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Analysis of Energy Savings in a Clean Room Air-conditioning System

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Abstract: To address the issue of the substantial energy cost and operating cost of an all-return air system for a clean room, we changed the former system to a 2nd return air system. With the newest building energy simulation program, Energy Plus, we simulated and compared the summer energy consuming conditions of the two systems. Results prove the superiority of the 2nd return air system, and the validity of the simulation. Also, the air system energy performance in summer was illustrated with typical meteorological hour-to-hour data.

Key words: clean room; a typical meteorological year; Energy Plus; all return air system; 2nd return air system

1 INTRODUCTION

With the rapid development in high-tech field of our country, such as electronic and pharmacy, the application of cleaning technology became much broader. Because of the large ventilation airflow rate of clean room, strict control of temperature, humidity and pressure, the constructive cost and energy cost for per square meter area are much higher than traditional air conditioning room. According to document [1], the energy cost per square meter of clean room is about 10~30 times greater than common air conditioning building. Therefore, designers and owners pay much more attention to the energy cost condition of clean room air conditioning system. In the autumn of 2002, committed by a pharmacy company, we alternated the existed air conditioning system in its pharmacy clean room for energy conserving consideration, and then analyzed the energy cost condition after renovation in summer.

Nowadays, with the computer technology's rapid development, computer simulation technology has become an important component of building pyrology, and a batch of simulation programs applying to building energy research were developed in succession. Here we use the energy simulation program, Energy Plus, to analyze the related energy cost of both the original and the alternated air conditioning system used in the pharmacy clean room, and also to simulate periodically the energy cost performance curve in summer with a typical meteorological year data.

2 SCHEME ANALYSIS

2.1Background and Goals

Now there are 24 air conditioning systems in the company, all of which are all return air system. To meet the cleaning request of the pharmacy clean room, the airflow rate of the AHU is much larger than that required by removing excess-heat and excess-humid. Under the summer operating condition, plenty of return air must be cooled and humidified first, and then heated to control the supply air temperature difference, which will cause great energy waste. At the same time, in 2003, the coal boiler of the company will be replaced by natural gas boiler, whose price is much higher, and also, the price of water and electricity will cause the suddenly rise in vapor cost. We carried out the energy conserving renovation of the AC system on principle of economy and using energy rationally.

2.2 Comparison of Several Renovation Schemes

To conserve energy of air conditioning during the operation, there were three renovation schemes.

1) Raising the temperature difference between supply air and return air and reducing the supply airflow rate: This method was usually applied to

ordinary comfortable air-condition system whose temperature and humidity required was not accurately controlled. For the specificity of techniques in industry clean room, its temperature and humidity accuracy was strictly controlled, thus this method was not applicable.

2) Fan rotation speed control: to control with tranceduce motor was an advanced method for airflow rate adjustment. The combination of tranceduce speed regulator with computer control had good performance in intelligent speed regulator, which performed well in electricity saving. However, because of the 24 hours continuously production in the clean room, the heat source, such as staff, equipment, light etc. was almost stable. The cooling load of the air supply area in the room was almost invariable throughout the year [2]. So the applying of tranceduce fan in AHU, through which to adjust fan rotate speed then to control airflow rate according to different operating condition, had no obvious energy saving effect in clean room.

3) 2nd return air system: 2nd return air pattern was usually used in air-condition system which requested symmetrical indoor temperature field, small supply air temperature difference, large airflow, but no reheater. As the design airflow rate of air conditioning system for cleaning mainly considered to meet the need of the cleanliness class, its air exchange rate was much higher than that needed to remove excess-heat and excess-humid. If all return air system was adopted, the large amount of return air, which reached the GMP standard could not be used directly, but had been cooled first, then heated, dehumidified or humidified etc. Only after these handling processes, it could be reused, which caused great energy waste. After changing the original AHU to 2nd return air system, some proportion of 2nd return air is mixed with 1st return air and fresh air after dehumidified in air cooler, so not only the cooling load of the reheater, but also some cooling load was saved because of the reduction of airflow rate through air cooler [2].

Based on the fact of the clean room and the analysis of the several scheme above, 2nd return air system was selected to improve the air conditioning systems. However, during the actual adjustment process, there were some deviations in the proportion of the 2nd return air and the theoretical calculation. Also, the actual status point after mixed with the 2nd return air didn't reach the supply air status point, so some reheating was still needed in summer to reheat the supply air to ensure the temperature accuracy of the supply air.

3 THEORETICAL ANALYSIS

Next, we analyzed the all-return air system and the 2nd return air system theoretically. Usually clean room was located in the interior of the building, using heat preservation materials like steel panel, so the heat transferred through the enclosure was not very much. Indoor cooling load was mainly made up of person thermal load, equipment, light etc. The indoor heat source and the heat generated by fan cause were a constant cooling load throughout the year. At the same time, because of the low person density in clean room, the latent heat, which caused the variety of humidity content, was considered very small. The whole cooling load of the clean room could be considered approximately to be sensible heat load, namely $\varepsilon \approx +\infty$.

3.1 Energy Cost Analysis of All Return Air System

Below was the all-return air handling process in summer.

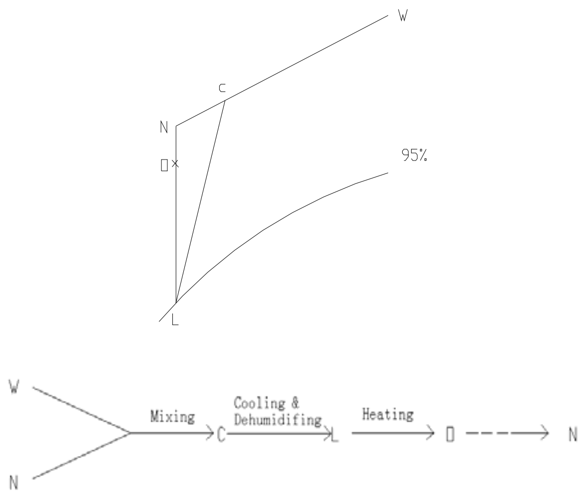

Fig.1 Air handling process of all return air system

Where point W was outside air status point, point N was clean room inside air status point, point C was the mixed point of all return air and outside air, point O was air conditioning point, Point L was the apparatus dew point of the air cooler in AHU, (relative humidity is 95%).

3.2 Energy Cost Analysis of 2nd Return Air System

The 2nd return air handling process in the summer was listed below.

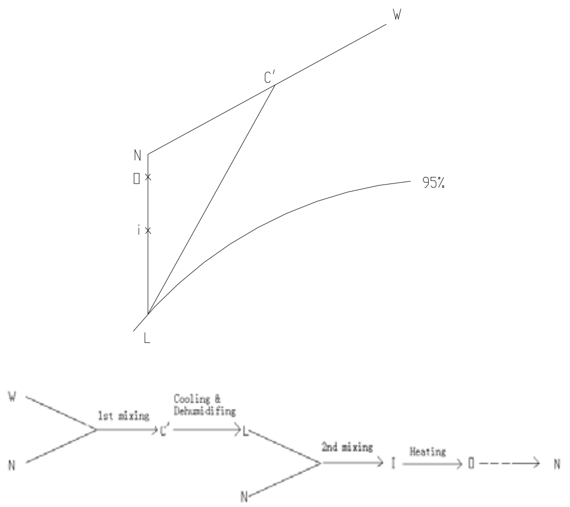

Fig.2 Air handling process of 2nd return air system

Where point W, N, L, O was the same to the all return air system, Point C' was the mixed point of 1st return air and outside air, point I was the mixed point of 2nd return air and the mixed air of 1st return air and fresh air.

3.3 Theoretical Calculation

The parameter of the status point in the i-d diagram above was assumed as below:

Outside design parameter in summer（point W）was dry-bulb temperature 33.4℃, enthalpy value 84.5kJ/kg.

Inside design parameter（point N）was dry-bulb temperature 21℃, enthalpy value 43 kJ/kg

Design air supply point parameter（Point O）was dry-bulb temperature 19℃, enthalpy value 41 kJ/kg

Apparatus dew point （point L）was 13℃, relative humidity 95%

The cooling load demanded of 2nd return system by air condition equipment was Q_0'. The cooling load demanded of all-return system was Q_0.

The reheating load of all-return system was Q_1. The reheating load of 2nd return system was Q_1'.

According to the parameters of the status point data above, we took 15% as fresh air percentage; assumed the 2nd return air percentage is t%,

Then through calculation [3], below was acquired,

$Q_0 = 14\ G, Kw$

$Q_0' = (13.725-7.5t\%)G, Kw$

$Q_1 = 5.5G, Kw$

$Q_1' = (5.5-7.5t\%)G\ , Kw$

It was found that, with the outside and inside design parameter in summer above, $t\% < 1$, so $Q_0' < Q_0$, $Q1' < Q_1$. Whatever cooling load or heating load demanded was considered, 2nd return air system was better than all return air system in energy efficiency. Furthermore, when $Q_1' = 0$,which means $t\% = 73\%$, the best point of the 2nd return air proportion appeared, which means that the mixed point of 2nd return air and the mixed air of 1st return air was just the point of supply air, reheating load was zero.

4. ENERGY PLUS SIMULATION

According to theoretical analysis and calculation above, we had chosen a representative air-handling unit for the testing renovation of 2nd return air system. Cleaning area for this AHU was a capsule clean room with a hundred thousand cleanliness classes. Indoor controlled dry-bulb temperature was 21±3℃, relative humidity was 55±5%, total cooling load for controlled area was 26KW, and design air exchange rate was 20 times per hour. Total supply air rate was kept constant 3900 m^3/h before and after renovation, and the fresh air accounted for 15%, which was 6000m^3/h to meet personnel hygienic and cleanliness class requirement. After renovation, primary return air rate was 17500 m^3/h and secondary return air rate was15500 m^3/h, accounting for 45% and 40% of total supply air rate individually. Energy cost simulating analysis on all and 2nd return air system under design meteorological condition was carried out with energy cost analysis software, Energy Plus, based on actual performance before and after renovation.

4.1 Model Establishment

Energy Plus was a multi-functional software used for energy cost analysis, lately developed by

Lawrence Berkeley National Laboratory and subsidized by American energy department. Whose calculation theory used was founded on unsteadily diathermanous theory, using reflection coefficient method to figure out dynamic load of building and relating HVAC system equipment.

Fig.3 bellow showed Energy Plus system structure. Figure 4 and 5 shows airflow joint chart of the all return air system and the 2nd return air system and main air handling equipments for renovation separately.

Energy Plus combines regional heat and quantity balance simulation with that of HVAC system of the building, and then figure out energy cost of the building in relevant meteorological condition through reaction and comparative analysis on regional controlled parameters (regional temperature and humidity) and HVAC system parameters related (supply air state, system load).

In this section, we chose outside calculating dry-bulb and wet-bulb temperature for summer air conditioning in Tianjin as outdoor design parameters. (Dry-bulb temperature 33.4 ℃, calculating daily range 8.1℃, relative humidity 55%.)

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Table1 showed hour-to-hour load for cooling and heating coils in all return air system and 2nd return air system described above under design meteorological condition. It can be found that cooling load fluctuated as outside dry-bulb temperature varying under ideal simulating condition (cooling and indoor return air, and apparatus dew point), which means this load was affected mainly by fresh air load. Heating load almost kept invariable.

Energy cost comparison of the two return air system showed 2nd return air system cost was much less than that of all return air system, whether for cooling coil load or heating coil load. Average load in a whole day for cooling coil of 2nd return air system was 112.4 KW and heating coil 21.3 KW, corresponding to 64.8% and 38.5% of all air returning system.

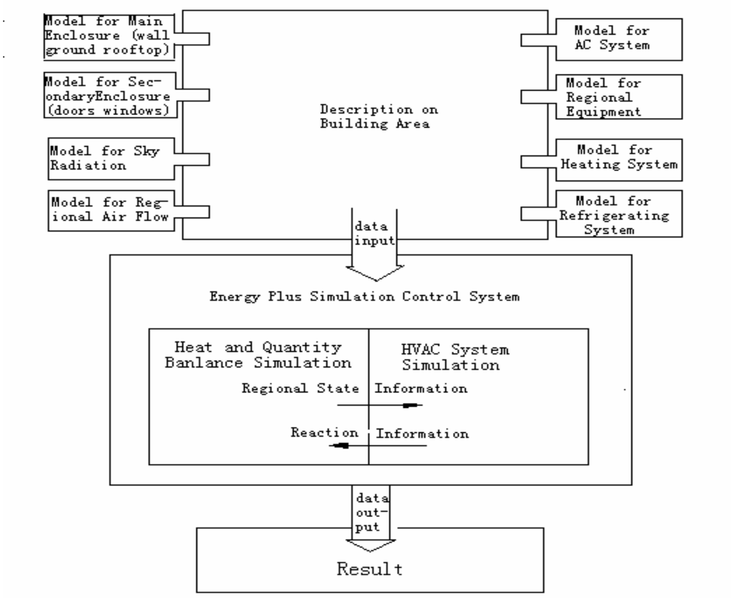

Fig.3 Energy plus system structure

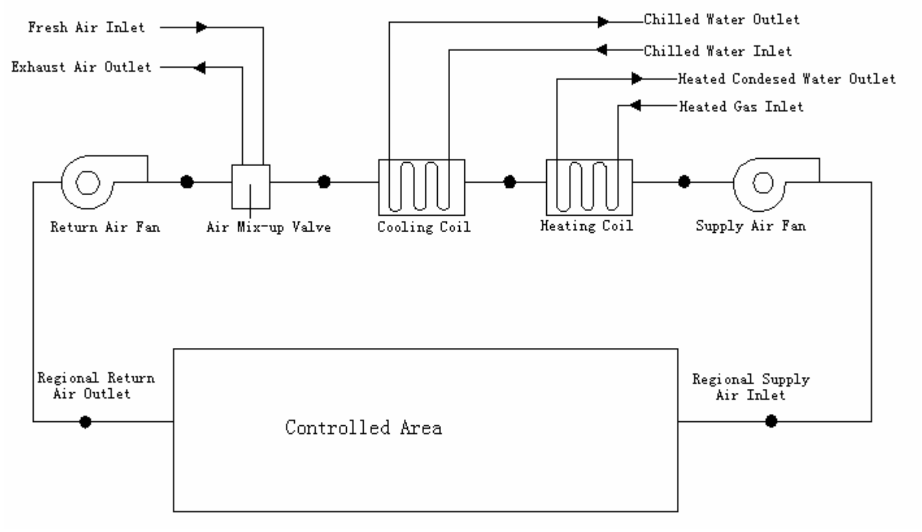

Fig.4 Air flow joints chart for all return air system

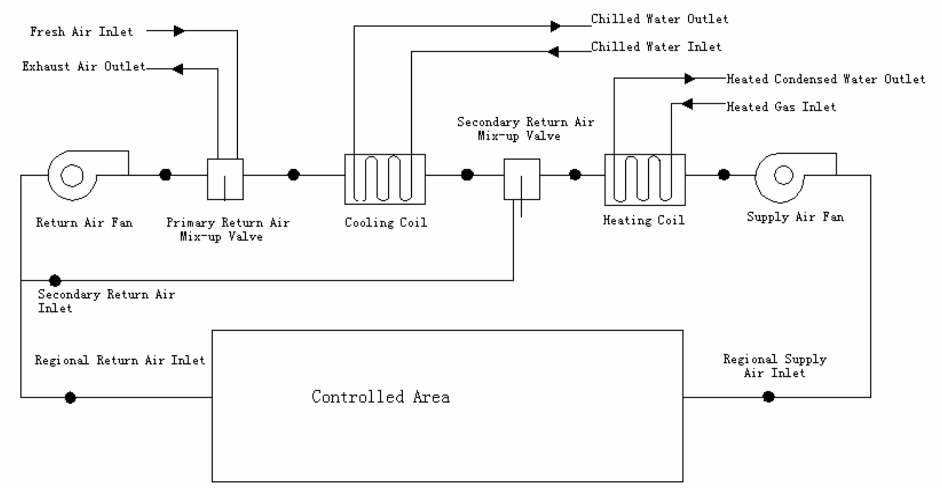

Fig.5 Air flow joints chart for 2nd return air system

Tab 1. Hour-to-hour load for all return air system and 2nd return air system under design meteorological condition

Time	Dry-bulb Temperature Outdoors (℃)	Dry-bulb Temperature Controlled Indoors (℃)	Relative Humidity Indoors (%)	All Return Air System		2nd Return Air System	
				Load for Cooling Coil (KW)	Load for Heating Coil (KW)	Load for Cooling Coil (KW)	Load for Heating Coil (KW)
1:00:00	24.8	21.0	57.3	167.6	55.3	108.0	21.3
2:00:00	24.5	21.0	57.3	166.8	55.3	107.5	21.3
3:00:00	24.2	21.0	57.3	166.2	55.3	107.0	21.3
4:00:00	24.0	21.0	57.3	165.9	55.3	106.8	21.3
5:00:00	24.1	21.0	57.3	166.0	55.3	106.9	21.3
6:00:00	24.4	21.0	57.3	166.7	55.3	107.3	21.3
7:00:00	25.0	21.0	57.3	168.0	55.3	108.3	21.3
8:00:00	25.9	21.0	57.3	170.0	55.3	109.8	21.3

9:00:00	27.0	21.0	57.3	172.5	55.3	111.6	21.3
10:00:00	28.3	21.0	57.3	175.3	55.3	113.8	21.3
11:00:00	29.6	21.0	57.3	178.2	55.3	115.9	21.3
12:00:00	30.7	21.0	57.3	180.6	55.3	117.7	21.3
13:00:00	31.5	21.0	57.3	182.3	55.3	119.0	21.3
14:00:00	31.9	21.0	57.3	183.2	55.3	119.7	21.3
15:00:00	31.9	21.0	57.3	183.1	55.3	119.6	21.3
16:00:00	31.4	21.0	57.3	182.1	55.3	118.9	21.3
17:00:00	30.7	21.0	57.3	180.5	55.3	117.7	21.3
18:00:00	29.7	21.0	57.3	178.3	55.3	116.1	21.3
19:00:00	28.7	21.0	57.3	176.1	55.3	114.3	21.3
20:00:00	27.7	21.0	57.3	174.0	55.3	112.8	21.3
21:00:00	26.9	21.0	57.3	172.1	55.3	111.4	21.3
22:00:00	26.2	21.0	57.3	170.6	55.3	110.3	21.3
23:00:00	25.6	21.0	57.3	169.4	55.3	109.3	21.3
0:00:00	25.2	21.0	57.3	168.4	55.3	108.6	21.3
Average Value in a Whole Day	27.5	21	57.3	173.5	55.3	112.4	21.3

Tab 2. Actual operation load for 2nd return air system in summer

	All Return Air System			2nd Return Air System		
	Theoretical Value	Energy Plus	Deviation	Measured Value	Energy Plus	Deviation
Load for Cooling Coil (KW)	186.7	173.5	-7.07%	122	112.5	-7.79%
Load for Heating Coil (KW)	73.3	55.3	-24.56%	32.5	21.3	-34.46%

Tab3. Room temperature and humidity before and after renovation

Room NO.		Temperature			Humidity		
		Controlled Precision	Temperate Fluctuate Range	Temperate Outrange Rate	Controlled Precision	Humidity Fluctuate Range	Humidity Outrange Rate
Room 1.	All Return Air System	18-24℃	19-27℃	14.3 %	45-65%	31-74%	+16.3%—-16.3%
	2nd Return Air System	18-24℃	19-25℃	4.7 %	45-65%	47-92%	+49.1%
Room 2.	All Return Air System	18-24℃	17-24℃	-4.7 %	45-65%	33-65%	+21.8%
	2nd Return Air System	18-24℃	18-22℃	0	45-65%	43-69%	+7.2%—-3.6%

4.2 Practical Measurement Result Testing

In order to test the accuracy and credibility of the simulating results and evaluate energy conserving effect after renovation, a whole week's following measurement was carried out for cold and heat source supplied to this air conditioning system in the summer of 2003, when refrigeration started to run. Table2 shows sorted values of the average load in a whole day for cooling coil and heating coil. Because renovation was in October 2002, for the influences of season, no data of practical measurement on summer energy cost for all return air system. Analysis here was founded on theoretically calculation according to i-d diagram. All data were listed in table 2.

Comparing on load of cooling and heating coil for a same air conditioning system showed relatively similar result (7%-8%) between Energy Plus simulating load and practical (theoretical) load for cooling coil, while much different results (20-35%) for heating coil. However, considering influences of various factors in operating, e.g. temperature rise of fan, heat exchange efficiency between coils and air, irreversible entropy increment in the process of primary and secondary return air mixture, etc., simulating results almost correspond to the fact in both data and changing tendency, so it provided great reference and help to energy cost study on air conditioning, especially when hour-to-hour load varied with weather.

4.3 Indoor Environment Quality Evaluation

To inspect influences on clean room environment quality after renovation, two rooms were chosen at random. Basing on the monitoring result on table 3 showed an obvious improvement on room temperature and humidity, with the exception that when 2nd return air system was used, affected by production process or movement of employees; humidity in room 1 fluctuated greatly sometimes.

4.4 Energy Conserving Analysis in Summer Period with a Typical Meteorological Year (TMY)

In order to represent the energy performance of system under actual meteorological condition and the energy conserving effect in summer period, meteorological data of typical meteorological year (TMY) were chosen for periodically simulation. TMY was a defined year chosen from late 10 years, in which climate information each month is close to the monthly average values of the recent 30 years. Smooth disposal of conjoint months was needed, for the reason that monthly average values were derived from different years and were discontinuous. Analysis was based on hour-to-hour meteorological data for a typical meteorological year of Beijing got from US. Lawrence Berkeley National Laboratory (for the reason that Tianjin is near to Beijing in geography and meteorological conditions are similar to Beijing).

Analysis on the operating and adjustment of air conditioning system in a year period shows [4], summer operating condition was achieved when enthalpy of outside air was higher than inside controlled value. However, outside temperature alone was taken for standard for summer operating condition in practice for its convenience. Analysis on daily maximum temperature curve for a typical meteorological year in Beijing showing, air conditioning system in Tianjin operated under summer condition from early May to late October, which accords with the fact.

Figure 6 below shows daily energy cost in a summer period before and after renovation (simulating period from 1st May to 30th September).

It could be seen clearly from Fig.6 that daily energy cost of 2nd return air system reduced greatly than all return air system, no matter for cooling coil or heating coil, and the energy conserving effect became more significant with the rise of outdoor temperature. Statistics of simulating result showing, for a whole summer period, load for cooling coil was 1764476MJ, and for heating coil the value was 727148MJ in all return air system, while in the 2nd return air system under the same meteorological and periodical condition, the load was 1079684MJ and 27835MJ individually. 2nd return air system can save 39% quantity of cold and 61% quantity of heat in a summer period, which was quite significant.

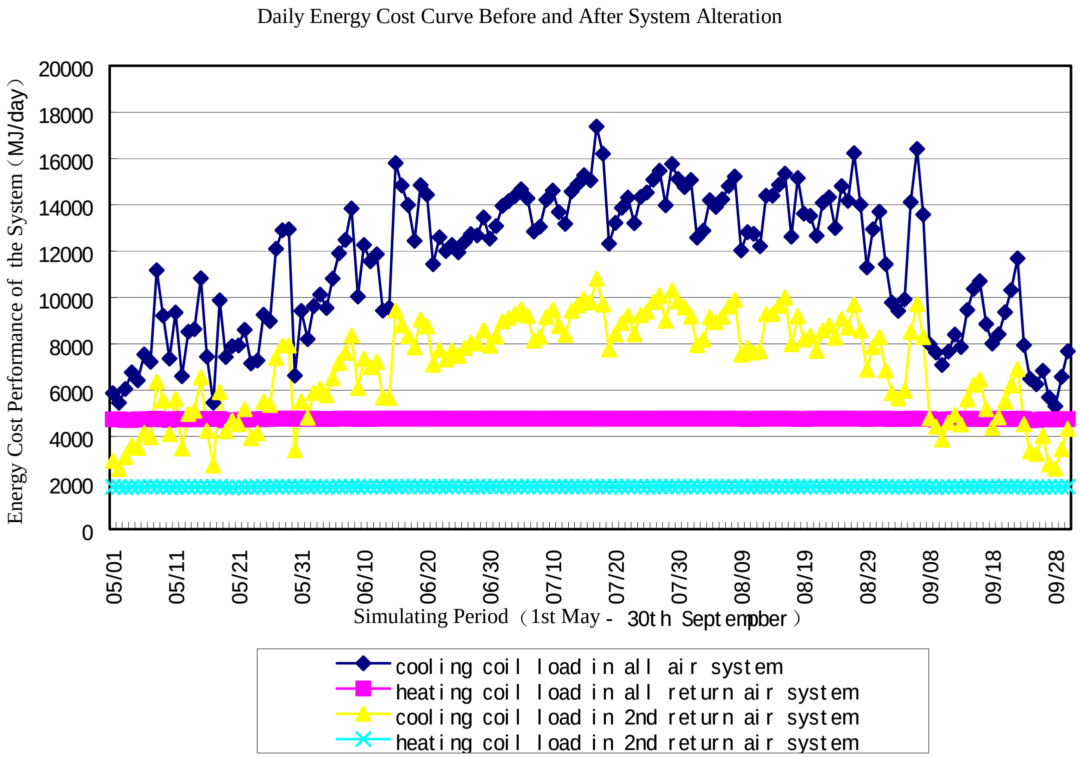

Fig6. Daily energy cost curve of air conditioning system with a typical meteorological year

5.CONCLUSIONS

Results of renovation to HVAC system of this clean room showed that 2nd return air system was superior to all return air system in energy conserving and indoor environment control for this kind of cleaning projects with large area, lower request for temperature and humidity while relatively higher request for cleanliness class. Less reheating energy was needed to adjust supply air temperature for utilization of afterheat in 2nd return air system. This could save energy as well as cost on enlarging capability of water and electrician, and could provide great economical and social benefit.

Energy Plus, as an energy cost analyzing software, did simulating calculation to a certain building with correlative HVAC system. Comparative analysis on simulative and actual data showed its capability of reflecting dynamic load variation authentically, which gave help to design and operation on HVAC system and provides effective assistant means to research on energy cost of building.

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