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Analysis and Estimation of Within the Desiccant and Without the Desiccant Dehumidification System Performance

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***Abstract*—In this research the heat recovery in the HVAC system of a newly constructed general hospital building was considered for the analysis and estimation of desiccants dehumidification system performance. In the previous two decades, expanded urbanization and industrialization have caused a huge arise of the energy utilization of buildings. Energy utilization via air conditioning represents 1/3 of the aggregate energy utilized by the developed society. Cooling and dehumidifying new ventilation air comprises 20– 40% of the total energy stack for cooling in hot and moisturized areas. Currently, there is developing interest for energy saving advancements in buildings, thus to this, energy efficient technologies are ending up more mainstream among researchers and developers. In this respects, to satisfy energy protection demands, improvement of advance heat or energy recuperation with energy proficient ventilation system has been focused in this study. Hourly Analysis Program (HAP) was used for key features to design HVAC system and to estimate annual energy consumption. Using HAP, annual energy estimation of cooling and heating loads were determined. The energy saved using HRWheel-1800, HRWheel-1700 and HRWheel-2000 was 39%, 28% and 19% respectively. The required air flow rate computed as 9500CFM, 7688CFM, and 7848CFM respectively.**

Index Terms—Desiccant Dehumidification System (DDS); Heating Ventilation and Air Conditioning (HVAC); Hourly Analysis Program (HAP)

I. INTRODUCTION

Pakistan is facing severe energy crises for the last two decades. The major cause for the current shortfall in electricity is increasing human population, modernization and urbanization. Electricity shortfall during 2017-18 in Pakistan, through all sources is expected around 7000 MW. Energy crisis issue will prevail for long years to come if sincere and planed efforts are not initiated promptly and urgently on national bases.

One of the high energy consumption field is the Heating ventilation and air conditioning (HVAC). Desiccant dehumidification is a process of treating the air to maintain the temperature and humidity to a comfort level. To maintain the desired temperature and humidity, a considerable amount of energy is utilized. In the last few years, urbanization and industrialization have increased to a certain extent and due to which energy consumption has risen strangely. Almost one third of society's energy consumption is utilized by air conditioning. More accurately, main portion of energy is consumed to cool and dehumidify fresh air. Hourly analysis program was used to calculate and for estimation of energy of commercial buildings as well as domestic. HAP application is a computer tool which helps engineers in scheming HVAC systems for commercial homes. HAP is a two in one tool, first it is used to comprise the different loads and second is to estimate the building energy. Many researchers have worked on the application of heat recovery wheels to overcome the energy consumption of HVAC system. They have the use of this technique under various weather conditions. Jose Fernandez-Seara et al.[1] performed many tests on the heat exchanger to check out its performance under reference conditions when integrated within the system. Duration of each experiment was four hour. Operation of heat exchanger was unsteady during the first hours but later on it became stable. Efficiency and the heat transfer remain almost constant during the whole period of time. During the experiment, pressure drop rises and this is due to the condensation on the surface of exchanger. After that they carried the number of experiments by varying the outdoor design conditions. They varied the temperature of fresh air, relative humidity of exhaust air and the flow rate at a time by keeping all other variables constant. Heat transfer decreases rapidly while efficiency of the exchanger remains almost same as the inlet temperature increases. Heat transfer and efficiency have no significant effect by the change in relative humidity. Effect on both these two was within 10%. A rapid rise in heat transfer has been seen due to the increase in flow rate but by increasing flow rate, efficiency decreases. C.-A. Roulet et al.[2] pointed that performance of a heat recovery unit can be evaluated from the worldwide effectiveness of HR unit and particular net energy saving or coefficient of performance (COP). We should consider the worldwide effectiveness of the whole framework not the nominal proficiency of the HR unit only. They performed measurements on a number of heat recovery units. They used the technique of tracer gas dilution. During their measurements, they found that SNES may be even negative. In the best case, the value of SNES was 2.7. For those systems having nominal efficiency 70%, global which is the real efficiency reduced to 43%. It's not always fruitful to use the heat recovery unit. During the measurements, they found some systems with efficiency less than 10%. Those systems used more energy than they saved. Efficiency of the system depends on the infiltration, exfiltration, internal and external recirculation rates and on parasitic leakages and shortcuts. Younness EI Fouih et al.[3] considered three different ventilation systems and evaluated their performance in seven different climatic zones in France. For this purpose they considered three different types of buildings. These categories were flat, house and a small office. They suggested that we should consider the total energy consumption consisting of ventilation, cooling and heating consumption for energy comparison. For office, HRV (heat recovery ventilation system) was less proficient than the MEV (mechanical extract ventilation system) for each climatic zone. Performance of HRV was almost same for the flat and the house case. For the flat and house, energy consumption of HRV is less than that of MEV. Energy consumption of HRV was more and less than that of HCV (humidity controlled ventilation) for climatic zones. They showed that system parameters impact the energy based performance of the system. Efficiency of the heat exchanger and the SFP (specific fan power) of the fan have the vital role on the overall energy utilization of the system. Shahram Delfani et al.[4] evaluated experimentally four different combinations of heat recovery with the ventilation system in different climatic regions of Asia. In the two systems, they used 100% fresh air, and in the two they used 30% fresh air. They considered 40 different test points for this purpose. Their experimental setup could vary the outdoor design conditions. Apparatus could calculate the temperature, humidity and velocity at different points in the system. They calculated the energy consumption of each system. Consumption of MHCC (with mixer and heat exchanger) was lowest than any other system in all the systems. Consumption of HCC (with single heat exchanger) was highest in every case of the selected points. They found that by using an extra heat exchanger, we can reduce the consumption of heating, cooling and ventilation system by almost 32%.

Yanming et al.[5] worked on the applicability of the heat recovery unit to reduce energy consumption especially in the winter season in different zones of China. They surveyed for the comfort temperature in super markets with high occupant densities. They suggested that comfort temperature is closely related to the clothing insulation of the occupants. They found that latent recovery is not suitable in winter because values of humidity ratio are much above the critical values. They showed that sensible recovery is suitable and it depends on the outdoor temperature. They divided China into five zones on the climatic basis and into three regions according to the temperature of outdoor climate. They calculated two critical temperatures for heat recovery. If we use a rotational fan with variable speed instead of constant speed, heat recovery has always a positive impact on energy saving except Guangzhou. Mohammad Rasouli et al.[6] studied the uncertainties about the energy and economic performance of an energy recovery unit due to unpredictability in different building related parameters and also due to different HVAC parameters. To carry out their research, they selected an office building in Chicago. After their research, they come to the conclusion that cooling load, heating load and the annual heating energy are more sensitive to ventilation rate. Internal loads have more effect on the annual cooling energy. They suggested that the size of HVAC equipment can be minimized by using energy recovery unit. They considered an ERV with 75% sensible and 60% latent effectiveness. They showed that this ERV can decrease the cooling and heating load by 30% and 18% respectively. This ERV normally has the period of 2 years. Envelope related parameters and internal loads have no significant effect on payback period of ERV. While with increasing ventilation rate, payback period can reduce greatly. Payback period of an ERV is strongly dependent on its own parameters which includes effectiveness, initial cost and on the parameters of HVAC equipment which includes initial cost and the efficiency mainly.

Y.P. Zhou et al.[7] developed the model of energy recovery by using Energy Plus software. They investigated its performance and its availability under different temperatures and also under different weathers. For this purpose, they considered two different cities of China i.e. Shangai and Beijing. Availability of ERV is high enough in both the cities. Its range is between 0.6-0.9 for Shangai as well as Beijing. Performance of ERV is little bit better in Shangai as compared to that in Beijing. Some cooling recovery exists even in winter season. But its value is very small as compared to the heating recovery in winter season. While increasing the set-point, energy recovery performance decreases. In the same manner, if we elevate the set-point, heating recovery also increases and if we raise the set-point, cooling recovery decreases but the change is not significant. Availability of ERV is low in summer as compared to that in winter. There is a significant amount of heating recovery in summer. In summer season, if we increase the set-point, performance of ERV also increases. Performance of ERV is better in Shangai than that in Beijing. If we increase the set-point, cooling recovery decreases and heating recovery rises. Using ERV is not useful in Beijing when set-point is higher than 24 °C. Latent recovery is the main factor for better performance of ERV in Shangai in warmer months. The present study focusses to fulfill energy conservation demands, improvement of advance energy or heat recovery with energy-efficient ventilation systems. This project is essentially an effort to express desiccant dehumidification system behavior.

II. Researcher Methodology

A. Analysis of Desiccant Dehumidification

Peak cooling coil load, design airflow CFM and the fan motor BHP were calculated for the sizing of the HVAC system. Proceeding further annual energy consumption by cooling, heating, supply fan, lighting and other electrical equipment was determined through simulation tool. Research has been carried out for the weather conditions of Lahore because Typical Metrological Year 2 file was only available for this city of Pakistan. Hourly Analysis Programmed software's was used for research. Hourly analysis application (hap) is a computer tool which helps engineers in scheming HVAC systems for commercial homes. HAP is equipment in a single. Foremost it is a tool for estimating loads and designing structures. Some other, it's far a tool for simulating constructing strength use and calculating strength of the system. In this capacity it's tons useful for plan design and detailed design energy estimations. HAP makes use of the ashrae transfer reason approach for load calculations and precise 8,760 hour-with the aid of-hour model techniques for the power research. This application is unconfined as separate, however like products. The "hap system design load" program can provide system layout and cargo estimating capabilities. The overall "hap" program offers the same system design competencies plus energy research features. Until now calculations were made without using heat recovery. Now we will incorporate heat recovery units into the system. Now all the above mentioned parameters were determined with the inclusion of heat recovery units into the system. Analysis has been carried out by using three different types of Heat Recovery units which are as follows

- HR Wheel-2000-SG-200
- HR Wheel-1700-MS-200
- HR Wheel-1800-MS-270

In the above code, HRW is the abbreviation for Heat Recovery Wheel, figures after HRW e.g. 2000, 1700 are the diameters of wheels. SG denotes for silica gel and MS for Ecosorb 300 (Molecular Sieve 3A). Figures after SG or MS e.g. 200 and 270 are the depths of wheels.

Two different types of desiccants have been used for the analysis:

- First one is the Silica Gel
- Second desiccant is the Ecosorb 300 (Molecular Sieve 3A

i. Analysis without Heat Recovery Wheel

Air system sizing summary for treated fresh unit without heat recovery

Air system information

Air system name ………………. TFU | Number of zones…… 1
Equipment class………………. CW AHU | Floor Area………….. 3484.8 ft^2
Air system Type………………. SZCAV | Location……………. Karachi, Pakistan

Sizing Calculation Information

Zone and Space Sizing Method:
Zone CFM…… Sum of Space airflow Rates | Calculation Months… Jan to Dec
Space CFM…… Individual peak space load | Sizing Data……….… Calculated

Central Cooling Coil Sizing DATA

Total coil load………………………………… 82.0 Tons | Load Occurs at………… Jun 1500
Total coil load…………………………………982.3 MBH | OA DB/WB……………..…99.0/82.0 F
Sensible Coil load…………………………..… 4822.1 MBH | Entering DB/WB………… 98.2/81.3 F
Coil CFM at Jun 1500…………………………10001 CFM | Leaving DB/WB…………. 55.4/54.9 F
Maximum block CFM……………………….10001 CFM | Coil ADP……………………….45.6 F
Sum of Peak Zone CMF……………………. 10002 CFM | Bypass Factor……….…………..0.110
Sensible Heat Ratio…………………………………. 0.487 | Resulting RH…….……….…….. 52 %
Ft^2/Ton……………………………………………….. 41.9 | Design Supply Temp……….…. 56.0 F
BTU/(hr-ft^2) ……………………….………….. 287.5 | Zone T- stat check………… 1 of 1 OK
Water flow@ 10.0 F rise……………………… 195.58 gpm | Max zone Temp Deviation…….. 0.0 F

Central Heating Coil Sizing

Max coil load………………………………… 226.8 MBH | Load occurs at………………. Des Htg
Coil CFM at Des Htg………………………… 10002 CFM | BTU/(hr-ft2) …………………… 66.7
Water flow@ 20.0 F rise………………………. 22.59 gpm | Ent. DB/Lvg DB………... 50.4/71.3 F

Table 1 Design Cooling System Data

component	location	dbt (f)	airflow (cfm)	sp. humidity	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	98.0	9501	0.1985	404	255791	475754
vent – return mixing	outlet	99.8	10001	0.01924	400	-	-
supply fan	outlet	97.2	10002	0.01924	400	4548	-
central cooling coil	outlet	54.5	10000	0.00869	405	472085	500271
central heating coil	outlet	55.9	10001	0.00869	404	26445	-
cold supply duct	outlet	58.7	9501	0.00869	405	-	-
return plenum	outlet	77.0	9501	0.00924	478	0	-
duct leakage air	outlet	55.9	501	0.00869	405	-	-
return duct	outlet	74.5	10005	0.00921	405	-	-

Table 2 Design Heating System Data

component	location	dry bulb temp (f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	48.9	0.00365	9501	404	-196168	0
return-vent mixing	outlet	49.9	0.00365	10001	405	-	-
supply fan	outlet	50.5	0.00365	10002	401	4449	-
central cooling coil	outlet	50.6	0.00365	10001	400	0	0
central heating coil	outlet	71.5	0.00365	10002	405	225675	-
cold supply duct	outlet	71.2	0.00366	9501	401	-	-
return plenum	outlet	67.9	0.00365	9501	405	0	-
duct leakage air	outlet	71.5	0.00365	505	405	-	-
return duct	outlet	68.4	0.00366	10001	402	-	-

ii. Analysis with Heat Recovery Wheel

Air system sizing summary for treated fresh unit with heat recovery

Air system information

Air system name ………………. TFU
Equipment class………………. CW AHU
Air system Type………………. SZCAV
Number of zones…… 1
Floor Area………….. 338.8 ft^2
Location……………. Karachi, Pakistan

Sizing Calculation Information

Zone and Space Sizing Method:
Zone CFM…… Sum of Space airflow Rates
Space CFM…… Individual peak space load
Calculation Months… Jan to Dec
Sizing Data……….… Calculated

Central Cooling Coil Sizing DATA

Total coil load………………………………… 46.9 Tons
Total coil load…………………………………972.5 MBH
Sensible Coil load…………………………..… 482.1 MBH
Coil CFM at Jun 1500…………………………10001 CFM
Maximum block CFM……………………….10001 CFM
Sum of Peak Zone CMF……………………. 10001 CFM
Sensible Heat Ratio…………………………………. 0.488
Ft^2/Ton……………………………………………….. 41.9
BTU/(hr-ft^2) ……………………………..………….. 287.5
Water flow@ 10.0 F rise……………………… 194.58 gpm
Load Occurs at……………… Jun 1500
OA DB/WB……………..…98.0/82.0 F
Entering DB/WB………… 98.5/81.4 F
Leaving DB/WB…………. 54.5/54.0 F
Coil ADP………………………46.7 F
Bypass Factor……….…………..0.110
Resulting RH….…………..…….. 50 %
Design Supply Temp………..… 55.3 F
Zone T- stat check………… 1 of 1 OK
Max zone Temp Deviation…….. 0.0 F

Central Heating Coil Sizing Data

Max coil load………………………………… 225.8 MBH
Coil CFM at Des Htg………………………… 10001 CFM
Water flow@ 20.0 F rise………………………. 22.59 gpm
Load occurs at………………. Des Htg
BTU/(hr-ft2) …………………… 66.8
Ent. DB/Lvg DB………... 50.5/71.4 F

Supply Fan Sizing Data

Actual Max CFM ………………...………. 10001 CFM
Standard CFM………………………………. 9995 CFM
Actual Max CFM/ft^2 ……………….…..……. 10002 CFM/ft^2
Fan Motor BHP…..………. 1.76 BHP
Fan Motor BHP…………..… 1.3 KW
Fan Static……………….. 0.61 in wg

a. *Analysis with HRW-2000-SG-200*

Table 3 Design Cooling System Data

component	location	dry bulb temp (f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	82.3	0.01677	7848	400	70414	283239
return- vent mixing	outlet	81.9	0.01639	8261	404	-	-
supply fan	outlet	82.3	0.01639	8261	404	3675	-
central cooling coil	outlet	53.6	0.00861	8261	404	256119	304952
central heating coil	outlet	57.6	0.00861	8261	404	35638	-
cold supply duct	outlet	58.4	0.00861	7848	404	-	-
return plenum	outlet	74.9	.00919	7848	493	0	-
duct leakage air	outlet	57.6	.00861	413	404	-	-
return duct	outlet	74.0	0.00916	8261	488	-	-

Table 4 Design Heating System Data

component	location	dry bulb temp (^{0}f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	68.0	0.00817	7849	401	8472	33675
return-vent mixing	outlet	68.0	.00813	8262	401	-	-
supply fan	outlet	68.4	0.00813	8262	401	385	-
central cooling coil	outlet	50.5	0.00725	8262	401	16949	3456
central heating coil	outlet	68.1	0.00725	8263	402	15734	-

cold supply duct	outlet	68.2	0.00725	7849	401	-	-
return plenum	outlet	68.3	0.00727	7849	402	0	-
duct leakage air	outlet	68.4	0.00725	414	401	-	-
return duct	outlet	68.5	0.00727	8262	405	-	-

b. *Analysis with HRW-1700-MS-200*

Table 5 Design Cooling System Data

component	location	dry bulb temp (f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	82.5	0.01749	7849	401	70414	309739
return-vent mixing	outlet	81.4	0.01707	8262	402	-	-
supply fan	outlet	82.5	0.01707	8262	403	3675	-
central cooling coil	outlet	53.5	0.00860	8262	405	258775	332155
central heating coil	outlet	57.7	0.00860	8262	405	38297	-
cold supply duct	outlet	58.5	0.00860	7849	404	-	-
return plenum	outlet	74.8	0.00921	7849	495	0	-
duct leakage air	outlet	57.7	0.00858	414	405	-	-
return duct	outlet	74.1	0.00917	8262	488	-	-

Table 6 Design Heating System Data

component	location	dry bulb temp (f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	69.0	0.00786	7848	400	8472	22373
return-vent mixing	outlet	69.0	.00783	7848	400	-	-
supply fan	outlet	69.4	0.00783	8261	400	3675	-
central cooling coil	outlet	50.0	0.00724	8261	400	168286	22963
central heating coil	outlet	68.0	0.00724	8261	400	156139	-
cold supply duct	outlet	68.0	0.00724	7848	400	-	-
return plenum	outlet	68.0	0.00726	7848	401	0	-
duct leakage air	outlet	68.0	0.00724	413	400	-	-
return duct	outlet	68.0	0.00726	8261	401	-	-

c. *Analysis with HRW-1800-MS-270*

Table 7 Design Cooling System Data

component	location	dry bulb temp (f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	80.5	0.01550	7688	400	56726	231541
return-vent mixing	outlet	80.5	0.01518	8092	405	-	-
supply fan	outlet	80.5	0.01518	8092	405	3599	-
central cooling coil	outlet	53.90	0.00862	8092	405	235713	252017
central heating coil	outlet	57.6	0.00862	8092	405	32835	-
cold supply duct	outlet	58.5	0.00862	7688	405	-	-
return plenum	outlet	74.8	0.00918	7688	495	0	-
duct leakage air	outlet	57.6	0.00862	405	405	-	-
return duct	outlet	74.0	0.00915	8092	490	-	-

Table 8 Design Heating System Data

component	location	dry bulb temp (f)	specific humidity (lb /lb.)	air flow (cfm)	co_2 level (ppm)	s. heat (btu/hr.)	latent heat (btu/hr.)
air ventilation	inlet	70.9	0.00853	7688	400	24067	46436
return-vent mixing	outlet	70.8	0.00847	8092	400	-	-
supply fan	outlet	71.2	0.00847	8092	400	3599	-
central cooling coil	outlet	50.4	0.00723	8092	400	181690	47688
central heating coil	outlet	68.0	0.00723	8092	400	154024	-
cold supply duct	outlet	68.0	0.00723	7688	400	-	-
return plenum	outlet	68.0	0.00726	7688	401	0	-
duct leakage air	outlet	68.0	0.00723	405	400	-	-
return duct	outlet	68.0	0.00726	8092	401	-	-

III. Results And Discussions

A. Cooling Load

Cooling load of the system without heat recovery was 81 tons as mentioned in research methodology. While inclusion of recovery units into the system, reduces the load of the cooling coil. Cooling load reduces to 49.2 tons by using 1700-MS-200 recovery unit. With the use of 2000-SG-200 unit, cooling load reduces to 46.8 tons. Cooling load reduces to least i.e. 40.6 tons by using 1800-MS-270.

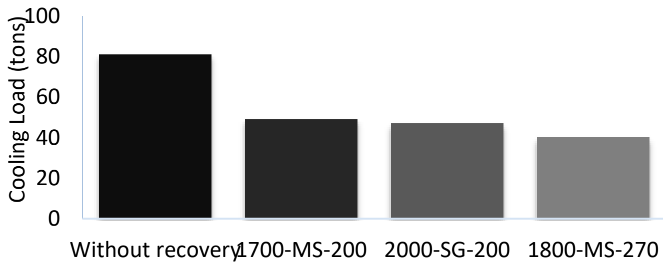

Figure 1 Cooling Load Comparison

B. Design Airflow

Design CFM of air required to maintain a better indoor air quality was 9500 without using recovery unit as shown in the figure 2. If we use recovery unit 1700-MS-200, CFM reduces to 7848. 2000-SG-200 yields the same value. While CFM of required air reduces down to 7688 by the use of 1800-MS-270.

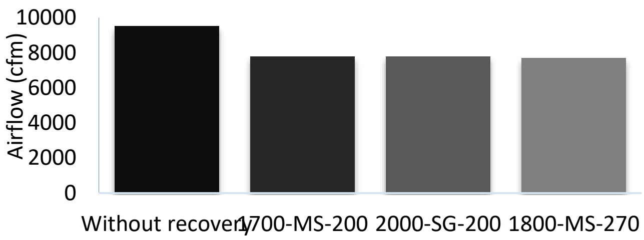

Figure 2 Design Airflow Comparisons

C. Fan Motor Sizing

Fan motor required for the system without recovery unit was of 1.75 BHP as shown in the graph below. Motor size reduces to 1.44 BHP by using 1700-MS-200. When the recovery unit 2000-SG-200 was incorporated into the system, fan motor of same BHP was required. Use of recovery unit 1800-MS-270 gives the minimum value of fan motor BP i.e. 1.41.

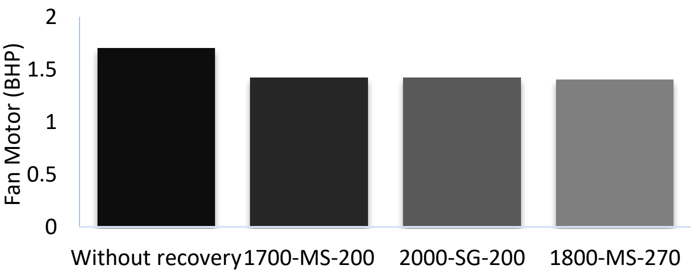

Figure 3 Fan Motor Sizing Comparisons

D. Annual Energy Consumption by Cooling Coil

Energy that will be consumed throughout the year by the cooling coil of the system without heat recovery unit will be 1286.3 MBTU as shown in the figure 4. While annual energy consumption by cooling coil is 1071.4 MBTU by 1700-MS-200 as well as by 2000-SG-200. System using 1800-MS-270 recovery unit gives less energy consumption i.e. 1051 MBTU.

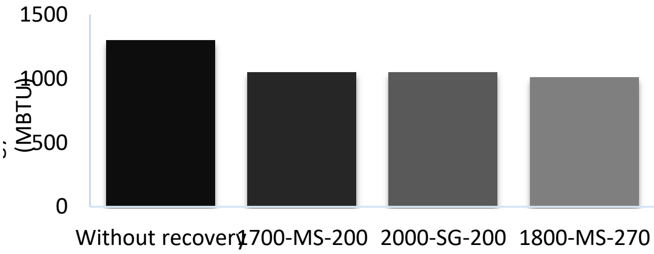

Figure 4 Annual Energy Consumption by Cooling Coil Comparison

E. Annual Energy Consumption by Heating Coil

Energy that will be consumed throughout the year by the heating coil of the system without heat recovery unit will be 184.5 MBTU as shown in the figure 5. While annual energy consumption by heating coil is 115.8 MBTU by 1700-MS-200 as well as by 2000-SG-200. System using 1800-MS-270 recovery unit gives less energy consumption i.e. 108.5 MBTU.

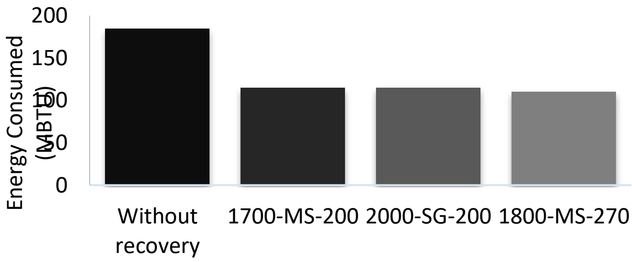

Figure 5 Annual Energy Consumption by Heating Coil Comparison

F. Annual Energy Consumption by Supply Fan

Energy that will be consumed throughout the year by supply fans of the system without heat recovery unit will be 3797 kWh as shown in the figure 6. While annual energy consumption of supply fans is 3163 kWh by 1700-MS-200 as well as by 2000-SG-200. System using 1800-MS-270 recovery unit gives less energy consumption i.e. 3101 kWh.

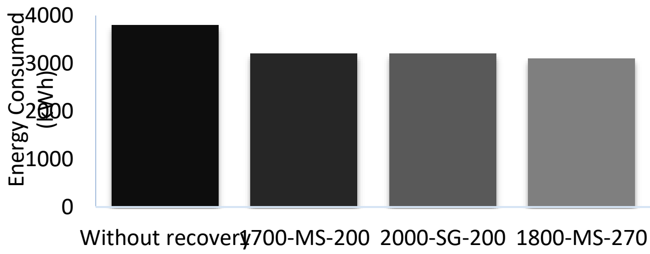

Figure 6 Annual Energy Consumption by Supply Fan Comparison

IV. Conclusion

- Maximum heat recovery is possible by using HR Wheel-1800-MS-270
- Cooling load reduces by 39% with the use of HR Wheel-1800-MS-270. While by using
  - HR Wheel-1700-MS-200 cooling load decreases only by 30% and by using
  - HR Wheel-2000-SG-200 cooling load decreases by 33%
- In reduction of designed CFM of airflow, HR Wheel-1800-MS-270 shows the best performance. CFM decreases by 19% by using HR Wheel-1800-MS-270. Both
  - HR Wheel-1700-MS-200 and HRWheel-2000-SG-200 decreases CFM equally by 17%
- BHP of fan motor is decreased by 19% by the use of HRWheel-1800-MS-270. While both
  - HRWheel-1700-MS-200 and HRWheel-2000-SG-200 decreases BHP of fan motor equally by 18%
- HRWheel-1800-MS-270 reduces the annual energy consumption of cooling coil by 18%. Reduction in annual energy consumption is 17% by using either HRWheel-1700-MS-200 or HRWheel-2000-SG-200
- Reduction in annual energy consumption of heating coil is 41% if we incorporate HRWheel-1800-MS-270 into the system. This reduction minimizes to 37% by using HRWheel-1700-MS-200 or HRWheel-2000-SG-200
- HRWheel-1800-MS-270 reduces the annual energy consumption of supply fans by 18%. Annual energy consumption is decreased by 17% by using either HRWheel-1700-MS-200 or HRWheel-2000-SG-200.

V. Nomenclature

BTU British thermal unit
CFM Cubic Feet per Minute
KWh Kilo Watt Hour
TR Ton of Refrigeration
DHRW Desiccant Heat Recovery Wheel
DB Dry Bulb Temperature
WB Wet Bulb Temperature
RH Relative Humidity
1Unit of Energy = 1KWh
TCCL Total Cooling Coil Load
THCL Total Heating Coil Load
DDs Desiccant Dehumidification System
BHP Brake Horse Power
SG Silica Gel
EMS Ecosorb Molecular Sieve

VI. Acknowledgement

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