Evaluation Of Refrigerating Load And Air Flow Performance In Air Conditioning Unit at Hospital Z

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Abstract— *This research was conducted to evaluate air conditioning systems in pharmaceutical warehouses against the ANSI/ASHRAE/ASHE ventilation standards 170 – 2008 and SNI-03-6572-2001. The method used is to calculate the cooling load on the FCU (Fan Coil Unit) using the CLTD (Cooling Load Temperature Difference) method and calculate the External Static Pressure on the ducting system. The analysis is carried out by comparing the results of calculating the FCU requirements with the actual specifications that have been implemented. Furthermore, the measurement results of the existing room air condition are compared with the standards for pharmaceutical warehouse applications. The calculation results show that the pharmaceutical warehouse has a cooling load of 47,472.72 Btu/hr, and has an external static pressure of 123 Pa. While the selected FCU (Fan Coil Unit) has a capacity of 78,771 Btu/hr and a maximum external static pressure of 130 Pa with a rated airflow of 4000 m3/hr. After 9 years of use, it is known that the rated airflow has decreased by around 37% of the FCU capacity. Existing room air conditions obtained the extreme temperature and RH of 23.36°C and 72%, respectively.*

Keywords: Cooling load, cooling load temperature difference, external static pressure, fan coil unit, air conditioning system

I. INTRODUCTION

The use of HVAC systems today has been widely used in various fields for specific purposes and uses. Human activities are already very dependent on HVAC. Without this system, many activities cannot be carried out properly, especially for indoor activities, such as in multi-storey buildings, hotels, malls, factories and hospitals. With consideration of technical and economical aspects, the use of HVAC or what is known as central air conditioning is widely used for air conditioning in multi-storey buildings, hospitals and factories, which have a lot of space and a large cooling load capacity.

HVAC (Heating, Ventilating, Air Conditioning) is a system of air conditioning facilities used in closed rooms to control the condition and temperature of the air in the room as desired. The use of HVAC in hospitals is urgently needed to support health operating rooms, isolation rooms, comfort rooms, and drug storage rooms. In principle, HVAC is almost the same as refrigeration, it's just that the function of HVAC is more complex, it doesn't only function as cooling, but is required to produce an air condition as desired.

Standardization in various fields has its own important role. The main objective of standardization in the field of HVAC is to provide the best facilities, to guarantee comfort, health, and product storage functions so that the quality remains optimal and good. The demands for HVAC systems, are generally adjusted to certain standards set. Given the importance of standardizing the HVAC system, it is necessary to conduct research on air conditioning systems.

II. METHODS

Data collection is carried out by measuring directly using measuring devices such as anemometers, hygrometers, and meters. The secondary data needed is nameplate, tool datasheet, building layout and ducting. Furthermore, the calculation of cooling load, external static pressure, and psychrometric is carried out. Calculation results are analyzed and compared with existing standards, to find out the existing conditions and what steps can be optimized.

3.1 Survey Result Data

The pharmacy warehouse is a building located on the basement floor at latitude 6.99°S and longitude 110.40°E, Semarang, Central Java. Pharmacy warehouse layout can be seen in Figure 1.

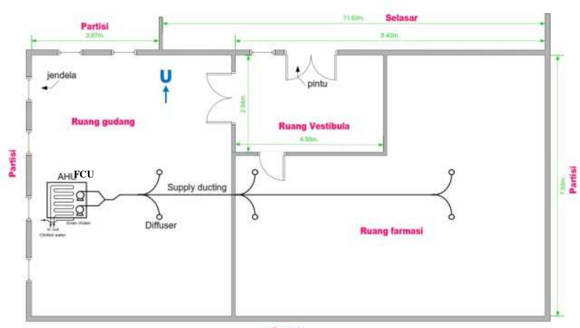

Figure 1. Pharmacy warehouse plan.

The pharmaceutical warehouse uses one Fan Coil Unit (FCU) with specification data as shown in table 1 below.

Table 1. FCU Specification Data

Fan Coil Unit (Midea)			
Model	MKS II 040DA	*Power Supply*	380-415V 3N-50Hz
Rated Air Flow	4000 m^3/h	*Motor Power*	0,55 kW
External Static Pressure (ESP)	130 Pa	*Unit Weight*	162 kg

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ISSN 0216-8685 (print); 2528-6889 (online)

Cooling Capacity (Recycling Air)	23,1 kW	*Heating Capacity (Recycling Air)*	27,7 kW
Cooling Capacity (Fresh Air)	50,3 kW	*Heating Capacity (Fresh Air)*	60,4 kW
Outline Dimension (LxWxH)		1010 x 1300 x 565 mm	

In ducting installations in pharmaceutical warehouses, it is divided into 27 duct item numbers so that ESP calculations are easier and more accurate. An isometric image of the ducting installation can be seen in Figure 2.

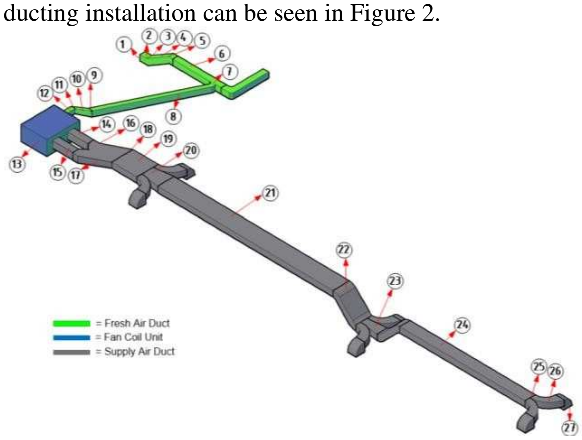

Figure 2. Installation of ducting in a pharmaceutical warehouse

3.2 Measurement Result Data

Room air condition data was obtained from the measurement results of several hygrometers placed at several points in the room. The results of hygrometer measurements can be seen in Tables 2 and 3.

Table 2. Actual Measurement Data of Warehouse Space

9/8/2022	**H1**		**H2**		**H3**	
Hour	**T [°C]**	**RH [%]**	**T [°C]**	**RH [%]**	**T [°C]**	**RH [%]**
09.30	22,8	67	22,5	72	23	69
10.00	22,4	69	22,5	72	23	69
10.30	22,5	70	22,8	72	23	69
11.00	22,5	71	22,8	72	23	69
13.00	22,3	70	22,5	71	23	69
13.30	22,4	71	22,5	71	23	69
14.00	22,4	70	22,8	71	23	69
14.30	22,3	70	22,7	71	23	69
15.00	22,4	70	22,6	71	23	69
Rerata	22,4	69,8	22,6	71,4	23	69

Table 3. Actual Measurement Data of the Pharmacy Room

9/8/2022	**H4**		**H5**		**H6**	
Hour	**T [°C]**	**RH [%]**	**T [°C]**	**RH [%]**	**T [°C]**	**RH [%]**
09.30	21,9	70	23,4	67	22,5	72
10.00	21,8	69	23,4	67	22,5	72
10.30	21,8	69	23,4	67	22,5	72
11.00	22,1	69	23,4	68	22,5	72
13.00	21,8	69	23,4	67	22,5	72
13.30	21,7	69	23,3	67	22,5	72
14.00	21,5	69	23,3	66	22,5	72
14.30	21,4	70	23,3	67	22,5	72
15.00	21,4	70	23,4	66	22,5	72
Rerata	21,7	69,3	23,4	66,9	22,5	72

Table 4. Actual Measurement Data Return grill

No	**Return Grill**		
	T_{edb} [°C]	V_{in} [m/s]	*Size* [cm]
1	23,9	2,1	60x60
2	23,5	2	
3	23,3	1,7	
4	23,8	1,2	
5	24	1,8	
6	24	2	
Rerata	23,7	1,8	

Table 5. Diffuser Actual Measurement Data

No	**Diffuser 1**			**Diffuser 2**		
	T_s [°C]	V_{out} [m/s]	*Size* [cm]	T_s [°C]	V_{out} [m/s]	*Size* [cm]
1	18,7	1,3	30x30	18,7	1,3	30x30
2	18,6	1,2		18,8	1,2	
3	18,7	1		18,6	1	
4	18,5	1,1		18,6	1,2	
5	18,8	1,3		18,5	1,3	
Rerata	18,66	1,18		18,64	1,2	
No	**Diffuser 3**			**Diffuser 4**		
	T_s [°C]	V_{out} [m/s]	*Size* [cm]	T_s [°C]	V_{out} [m/s]	*Size* [cm]
1	18,6	1,4	30x30	18,7	1,5	30x30
2	18,7	1,3		18,5	1,3	
3	18,9	1,1		18,8	1,1	
4	18,4	1,3		18,9	1,2	
5	18,5	1,4		18,5	1,5	
Rerata	18,62	1,3		18,68	1,32	

Continued Table 5

No	**Diffuser 5**			**Diffuser 6**		
	T_s [°C]	V_{out} [m/s]	*Size* [cm]	T_s [°C]	V_{out} [m/s]	*Size* [cm]
1	18,5	1,5	30x30	18,6	1,6	30x30
2	18,8	1,3		18,7	1,3	
3	18,9	1,2		18,5	1,1	
4	18,6	1,3		18,9	1,3	
5	18,5	1,5		18,5	1,5	
Rerata	18,66	1,36		18,64	1,36	

The following is a description of the parameters in Tables 4 and 5, where Tedb is the entering dry bulb temperature or mixing temperature of the return air and outside air, Ts is the supply air temperature, Vin is the return air velocity, Vout is the supply air speed and size. is the area of the cross-sectional area of the return grill/diffuser.

III. RESULTS AND DISCUSSION

3.1 Design Room Air Condition

- Room Temperature (T_r) : 22°C or 71,6°F
- RH (*Relative Humidity*) : 50%
- Moisture content of the air (W_r) : 58 grains/lbm dry air or 0,0082 kg/kg dry air

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ISSN 0216-8685 (print); 2528-6889 (online)

- OA (*Outside Air*) : 2
- ACH (*Air Change Hour*) : 4

3.2 Space Cooling Load Calculation Results

The following table Space Cooling Load Calculation Results is shown table 6.

Table 6. Space Cooling Load Calculation Results

Cooling Load	RSH [Btu/hr]	RLH [Btu/hr]
Eksternal Load		
Partition and Conduction Loads	14.567,42	-
Internal Load		
1. Infiltration Load		
1.1 Doorway Infiltration	144,16	544,62
1.2 Infiltration Open Shut Doors	374,22	1.413,72
2. Occupant Burden	2.835	2.925
3. Lighting Load	792,82	-
4. Equipment Load	2.022,45	-
Total	20.736,07	4.883,34
5. Other loads and *safety factor* (20% x RSH)	4147,21	-
Total	24.883,29	4.883,34
RTH	29.766,63	

In Figure 3 it can be seen that the Partition load and wall conduction value reaches a greater sensible load value of up to 51.01% and the lowest value is obtained in the Doorway Infiltration condition of 0.50% .

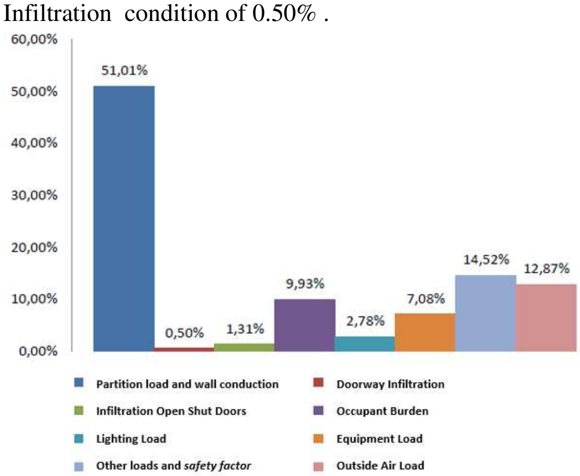

Figure 3 Sensible Load Percentage

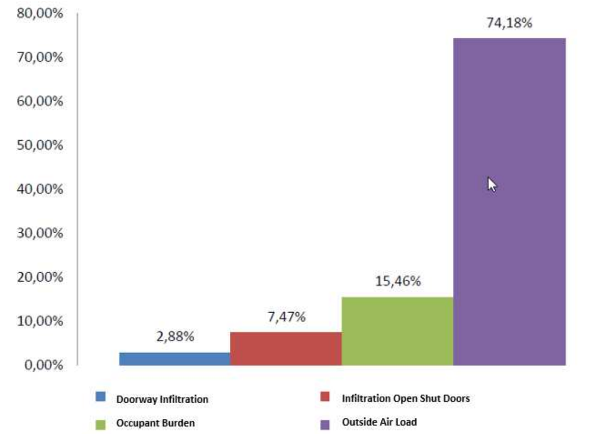

Figure 4 Percentage of Latent Load

Whereas in the latent load shown in Figure 4 the highest value is obtained at Outside Air Load of 74.18% and the lowest value is obtained of 2.88% at Doorway Infiltration. Table 7 is shown Distribution of Actual Air Discharge and table 8 is shown ESP Calculation Results.

Table 7. Distribution of Actual Air Discharge

	Air discharge (ft^3/menit)	Percentage
line *suction*		
Return grill	1373,08	93%
Outside air	99,17	7%
Total	1472,24	100%
line *Discharge*		
Diffuser 1	225,03	15%
Diffuser 2	228,85	16%
Diffuser 3	247,92	17%
Diffuser 4	251,73	17%
Diffuser 5	259,36	18%
Diffuser 6	259,36	18%
Total	1472,24	100%

Table 8. ESP Calculation Results

No	Duct Item	ESP (Pa)
1	*Fresh Air Intake Louver*	4,9768
2	*Straight duct*	0,0675
3	*Elbow mitered 45° duct*	0,6195
4	*Straight duct*	0,1187
5	*Elbow mitered 45° duct*	0,6195
6	*Straight duct*	0,1788
7	*Elbow radius 90° & 45° duct*	0,7802
8	*Straight duct*	0,8364
9	*Elbow 30° mitered duct*	0,5138
10	*Transtition from rec to rec duct*	0,1728
11	*Elbow 30° mitered duct*	0,9214
12	*Straight duct + damper*	0,3758
13	*Fan (ESP capacity)*	0
14	*Straight duct*	8,1286
15	*Straight duct*	8,1286
16	*Elbow mitered 30° duct*	6,1708
17	*Transtition from rec to rec duct*	1,6558
18	*Elbow mitered 30° duct*	10,349
19	*Straight duct*	2,563
20	*Transtition from rec to rec duct*	3,0354
21	*Straight duct*	9,1076

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22	*Rectangular with 4-45° Smooth Radius Ells to Avoid an Obstrution*	11,16
23	*Transtition from rec to rec*	1,8213
24	*Straight duct*	7,5993
25	*Symmetrical Wye, Dove Tail, Rectangular*	1,6948
26	*Elbow with Radius + damper*	5,9714
27	*Supply Square Ceiling Diffuser*	14,93
Total		102,5
Safety Factor 20%		20,499
Total		123

Based on calculations, the Grand Total Heat is 47,472.72 Btu/hr, this value is the total cooling load that must be taken by FCU. The calculated ESP value is obtained by entering the air discharge 2354.31 [ft3/minute] is 123 Pa, and the ESP specifications for the selected FCU are 130 Pa, so the choice of fan capacity is quite appropriate, namely on the basis of selecting the fan ESP value slightly higher than the calculated ESP. Based on table 9, the actual measurement results now show a total air discharge of 1472.24 ft3/minute or 2501.35 m3/day. This value is lower than the rated airflow specification on the FCU nameplate, which is 2354.312 ft3/minute or 4000 m3/hr, which means there is a decrease in CFM airflow of 882 ft3/minute or 37%. FCU installation and operation began in 2012, and until now, it has been in operation for 9 years with 24-hour daily operation. The decrease in FCU performance in supplying air can be due to various factors including the system effect factor, increased ESP in the ducting system and decreased performance of the mechanical and electrical systems of the fan motor due to age of use. Regarding a significant decrease in the rated airflow from the fan, it is better to carry out a further audit to improve the performance and efficiency of the fan, if there is a derating of the rotation of the motor, adjustments to the pulley ratio can be made by increasing the pulley on the motor or reducing the pulley on the impeller fan [13] .

Referring to the ASHRAE 170-2008 standard, the ACH value still meets the standard, namely 8.06 or above 4. While the OA value is 0.52, which means that it is still below the standard value, which is below 2. However, if you refer to the need for fresh air ventilation per person based on the SNI-03-6572-2001 standard for pharmacy rooms, the outside air discharge to the existing pharmacy room is 66, 11 [ft3/minute] or already above the minimum standard requirement of 44.49 [ft3/minute]. If you want to increase the flow of outside air, you can make a damper on the return side of the grill, so that the distribution of return air and outside air can be better regulated. The value of outside air will determine the availability of oxygen in the room. If oxygen levels are limited, then carbon dioxide levels will increase, causing a decrease in air quality.

The measurement results are shown in tables 2 and 3, that the temperature value meets comfort standards, but the RH value is quite high, which is around 70%. There are many problems if the room is too humid, cold and does not get sunlight, including fungi and viruses that will easily multiply [5]. Setting at a relatively high supply cold water temperature can reduce the electric power consumption of the compressor [14], but has an impact on the room's relatively high RH value. A fairly high RH value can be anticipated by lowering the maximum supply cold water temperature setting at 8°C so that the cooling coil capacity value becomes around 36,107.76 Btu/hr and installing an air heater with a capacity of around 2.12 kW.

While the existing conditions in the pharmaceutical warehouse have no indication of mold growth indoors. This can happen due to several factors, including no condensation in the ducting, walls, and roof. Air and room cleanliness is well maintained, sufficient OA (Outside Air) and ACH (Air Change Hour) values, with an ACH value of 8.06; while the OA value is 0.52. In addition, it has a fairly good distribution of air circulation with 6 diffusers, so it can prevent higher humidity from being trapped in one particular location, thus being safe from the growth of fungi and other microorganisms.

IV. CONCLUSION

Based on the results of calculations and analysis that the total cooling load in the pharmaceutical warehouse is 47,472.72 Btu/hr and the ducting installation system has an external static pressure of 123 Pa. The measurement and analysis results show that the air conditioning system in the pharmaceutical warehouse needs to be further optimized, such as the RH of the room which is quite high, which is around 70%, so as to support health and comfort for workers in the pharmaceutical warehouse, it is necessary to optimize it through a cold water setting of around 8° C and requires heating with a capacity of about 2.12 kW. In addition, a decrease in fan performance caused by the age factor can be adjusted to the pulley ratio between the impeller and the fan motor.

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ISSN 0216-8685 (print); 2528-6889 (online)