load ratio values. Figure 7 presents the results that were plotted from the case study. The difference of both the fan power values is expected to be zero for all the partial load values. However, it was noticed that the differences was close to zero only for higher partial load ratio values.

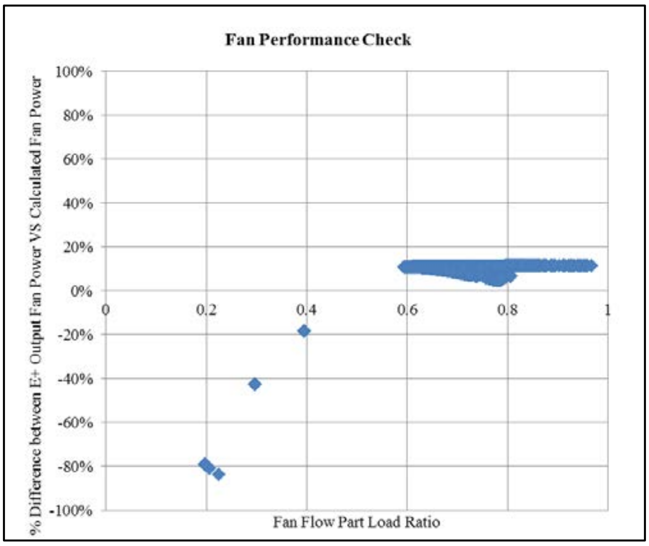

Figure 7: Fan performance check

In addition, the input "Motor in air stream fraction" would have an impact on cooling, heating, and pump energy consumption; if the motor is in the air stream, the waste heat from the motor and belt will goes into the air stream. So the heating energy would reduce, and the cooling would increase due to the waste heat dissipated to the air stream.

The difference of the energy consumption is listed in table 5:

Table 5. Energy Consumptions comparison for different "Motor in air stream fraction"

	Fuel	*Motor in air stream fraction = 0 Energy usage*	*Motor in air stream fraction = 1 Energy usage*	*% Energy Difference*
Heating (Gas)	Gas	187.94 MWh/a	188.53 MWh/a	0.31%
Heating (Electric)	Electricity-Grid	0.00 MWh/a	0.00 MWh/a	-
Heating (DES)	District Heating	0.00 MWh/a	0.00 MWh/a	-
Cooling	Electricity-Grid	30.48 MWh/a	29.82 MWh/a	-2.15%
Pumps	Electricity-Grid	0.21 MWh/a	0.21 MWh/a	0.33%
Fans	Electricity-Grid	101.86 MWh/a	101.86 MWh/a	0.00%
Lighting Interior	Electricity-Grid	36.66 MWh/a	36.66 MWh/a	0.00%
Lighting Exterior	Electricity-Grid	2.46 MWh/a	2.46 MWh/a	0.00%
Hot Water (Gas)	Gas	0.00 MWh/a	0.00 MWh/a	-
Hot Water (Electricity)	Electricity-Grid	8.84 MWh/a	8.84 MWh/a	0.00%
Hot Water (DES)	District Heating	0.00 MWh/a	0.00 MWh/a	-
PROCESS LOAD	Electricity-Grid	55.25 MWh/a	55.25 MWh/a	0.00%

ASHRAE 90.1-2007 Appendix G3.1.2.2-Equipment Capacity requires that "*Unmet load hours for the proposed design or baseline building designs shall not exceed 300 (of the 8760 hours simulated).* " and G3.1.2.4-Fan System Operation states that "*Supply and return fans shall operate continuously whenever spaces are occupied and shall be cycled to meet heating and cooling loads during unoccupied hours*."

Figure 6 shows a sample day fan operation check. If there is a cooling or a heating demand, the fan cycles on and off each timestep. However, the fan night cycled control is not always based on the heating or cooling load demand at each step, but may also depend on the fan operation status of last time step. Even if there are still cooling or heating needs at a timestep, the fan would cycle off if the fan cycled on during the previous timestep.

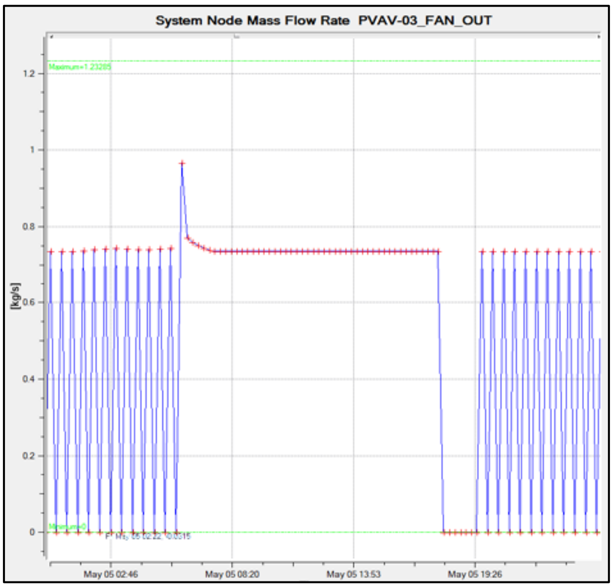

Figure 8: Fan Operation Sample Day Check

This fan night cycle operation control strategy in EnergyPlus™ resulted in excess unmet load hours. In order to meet the ASHRAE unmet load hours requirement without any post-process calculation, the fan needs to be switched on during the night to reduce the unmet load hours. For variable air volume systems with terminal reheat systems, the operation of the fan will have enormous influence on the overall energy consumption. The night cycle operation also impacts the following parts of the system: heating, cooling, hot/ cold water pumps. Table 6 and Figure 9 presents the analysis of unmet load hours and overall energy consumption for the two situations above.

Table 6: Unmet load hrs using Fan night cycle in EnergyPlus

	w Fan Night Cycle Time Setpoint Not Met [h]	*Fan Always On Time Setpoint Not Met [h]*
During Heating	2807.00	0
During Cooling	24.00	0
During Occupied Heating	16.50	0
During Occupied Cooling	0.00	0

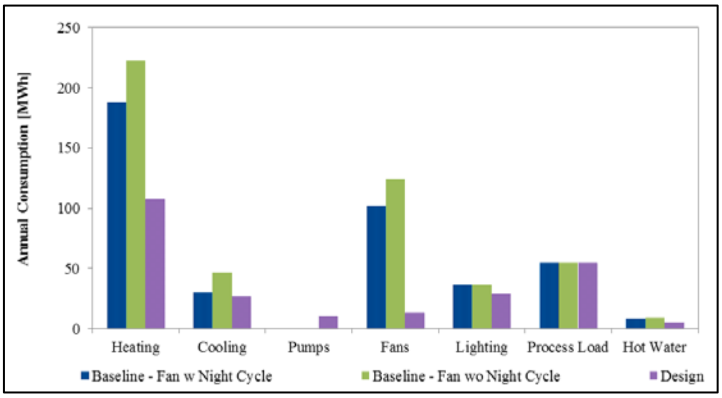

Figure 9: Energy Usage Comparison for Different Fan Operation Controls

The mechanical loads can be met by maintaining the system operation when there is a cooling / heating demand. Figure 8 shows the analysis of the unmet load hours from the simulation results. The fan operation hours in the graph refers to all those instances when the system is switched on when there is a heating/cooling demand. The fan cycled off hours refer to the hours when the system remains switched off while there is a demand. The fan adjusted operation hours refer to the all those instances when there is a demand for heating or cooling. Based on these fan adjusted operation hours, the system fan operation schedules can be programmed in EnergyPlus™ based on the load calculation of each zone under each VAV system.

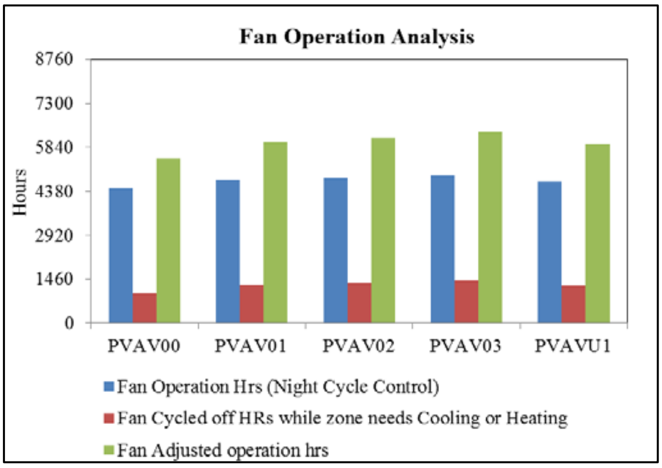

Figure 5: Fan Operation Analysis

3 Conclusion

Simulation software contains code and default values that are not necessarily clear, and which may affect the performance of the simulation. It is important to find these values, to make sure that the calculations for the inputs into the simulation comply with Appendix G or the specifications given to the modeller. In this case, the paper indicated that the motor impeller efficiency in EnergyPlus™ is assumed to be 0.78. Unless all component variables are filled in the pump object, the pump will perform with this value. Appendix G specifies the pump efficiency to be modelled using 19W/gpm. In order to comply with this, it is important to understand all the inputs that combine to create this efficiency.

The cooling tower object has a fan power sizing factor of 0.0105 which it uses in conjunction with the nominal capacity input. In order to model the cooling tower with the Appendix G specified efficiency of 38.2 gpm/hp, the fan power must be calculated based on the adjusted fan power sizing factor.

The AvailabilityManager:NightCycle does not have proper fan cycling control during the unoccupied hours. The fan operation schedule must be programmed based on the zone cooling and heating loads demand.

It is most important to make sure all the variable inputs are filled with values that have been calculated by the modeller, as there are often default values in the simulation program code which may affect the performance of the component in the energy modelling software. The variables with default values should be used with caution as they may have considerable effect on the end-results.

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