THE PRACTICAL APPLICATION OF BUILDING SIMULATION TOOLS IN DESIGNING A REAL BUILDING

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ABSTRACT

A Building Energy simulation tool and a Thermal Comfort simulation tool were used to design, select and specify various components and combinations of components for a 1.3 million square foot building in Los Angeles.

The Thermal Comfort simulation tool was used to optimize the design of the conditioning system for the building occupants. In addition, various alternatives were analysed, and different operating and redundancy scenarios were investigated. The parameters from the optimized comfort design were then integrated into the operating schedules of the building simulation tool. The building simulation tool was used to define the operating energy consumption of the building as well as refine operating strategies for a fully and partially occupied building. One of the major concerns of the client was the design of the chiller plant, specifically: the number of chillers, the number of chillers operating at one time, chiller plant redundancy and the possible requirement of a dedicated night time or weekend chiller. This paper will explain the criteria for designing and operating comfortable work spaces and how these optimized designs are integrated into the simulation to assess building loads, lower energy consumption and enhance overall building performance.

INTRODUCTION

This paper addresses the application of building simulation in real world design. Many questions have been raised regarding whether or not building simulation can be practically used in the design of a building and its systems. The authors believe that the design process has been accelerated and improved. The data made available from the simulation programs have provided valuable information for all members of the design team.

Three different programs were used to simulate several conditions for spaces within the building. The first program was used to simulate occupant comfort in different locations within the building. The second program used was an Energy simulation program to assess energy consumption of a number of energy conservation measures. The third program used was a computational fluid dynamics study of the atrium space. (Due to the complexity of the analysis the CFD results were ommitted from this paper.)

ANALYSIS METHODOLOGY

Load Analysis

The overall building and space loads were calculated using two parallel methodologies and verified by the energy simulation software.

A Microsoft Excel spreadsheet was developed to calculate the building space loads, including envelope loads, occupancies, lighting power densities, equipment power densities and any additional plug loads. The spreadsheet also calculated outside air loads for each of the air handling unit coils. The spreadsheet was continually updated from the schematic design phase through to bridging documents.

In addition to the spreadsheet, a full load calculation was conducted using Trane Trace. The Trace calculation was conducted later in the design phase and provided validation of the Excel spreadsheet.

Comfort Analysis

The first simulation analyzed the occupant comfort conditions in the different spaces. The ROOM module of E+TA was used to determine the comfort conditions in each space for the different conditioning systems. Each of the different spaces were required to have conditioning systems that would provide comfort conditions that would remain within predicted mean vote (PMV) limits of +/- 0.5. The output from the comfort analysis program also supplied data such as space dry bulb and mean radiant temperatures that were used to provide space operating temperatures.

The results and analysis of the comfort conditions provided the design parameters that were subsequently entered into the energy analysis software for each of the space conditioning systems.

Energy Analysis

Energy analysis was then conducted on several different space conditioning systems. For the design conditioning system, several central plant configurations were analyzed for energy efficiency using VisualDOE, which utilizes the DOE 2.1 energy simulation engine. The entire building was entered into the energy simulation model based on the architectural CAD files and the Basis of Design documents. DOE 2.1 analyzed the energy consumption for each conditioning system of all 8,760 hours of the year.

Chiller Plant Analysis

Once the envelope and conditioning system had been optimized using the simulation programs the chiller plant was further analyzed. The chiller plant was analyzed with respect to chiller load configuration and chiller efficiency.

Both the energy simulation software and an Excel spreadsheet were used to conduct a more detailed analysis of various central plant configurations.

Computational Fluid Dynamics

The large volume atrium space, which contained a significant amount of glazing, could not be accurately modeled by either the comfort or energy simulation software alone. For this particular space, extensive analysis was undertaken with a computational fluid dynamics (CFD) model. The CFD model on this project was developed in StarCD. The outputs from the CFD were then used to define and verify the atrium calculations in both the comfort and energy modeling processes.

THE BUILDING

The building being analyzed has a total floor area of 1.1 million square feet, each floor is 46,000 square feet and the building has a large atrium. The exterior glazed surfaces were 60% of the building enclosure (excluding the atrium).

The base case for the conditioning system was an overhead variable air volume (VAV) system. An underfloor air distribution system was compared to the base case system.

MODELING

Comfort Results

The occupied spaces of the building were modeled; the results were studied and optimized to maintain conditions slightly lower than PMV +/- 0.5 during the summer. The results of the comparison are shown in the graphs at the end of the paper.

The following graphs represent the following conditions:

Figure 1 shows the internal temperatures for a typical space when supplying different volumes of air at 13°C by means of a conventional overhead VAV system.

Figure 2 shows the PPD (percentage of people dissatisfied) results for a conventional overhead VAV system. The results clearly show that when the space temperature is warmer (i.e. 22-24°C) the PPD results are lower than 10%, which is the recommended maximum. As the space temperatures are reduced as a result of supplying more air to the space, comfort conditions rapidly deteriorate. The maximum PPD is nearly 60% for a space ventilated with 8 air changes.

Figure 3 shows the PMV (predicted mean vote) results for a conventional overhead VAV system. The essence of the PMV scale is an indication of whether the space is too hot or too cold. As all the results are negative the space is too cold.

Figure 4 shows the dry bulb temperatures in the spaces using displacement ventilation systems. Space temperatures are maintained between 22 and 24°C. The air volume required to maintain these temperatures is 2-3.5 air changes which is considerably lower than 6-8 air changes as requested in the design guide. Another advantage of the underfloor displacement system is that equipment size is downsized, and due to the supply temperature of 17°C, the supply ductwork does not require insulation.

Figure 5 shows the PPD (percentage of people dissatisfied) results for an underfloor displacement ventilation system. The results for the three alternatives are all lower than 10% PPD.

Figure 6 shows the PMV (predicted mean vote) results for an underfloor displacement ventilation system. The results are between –0.3 and 0, which indicates that the spaces will be slightly cool to thermally neutral.

Energy Analysis

The results of the base case VAV system indicated that the annual energy consumption would be about 46.2 kBtuh/sq.ft yr. When comfort set points were used for an underfloor air distribution system the annual energy consumption was predicted to be about 43.4 kBtuh/sq.ft-yr. Daylight lighting controls were added and the building occupancy was reduced to reflect the expected operating occupancy of the building. These two revisions to the model reduced the annual energy consumption down to 32.8 kbtuh/sq.ft-yr, a significant decrease from the base case of 46.2 kbtuh/sq.ft-yr.

Figure 7 shows the annual energy consumption of 15 alternatives. The results show that for this building and its systems the annual energy consumption was reduced from the original mandate of 55kbtuh/sq.ft-yr to 46.2 kbtuh/sq.ft-yr, which is a 16% reduction. The probable annual energy consumption will be about 33 kbtuh/sq.ft-yr, which is a 40% reduction compared to the mandate.

Figure 8 shows the results of the electrical consumption for the whole year for each alternative. The figure indicates the greatest energy reduction is associated with the fan energy of the underfloor air distribution system and the use of natural daylight to reduce the energy consumption of the artificial lighting.

Chiller Plant Performance

The next step was to optimize the chiller plant. Based on the energy-load analysis and verified against a load spreadsheet, the building peaked at about 2,000 tons. The reserve capacity on the building raised the total required load of the chiller plant to 2,400 tons, to be distributed across three 800 ton chillers.

Three chiller plant configurations were considered during the energy analysis:

A. Constant speed chillers, with constant primary flow, variable secondary flow and constant condenser flow;

B. Variable speed chillers, with variable primary flow, variable secondary flow and constant condenser flow;

C. Variable speed chillers with variable primary flow, variable secondary flow and variable condenser flow.

A number of other configurations and load distribution between the chillers, aside from those mentioned here, could also be analyzed using the method described in this paper.

At the time the analysis was conducted, no chiller performance curves for variable speed chillers under both variable condenser and constant condenser flow were available; in addition, the energy analysis software was not capable of modeling variable primary flow pumping or variable condenser flow pumping. For this reason the chiller plant configurations were analyzed in a spreadsheet to determine the overall plant performance at varying building load conditions; the overall plant efficiency was expressed in kW/ton.

The Excel spreadsheet used to analyze the chiller plants was broken down by plant components (primary pumps, secondary pumps, condenser water pumps, cooling towers and chillers) and calculated the part load performance of each component. For each part load condition the overall part load performance was determined.

The spreadsheets used to analyze the variable speed chillers with variable primary flow, variable secondary flow and constant condenser flow are shown in Table 2 and Table 3.

A summary of the three chiller plant part load energy consumptions is shown in Table 4.

These part load efficiencies were then applied to the 8,760 hour load output data available from the energy load analysis to provide a reasonable representation of the overall energy consumption of the three configurations under the building load profile. The results from this analysis can be seen in Table 1.

Chiller Plant Annual Electrical Consumption	
Configuration	kW/Year
Primary/ Secondary	2,134,257
VFD Chillers Const. Condenser Flow	2,013,536
VFD Chillers Var. Condenser Flow	1,952,017

Table 1: Annual Chiller Planet Energy Consumption

This table was used to determine that while VFD chillers with variable condenser flow provided energy savings over the constant condenser flow with VFD chillers, the real savings were seen with the implementation of the VFD chillers themselves.

RESULTS DISCUSSION

The explanation of energy simulation and thermal comfort analysis can be problematic to say the least. Historically the performance of a building and its systems are expressed as a series of checksums, kBtu/hr yr, sq.ft/ton and CFM/sq.ft. (for this particular building these were 33 kBtu/sf.yr, 550 sq.ft./ton and 1.2 CFM/sq.ft respectively).These are usefully engineering checks for buildings designed using more traditional design and analysis methods. However, the introduction of comfort levels in the form of PPD/PMV, and extensive energy modeling and analysis push the envelope of efficient building design and tend not to conform to the checksums values expected from traditionally designed buildings. In order to successfully review a building using a design methodology as described in this paper, reviewers require an in-depth understanding of the comfort analysis calculations as well as the ability to interpret the results successfully. The same is true of annual energy analysis.

Through the process of this analysis it was found that there was a strong correlation between the loads obtained from the spreadsheet, energy analysis software and Trane Trace load calculation. It is important to note that while the performance based design methodology described here gains acceptance in the United States it may still be necessary for design engineers to carry traditional calculations throughout the design process to allay concerns of either reviewers or other members of the design team. The correlation between the calculations methods (spreadsheet buildng load 2,000 tons, energy analysis building load 1,950 tons, Trace calculation 1,900 tons) should be used by design engineers to advance the use of performance based designs to continually push the envelope of efficient building design.

CONCLUSIONS

The extensive use of an energy simulation is invaluable in analyzing the performance of such a large building. Once the model has been correctly built, the various alternatives that can be analyzed provide a broad spectrum of information, which helps nurse the design in certain directions. One point often forgotten on the analysis of buildings and their associated systems is part load operation. Obviously peak loads were used for equipment selection, but the part load analysis was used to select the number of units to be used for each system. The energy simulation was invaluable in selecting the number and size of chillers for example. The comfort analysis is beneficial in determining optimal operating parameters to maintain occupant comfort in spaces. As the results show, design parameters can be developed from the comfort analysis and then used to provide operating set points for the annual energy simulation program. The end benefit of this approach is that the eventual occupants will have a comfortable environment, and the building will consume less energy and therefore cost less to run.

CONTINUING RESEARCH

Significant work is required on the part of HVAC software developers, both energy analysis and load calculations, to meet the design needs of engineers and the sustainable, efficient designs that are emerging with greater frequency in the North American market. Until this occurs the methodology outlined in this paper can be utilized and refined to design more efficient and comfortable buildings.

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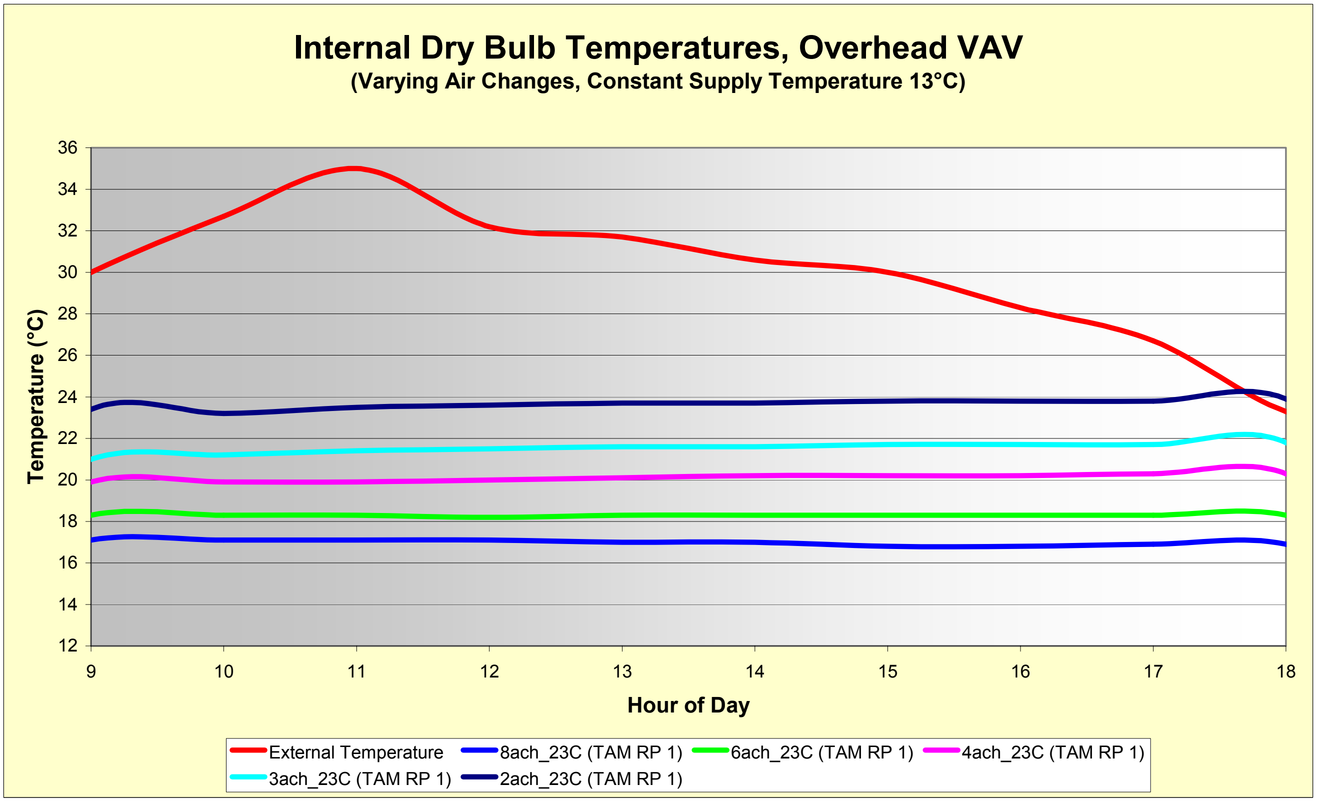

Figure 1: Internal Dry Bulb Temperatures for an Overhead VAV system at constant supply air temperatures varying air volumes.

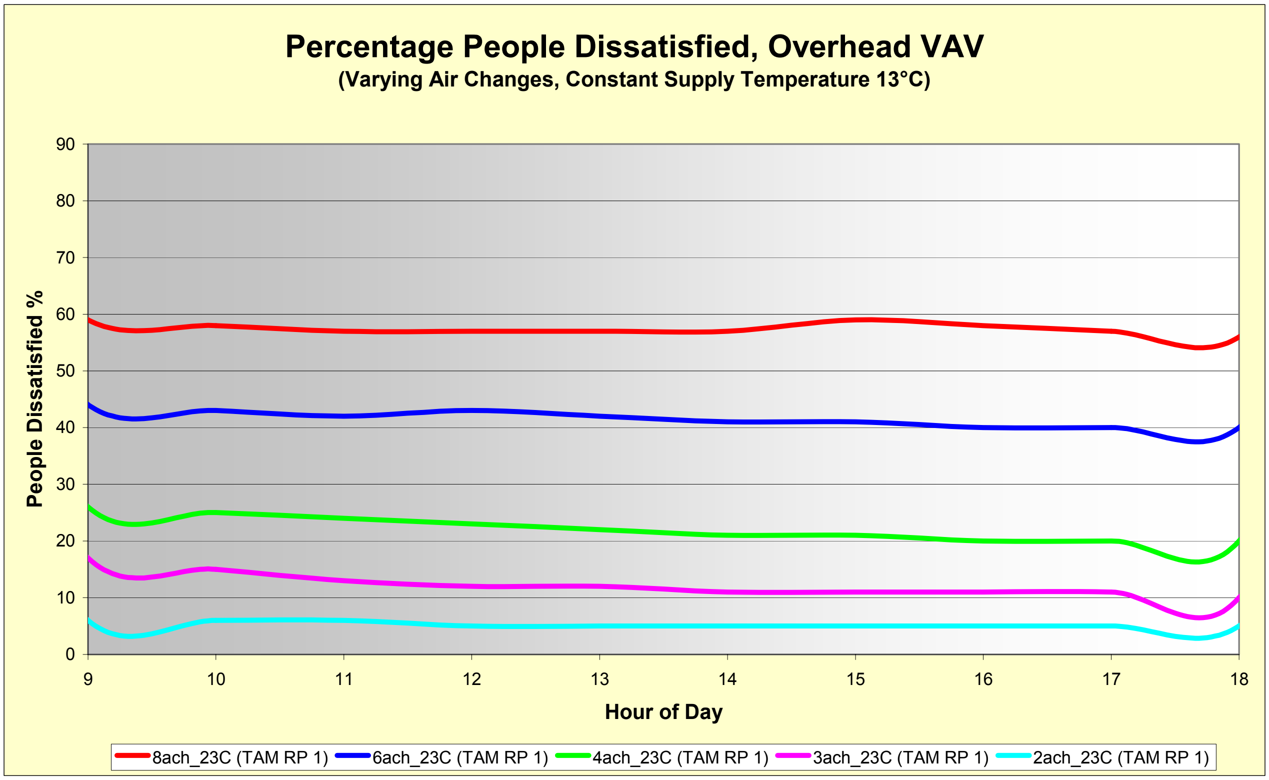

Figure 2: PPD for an Overhead VAV system; different supply air volumes at a constant temperature

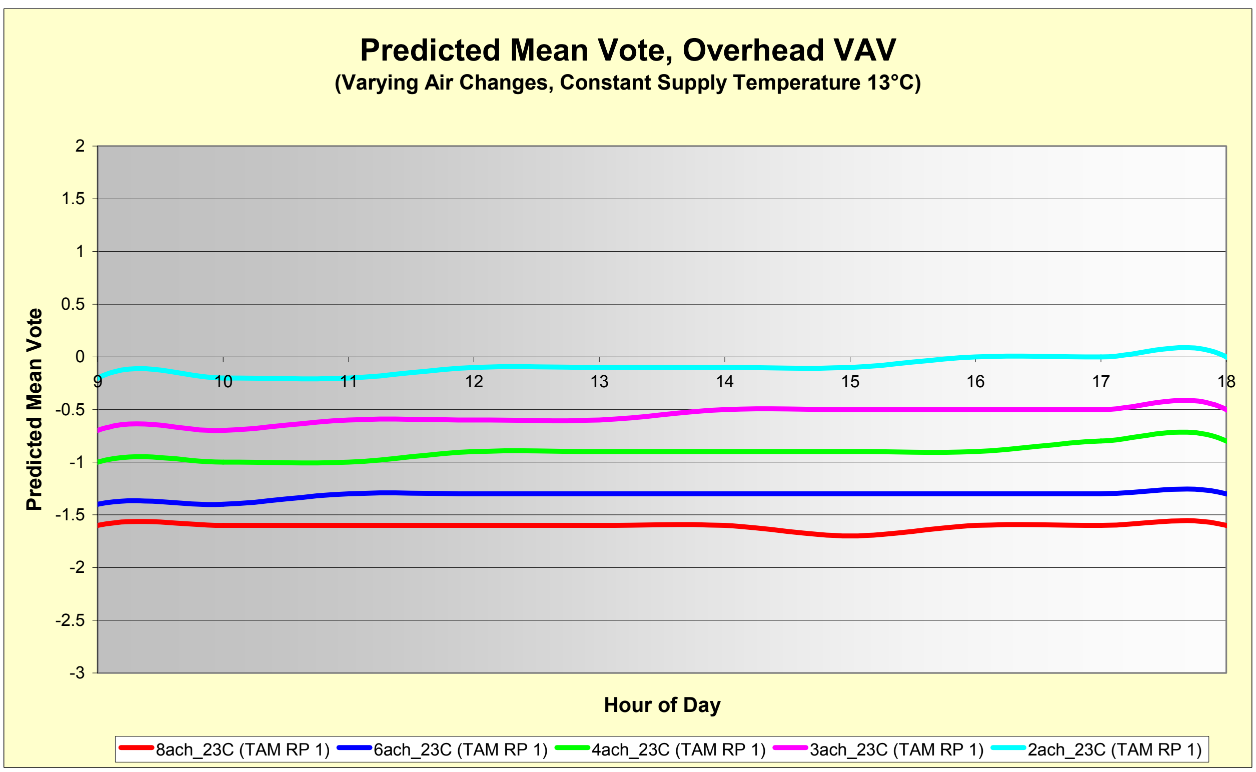

Figure 3: PMV for an Overhead VAV system; different supply air volumes at a constant temperature.

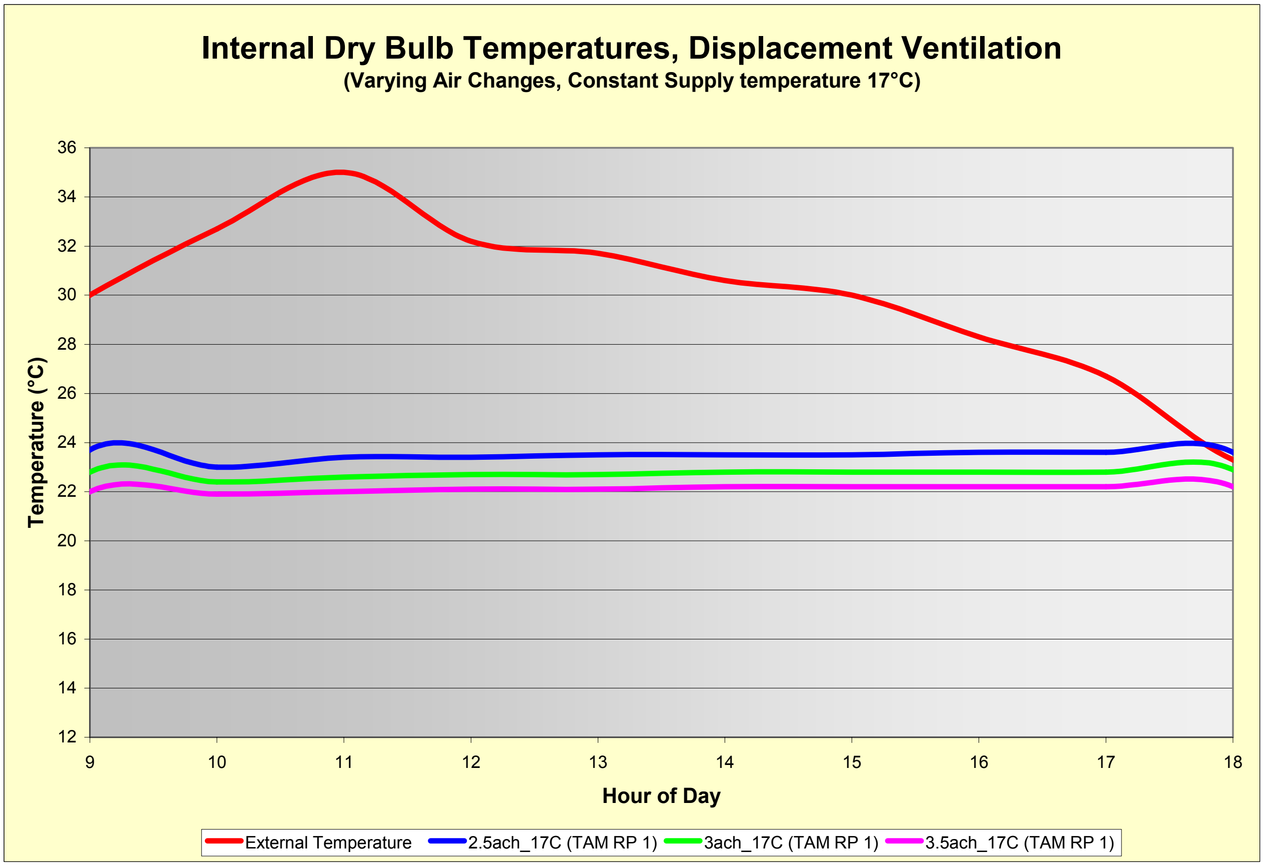

Figure 4: Internal dry bulb temperatures for a Displacement ventilation system; different volumes at constant temperature.

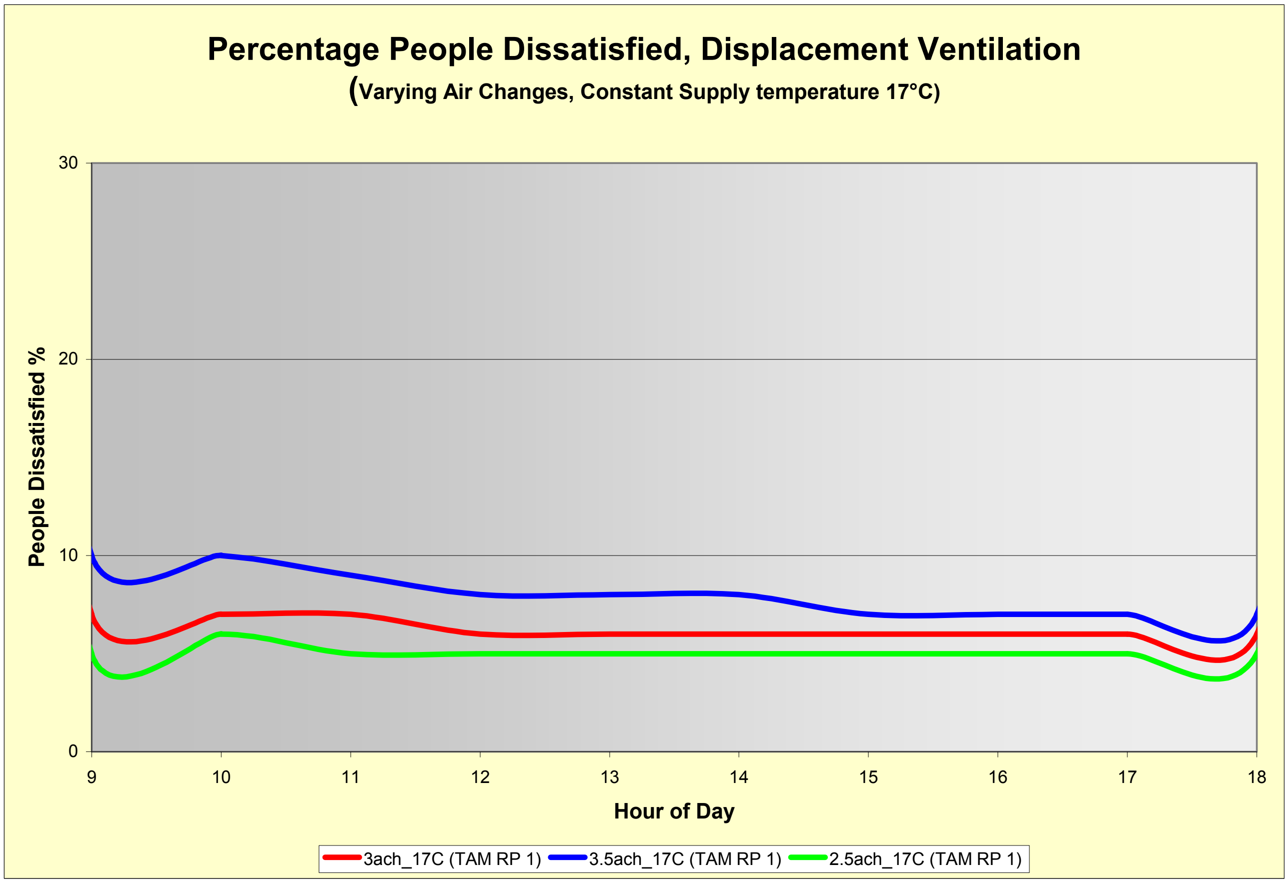

Figure 5: PPD for a Displacement Ventilation system; different supply air volumes at a constant temperature

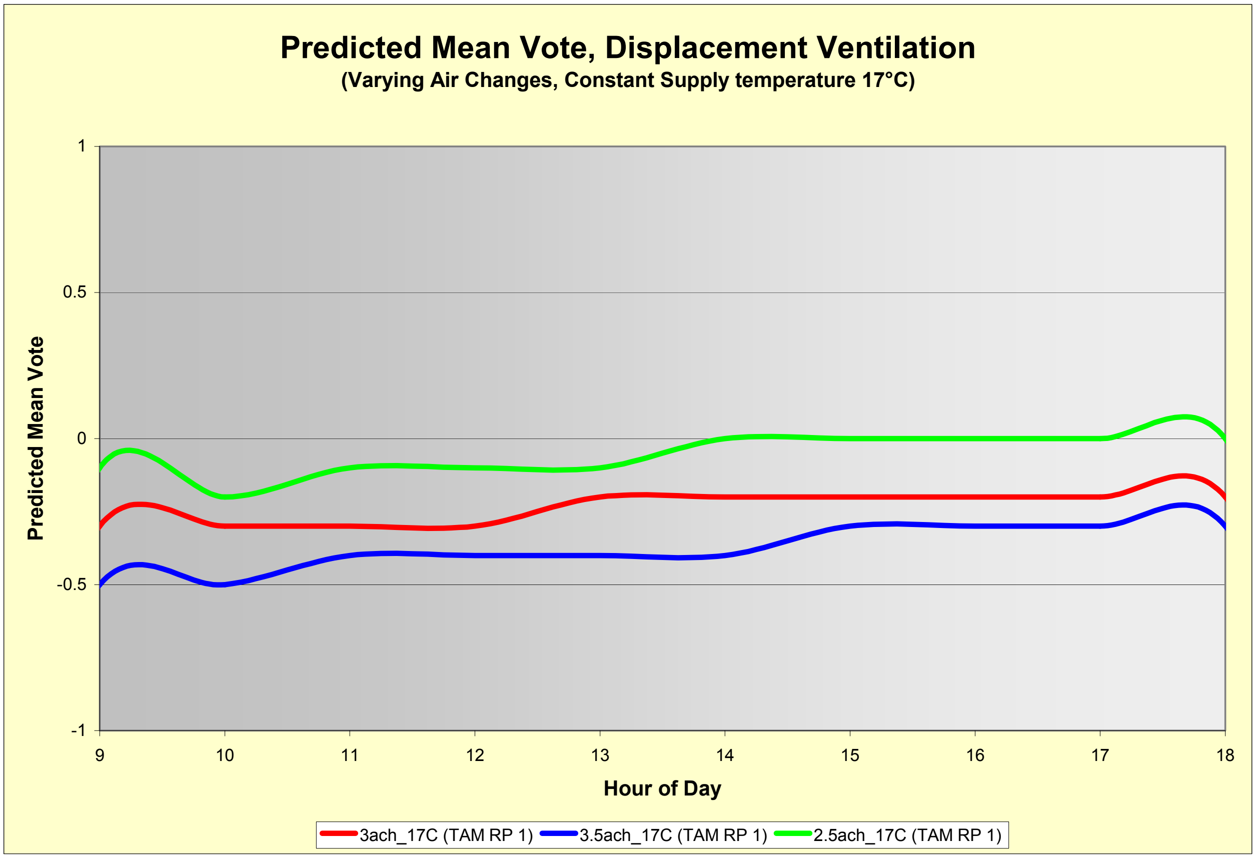

Figure 6: PMV for a Displacement Ventilation System; different supply air volumes at a constant temperature.

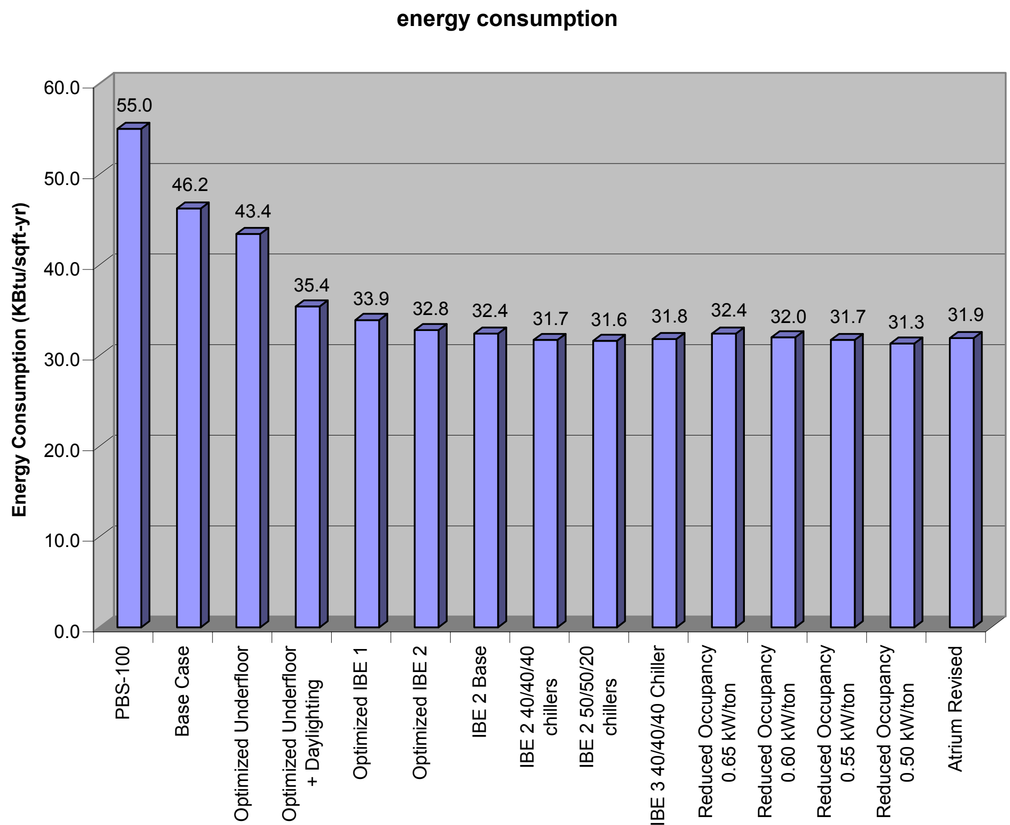

Figure 7: Annual energy Consumption for the Base Case and each Alternate.

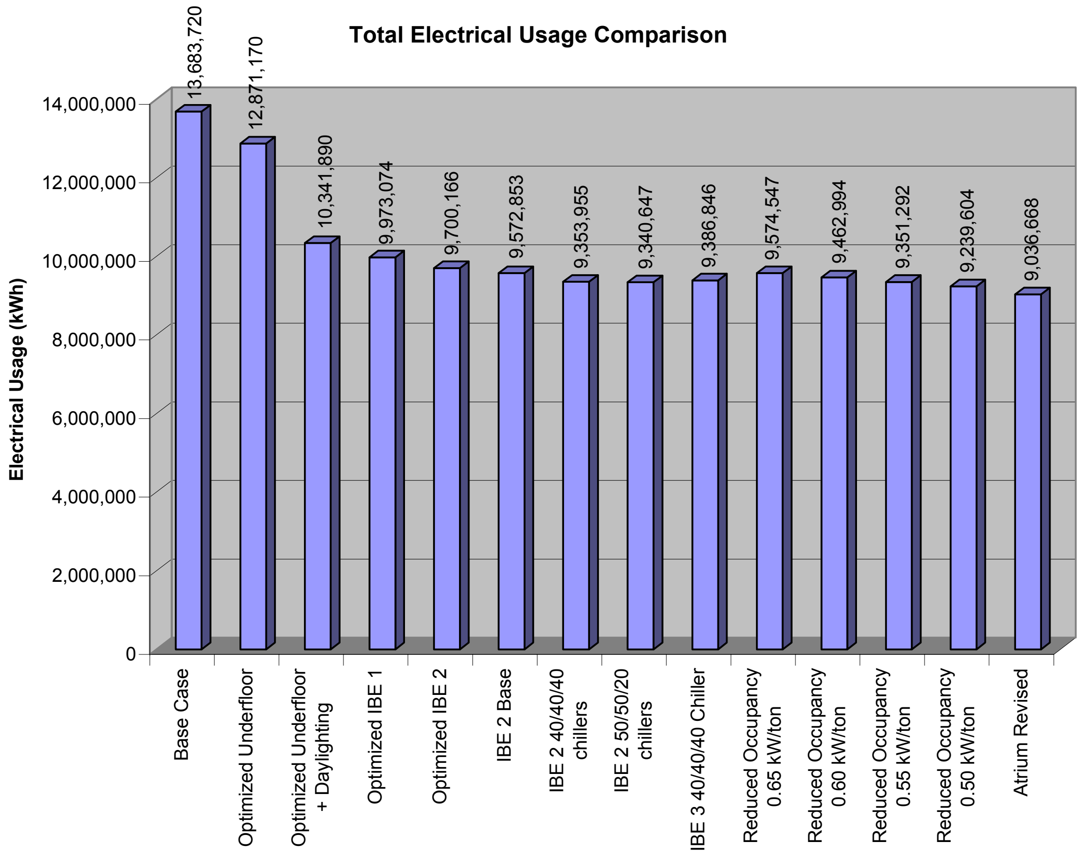

Figure 8: Total annual Energy Consumption in kWh for the Base Case and each Alternate.

Table 2: Part Load Calculations for Central Plant Part I

System Description:												
constant speed compressors, constant condenser water flow, constant primary chilled water flow, variable secondary chilled water flow.												
The condenser water is maintained at 80-90 F **under all part load conditions**. The only exception is when the threshold for min. flow through the cooler is reached.												
Chilled water temperature is maintained at 42-54 F **under all part load conditions**. The only exception is when the threshold for min. flow through the evaporator is reached.												
	= shading denotes calculated cells											
	= denotes information to be entered											
Load/Chiller=	800	Tons										
CWS=	42	F		CWS=	90	F						
CHWR=	54	F		CWR=	80	F						
Percentage of Building Peak Load	**4.7**	**6.7**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**
Total Load (tons)	112.80	160.80	240.00	480.00	720.00	960.00	1,200.00	1,440.00	1,680.00	1,920.00	2,160.00	2,400.00
Chillers												
number of chillers running	1.00	1.00	1.00	1.00	1.00	2.00	2.00	2.00	3.00	3.00	3.00	3.00
kW/ton	**1.28**	**1.06**	**0.87**	**0.65**	**0.60**	**0.65**	**0.61**	**0.60**	**0.62**	**0.61**	**0.60**	**0.58**
kW chiller	143.93	170.77	209.04	312.48	428.40	624.96	732.00	856.80	1,046.64	1,169.28	1,285.20	1,401.60
Total primary chilled water GPM	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	3,200.00	3,200.00	3,200.00	4,800.00	4,800.00	4,800.00	4,800.00
gpm/ton	14.18	9.95	6.67	3.33	2.22	3.33	2.67	2.22	2.86	2.50	2.22	2.00
CHWS	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00
CHWR	43.69	44.41	45.60	49.20	52.80	49.20	51.00	52.80	50.40	51.60	52.80	54.00
load chiller 1 tons	112.80	160.80	240.00	480.00	720.00	480.00	600.00	720.00	560.00	640.00	720.00	800.00
load chiller 2 tons	0.00	0.00	0.00	0.00	0.00	480.00	600.00	720.00	560.00	640.00	720.00	800.00
load chiller 3 tons	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	560.00	640.00	720.00	800.00
Primary Chilled Water Loops												
chiller Primary pump 1(GPM)	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00
CHWS	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00	42.00
CHWR	43.69	44.41	45.60	49.20	52.80	49.20	51.00	52.80	50.40	51.60	52.80	54.00
chiller primary pump 2 (GPM)						1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00
CHWS						42.00	42.00	42.00	42.00	42.00	42.00	42.00
CHWR						49.20	51.00	52.80	50.40	51.60	52.80	54.00
chiller primary pump 3 (GPM)									1,600.00	1,600.00	1,600.00	1,600.00
CHWS									42.00	42.00	42.00	42.00
CHWR									50.40	51.60	52.80	54.00
Total Primary GPM	1,600.00	1,600.00	1,600.00	1,600.00	1,600.00	3,200.00	3,200.00	3,200.00	4,800.00	4,800.00	4,800.00	4,800.00
chiller pump 1(kW)	25.10	25.10	25.10	25.10	25.10	25.10	25.10	25.10	25.10	25.10	25.10	25.10
chiller pump 2 (kW)						25.10	25.10	25.10	25.10	25.10	25.10	25.10
chiller pump 3 (kW)									25.10	25.10	25.10	25.10
Total primary chilled water kW	25.10	25.10	25.10	25.10	25.10	50.20	50.20	50.20	75.30	75.30	75.30	75.30
Secondary Chilled Water												
Total Secondary Chilled Water GPM	225.60	321.60	480.00	960.00	1,440.00	1,920.00	2,400.00	2,880.00	3,360.00	3,840.00	4,320.00	4,800.00
secondary pump 1 GPM	112.80	160.80	240.00	480.00	720.00	960.00	1,200.00	1,440.00	1,680.00	1,920.00	2,160.00	2,400.00
secondary pump 1(kW)	0.08	0.16	0.37	1.46	3.29	5.84	9.13	13.14	17.89	23.36	29.57	36.50
secondary pump 1 GPM	112.80	160.80	240.00	480.00	720.00	960.00	1,200.00	1,440.00	1,680.00	1,920.00	2,160.00	2,400.00
secondary pump 2 (kW)	0.08	0.16	0.37	1.46	3.29	5.84	9.13	13.14	17.89	23.36	29.57	36.50
Total secondary chilled water kW	0.16	0.33	0.73	2.92	6.57	11.68	18.25	26.28	35.77	46.72	59.13	73.00

Table 3: Part Load Calculations for Central Plant Part II

Cooling Towers												
number of cooling towers	1.00	1.00	1.00	1.00	2.00	2.00	2.00	3.00	3.00	4.00	4.00	4.00
Tower 1 Load tons	112.80	160.80	240.00	480.00	360.00	480.00	600.00	480.00	560.00	480.00	540.00	600.00
Tower 2 Load tons					360.00	480.00	600.00	480.00	560.00	480.00	540.00	600.00
Tower 3 Load tons								480.00	560.00	480.00	540.00	600.00
Tower 4 Load tons										480.00	540.00	600.00
Tower 1 GPM	270.72	385.92	576.00	1,152.00	864.00	1,152.00	1,440.00	1,152.00	1,344.00	1,152.00	1,296.00	1,440.00
Tower 2 GPM					864.00	1,152.00	1,440.00	1,152.00	1,344.00	1,152.00	1,296.00	1,440.00
Tower 3 GPM								1,152.00	1,344.00	1,152.00	1,296.00	1,440.00
Tower 4 GPM										1,152.00	1,296.00	1,440.00
tower 1 fan kW	4.79	6.83	10.20	20.40	15.30	20.40	25.50	20.40	23.80	20.40	22.95	25.50
tower 2 fan kW	0.00	0.00	0.00	0.00	15.30	20.40	25.50	20.40	23.80	20.40	22.95	25.50
tower 3 fan kW	0.00	0.00	0.00	0.00	0.00	0.00	0.00	20.40	23.80	20.40	22.95	25.50
tower 4 fan kW	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	20.40	22.95	25.50
kW/ton	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
kW Cooling Towers	4.79	6.83	10.20	20.40	30.60	40.80	51.00	61.20	71.40	81.60	91.80	102.00
Condenser Water												
GPM/pump	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00
condenser pump 1 (kW)	46.70	46.70	46.70	46.70	46.70	46.70	46.70	46.70	46.70	46.70	46.70	46.70
condenser flow temp	81.20	81.72	82.56	85.12	87.68	85.12	86.40	87.68	85.97	86.83	87.68	88.53
condenser return temp	80.00	80.00	80.00	80.00	80.00	80.00	80.00	80.00	80.00	80.00	80.00	80.00
GPM/pump						2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00
condenser pump 2 (kW)						46.70	46.70	46.70	46.70	46.70	46.70	46.70
condenser flow temp						85.12	86.40	87.68	85.97	86.83	87.68	88.53
condenser return temp						80.00	80.00	80.00	80.00	80.00	80.00	80.00
GPM/pump									2,250.00	2,250.00	2,250.00	2,250.00
condenser pump 3 (kW)									46.70	46.70	46.70	46.70
condenser flow temp									85.97	86.83	87.68	88.53
condenser return temp									80.00	80.00	80.00	80.00
Total condenser water GPM	2,250.00	2,250.00	2,250.00	2,250.00	2,250.00	4,500.00	4,500.00	4,500.00	6,750.00	6,750.00	6,750.00	6,750.00
Total condenser water kW	46.70	46.70	46.70	46.70	46.70	93.40	93.40	93.40	140.10	140.10	140.10	140.10
Pump Totals												
total pumping power kW	71.96	72.13	72.53	74.72	78.37	155.28	161.85	169.88	251.17	262.12	274.53	288.40
Total Plant Energy												
total energy kW	220.69	249.73	291.77	407.60	537.37	821.04	944.85	1,087.88	1,369.21	1,513.00	1,651.53	1,792.00
Configuration 1	1.96	1.55	1.22	0.85	0.75	0.86	0.79	0.76	0.82	0.79	0.76	0.75

Chiller Plant Electrical Consumption at Part Load										
Percentage Load **Configuration**	10% (240 Tons)	20% (480 Tons)	30% (720 Tons)	40% (960 Tons)	50% (1200 Tons)	60% (1440 Tons)	70% (1680 Tons)	80% (1920 Tons)	90% \ (2160 Tons)	**100% (2400 Tons)**
Primary/ Secondary	1.22	0.85	0.75	0.86	0.79	0.76	0.82	0.79	0.76	**0.75**
VFD Chillers Var. Condenser	1.25	0.77	0.66	0.78	0.68	0.67	0.74	0.67	0.68	**0.71**
VFD Chillers Const. Condenser	1.32	0.78	0.66	0.79	0.72	0.67	0.75	0.70	0.67	**0.68**

Table 4: Chiller Plant Energy Consumptions at Part Load Conditions