Design of Air Conditioning System (City-multi) for Seminar Rooms of Technological University (Kyaukse)

Ms.Htwe Tin

***Abstract:*-Chiller and city-multi systems are being used for large buildings. Although chiller system occupies large space, city - multi system is a compact air conditioning system. It does not need large space to locate outdoor and indoor units, liquid and gas pipe lines as other systems. City-multi system with R410A was chosen to arrange air conditioning system for Technological University (Kyaukse). R410A has several advantages than R22 and its operating pressure is 1.6 times higher than R22'S at the same temperature. System compact size design can be obtained by using R410A. Cooling loads for seminar rooms of this university were calculated to compare the system performance and power consumption for R410A and R22.**

***Key-Words:*-air conditioning , city-multi, R 410A, cooling load, performance, power consumption**

I. Introduction

Refrigerant R410A is one of the substitutes currently accepted as a replacement for the commonly used $HCFC_S$ (hydrochlorofluorocarbons). Refrigerants such as R-12 and R-22 are family of chemicals that contain chlorine, fluorine, and carbon. The chlorine content in these compound causes the depletion of the ozone layer. R 410 A is a type of HFC_S and it does not contain chlorine. It has an Ozone Depletion Potential rating of 0.00 verus 0.05 for R 22. Its boiling point (-51.4°C) is lower than R 22 (-40.8°C) and the quantity of heat received by one kilogram of refrigerant from the space being cooled is larger than R 22. And again its operating pressure is 1.6 times higher than R22. So its specific volume is less than R22, and smaller pipe diameters and small system size per KW are obtained by using R410 A. Although installation and service procedures are similar for R 410 A as the methods for R22, there are several critical differences. Systems using R410 A must be designed for components are not interchangeable and cannot be matched with R22 components. So, equipment and piping must be designed for that increased pressure. Components used on R410 A air conditioners use thicker metals to withstand the higher operating pressures [1].

Manuscript received January 4, 2009. This work was supported in part by the Ministry of Science and Technology, Union of Myanmar. Ms. Htwe Tin is with the Technological University (Kyaukse). Contact Phone: 095-066-50418 Mechanical Engineering Department, e-mail: dhtwetin@gmail.com

II. DESIGN CONSIDERATION OF COOLING LOADS

Technological University (Kyakse) is located in Mandalay division of Myanmar Country. Design condition is first selected before considering the cooling load. Dry bulb temperature 95°F (DBT), relative humidity 65% (RH), and daily range 22.6°F were chosen for outside design condition (ODT) of Technological University (Kyaukse).Dry bulb temperature 78°F and 50% RH were chosen for inside design conditions [2].

A. Heat gain through exterior structure

Solar radiation forms the greatest single factor of cooling load in buildings. Radiant energy from the sun is absorbed by the room materials, both the structure and furnishings. Cooling load temperature difference accounts for the heat storage effect [3].

B. Conduction heat gain through exterior structure

The conduction heat gain through the exterior roof, walls and glass are each determined by the following equations

For walls and glass

$$Q = U \times A \times CLTD_C \tag{1}$$

Q = net room conduction heat gain through roof, wall or glass, BTU/hr

U = overall heat transfer coefficient for roof, wall or glass, BTU/hr ft^2 °F

A = area of wall, roof or glass, ft^2

$CLTD_C$ = corrected value of cooling load temperature difference, °F that accounts for heat storage effect.

For wall

$$CLTD_C = [(CLTD + LM)\,K + (78 - t_R) + (t_0 - 85)]\,f \tag{2}$$

For glass

$$CLTD_C = (CLTD) + (78 - t_R) + (t_0 - 85) \tag{3}$$

C. Heat gain by solar radiation through glass

The net heat gain by solar radiation through glass can be calculated by the following equation.

$$Q = SHGF \times A \times SC \times CLF \tag{4}$$

SHGF = maximum solar heat gain factor, BTU/hr ft^2

A = area of glass, ft^2

SC = shading coefficient

CLF = cooling load factor for glass

D. Transmission gain through interior structure

The heat flows from interior unconditioned spaces to the conditioned spaces can be calculated by the following equation.

$Q = U \times A \times TD$ (5)

TD = temperature different between conditioned and unconditioned space, °F

E. Infiltration and outside air

The equations for determining the sensible and latent loads for ventilation and outside air are as follow.

$Q_S = 1.08 \times CFM \times TC$ (6)

$Q_L = 0.68 \times CFM \times (\omega_{Hi} - \omega_{Ho})$ (7)

Q_S, Q_L = sensible and latent cooling load from ventilation and infiltration air, BTU/hr

CFM = air ventilation and infiltration rate, ft^3/min

TC = temperature change between outside and inside air, ºF

$\omega_{Hi,o}$ = specific humidity for outside and inside conditions, gr.wv/lb-da

If $(CFM)_{venti}$ is greater than $(CFM)_{infil}$, $(CFM)_{infil}$ can be neglected.

For ventilation,

$Q_S = 1.08\ [((CFM)_{venti} \times (B.F)]\ \Delta T$ (8)

$Q_L = 0.68\ [(CFM)_{venti} \times (B.F)] \times \Delta\ \omega_H$ (9)

ΔT = temperature difference between outside and inside condition, ºF

B.F = By pass factor of cooling coil

If the ventilation air is less than the infiltration air (cfm), sensible and latent heat can be calculated as follows.

$Q_S = Q_{S\,(Infil)} + Q_{S\,(venti)}$
$= [\,1.08 \times (CFM)_{Net\ infil} \times \Delta T] + [1.08 \times (CFM)_{venti} \times BF \times \Delta T]$ (10)

$Q_L = Q_{L\,(Infil)} + Q_{L\,(venti)}$
$= [\,0.68 \times (CFM)_{Net\ Infil} \times \Delta\omega_H] + [0.68 \times (CFM)_{venti} \times BF \times \Delta\omega_H$ (11)

$(CFM)_{Net\ infil} = (CFM)_{infil} - (CFM)_{venti}$

F. Determination of infiltration air

For window,

$$(CFM)_{window} = \frac{\text{Room volume}(ft^3)}{60} \times G \quad (12)$$

For doors

$(CFM)_{door}$ = 2.5 cfm/person x number of person (13)

$(CFM)_{Infil} = (CFM)_{window} + (CFM)_{door}$ (14)

G. Internal gain

Heat gain from lighting in the room can be calculated by the following equation[3].

$Q = 3.4 \times W \times BF \times CLF$ (15)

Q = net heat gain from lighting, BTU/hr

W = lighting capacity, watt

BF = ballast factors

CLF = cooling load factor for lighting

The equations for sensible and latent heat gain from people are,

$Q_S = q_S \times n \times CLF$ (16)

$Q_L = q_L \times n$ (17)

q_S, q_L = sensible and latent heat gain/ person, BTU/hr

n = number of people

CLF = cooling load factor for people

H. Miscellaneous gain

The sensible and latent heat are usually taken as 15% and 10% of room sensible and latent heat for miscellaneous gain [4].

$Q_S = 0.15 \times (RSH)_{sub\text{-}total}$ (18)

$Q_L = 0.1 \times (RLH)_{sub\text{-}total}$ (19)

I. Outside air through apparatus

The sensible and latent loads for ventilation air can be calculated by the following equations[4].

$Q_S = 1.08\ (CFM)_{vent}\ (1\text{-}BF)\Delta T$ (21)

$Q_L = 0.68(CFM)_{vent}\ (1\text{-}BF)\Delta\omega_H$ (22)

ΔT = temperature difference between outside and inside condition, ºF

III. RIGERATION SYSTEM

The following equation can be used to determine the dehumidified air quantity [5].

$$CFM_{da} = \frac{ERSH}{1.08\ (1\text{-}B.F)\ (T_{RM} - T_{adp})} \quad (23)$$

ERSH = Effective room sensible heat (Btu/hr)

T_{RM} = Room dry-bulb temperature (°F)

T_{adp} = Apparatus dew-point temperature (°F)

$$(CFM)_{sa} = \frac{RSH}{1.08\ (\Delta T)_{desir:}} \quad (24)$$

CFM = total supply air cfm

RSH = room sensible heat (Btu/hr)

ΔT = desired temperature difference between ODT and IDT condition

The evaporator entering and leaving temperatures can be determined by the following equation [5].

$$T_{edb} = T_{RM} + \frac{CFM_{oa} \times (T_{oa} - T_{RM})}{CFM_{sa}} \quad (25)$$

$$Tldb = T_{adp} + B.F \times (T_{edb} - T_{adp}) \quad (26)$$

The following figure shows the Pressure-enthalpy diagram for refrigerant R410A [1].

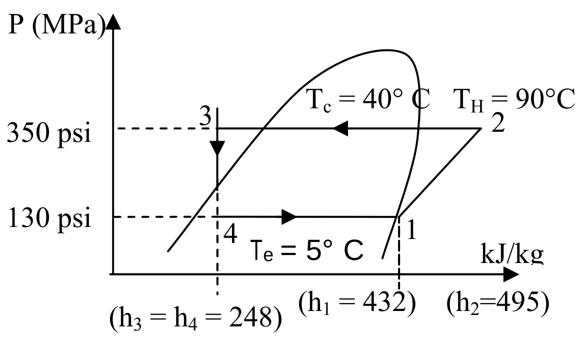

Fig. Pressure-enthalpy diagram for refrigerant R410A [1].

IV. DESIGN CALCULATION AND RESUTS

A. Design conditions

Building material, lighting, number of persons, outside and inside design conditions are considered as design conditions [6].

B. Area calculations:

Table 1. Area calculations for seminar rooms 3:12 and 3:13

Room	Item	Orientation	Area (ft^2)
3:12	glass	SW NE	$2W_1+2D_1$=117.16 $2W_1+2D_1$=117.16
	wall	SE	widthxheight=28x12 = 336
		NE	(40x12)-($2W_1$-$2D_1$)=280
		NW	28x12 = 336
		SW	(40x12)-($2W_1$-$2D_1$)=280
3:13	glass	SW NE	W_1+2D_1=81.66 W_1+2D_1=81.66
	walls	SE	widthxheight=28x12 = 336
		NE	35x12 – W_1 – $2D_1$ = 272
		~~NW~~	~~28x12 = 336~~
		SW	35x12 – W_1 – $2D_1$ = 272

Table 2. Area calculations for seminar rooms (108Aand108B)

Room	Item	Orientation	Area(ft^2)
108-A	glass	E	$2W_1+D_2$=85.77
		W	W_2 = 6.61
	walls	S	22 x 12 = 264
		E	(25x12)-$2W_1$-D_2 = 158
		NW	28 x 12 = 336
		W	(6x12)-W_2=59
108-B	glass	N	$2W_1+D_1$=85.77
		S	W_2=6.61
	walls	SE	widthxheight = 28 x 12 = 336
		N	(25x12)-$2W_1$-D_2=158
		W	(22x12)=264
		S	(6x12)-W_2=59

C. Estimated persons

Table 3. Number of estimated person and lightings for seminar rooms

Room	Number of persons	Number of fluorescent
3:12	60	8 (40 watts)
3:13	50	6 (40 watts)
3:14	60	8 (40 watts)
108-A	10	4 (60 watts)
108-B	10	4 (60 watts)

D. Determination of peak time

Peak time could be selected to obtain maximum heat gain and peak time for these rooms is considered both roof and glass[3].

Table 4. Determination of peak time for seminar room 3:12

Hour	Item	Types of heat gain	Heat gain Q (Btu/hr)
14:00	Roof	Conduction through roof	Q=UxAxCLTD$_C$ = 12947.20
	Glass	Conduction through glass	Q=UxAxCLTD$_C$ = 1340.08
	Glass	Solar radiation through glass	Q=SHGFxAxSCxCLF = 7827.88
	Total heat gain		22115.162
15:00	Roof	Conduction through roof	13708.80
	Glass	Conduction through glass	1400.99
	Glass	Solar radiation through glass	8662.85
	Total heat gain		23772.64
16:00	Roof	Conduction through roof	13708.80
	Glass	Conduction through glass	1461.907
	Glass	Solar radiation through glass	8454.11
	Total heat gain		23624.818

Table 5. Determination of peak time for seminar room 3:13

Hour	Item	Types of heat gain	Heat gain Q (Btu/hr)
14:00	Roof	Conduction through roof	11328.80
	Glass	Conduction through glass	933.96
	Glass	Solar radiation through glass	5455.59
	Total head gain		17718.35
15:00	Roof	Conduction through roof	11995.20
	Glass	Conduction through glass	976.41
	Glass	Solar radiation through glass	6037.52
	Total head gain		19009.13
16:00	Roof	Conduction through roof	11995.20
	Glass	Conduction through glass	1018.86
	Glass	Solar radiation through glass	5892.04
	Total heat gain		18906.10

Peak time 15 hours (3pm) for all SW walls and the month ,August were chosen as design month and time to calculate cooling loads for these rooms.

E. Determination of maximum heat gain for NE walls

Maximum heat gains were calculated at design time 15 hours (3pm) for each room.

Table 6. Maximum heat gains for NE walls at 15 hours

Room	Item	Types of heat gain	Heat gain Q (Btu/hr)
3:12	Wall	Conduction through wall	925.68
	Glass	Conduction through glass	1400.99
		Solar radiation through glass	2814.17
	Total heat gain		5140.84
3:13	Wall	Conduction through wall	899.232
	Glass	Conduction through glass	976.414
		Solar radiation through glass	1961.3
	Total heat gain		3836.96

F. Determination of maximum heat gain for E,W,N and S walls.

Maximum heat gains were estimated at design time 15 hours (3pm) for E,W,N and S walls of corner rooms [3].

G. Maximum heat gain for W and E walls of room (108-A)

Table 7. Maximum heat gain for W and E walls of room 108-A at 15 hours, Q (Btu/hr)

Item	Types of heat gain	Q
West(W)	Conduction through wall	112.92
	Conduction through glass	79.05
	Solar radiation through glass	575.86
Total head gain		767.84
East(E)	Conduction through wall	714.79
	Conduction through glass	1025.80
	Solar radiation through glass	2075.63
Total head gain		3816.23

H. Maximum heat gain S and N walls of room (108-B)

Table 8. Maximum heat gain for N and S walls of room (108-B) at 15 hours, Q (Btu/hr)

Item	Types of heat gain	Q
South(S)	Conduction through wall	123.19
	Conduction through glass	79.05
	Solar radiation through glass	130.87
Total head gain		333.12
North(N)	Conduction through wall	164.95
	Conduction through glass	1125.80
	Solar radiation through glass	1362.37
Total head gain		2553.13

I. Calculation of cooling loads for seminar room(3-12)

(1) Heat gain through exterior structure

For seminar rooms 3:12, SW and NE walls are considered as exterior structure.

For room 3:12, For SW – wall

Total heat gain by radiation and conduction through glass and wall = 10,648.48 Btu/hr

For NE – wall

Total heat gain by radiation and conduction through glass and wall = 5,140.84 Btu/hr

For roof

Heat gain by conduction through roof=13,708.80 Btu/hr

Total heat gain, $Q_{s_I} = 29498.121\text{Btu/hr}$

(2) Transmission gain through interior structure

For class room 3:12, SE and NW walls, floor are considered as interior structure.

For SE wall, $q_{s_{II\,(wall)SE}} = U\,A\,(TD)$

where, TD = 0

Because adjacent room 3:13 is conditioned.

$q_{s_{II\,(wall)SE}} = 0$

For NW wall, $q_{s_{II\,(Wall)NW}} = U\,A\,(TD) = 0$

where, TD = 0

Because adjacent room 108-B is conditioned room

$q_{s_{II\,(Wall)NW}} = 0$

For floor, For ground floor, $q_{s_{II\,(ground)}} = U\,A\,(TD)$

where, TD = 0

Because lower room, room 2:12 is conditioned.

$Q_{s_{II}} = 0$

(3) Infiltration and Outside air

$$(\text{cfm})_{\text{window}} = \frac{(40 \text{ x } 28 \text{ x } 12)}{60} \text{ x } 1.5 = 336.00$$

$$(\text{cfm})_{\text{door}} = 2.5 \text{ cfm / person x } 60 = 150.00$$

$$(\text{cfm})_{\text{Infil}} = (\text{cfm})_{\text{windows}} + (\text{cfm})_{\text{door}}$$

$$= 336.00 + 150.00 = 486.00$$

$$(\text{cfm})_{\text{venti}} = 15 \text{ cfm / person x } 60 = 900.00$$

$$Q_{S_{III}} = 1.08 \text{ x}900 \text{ x}0.2 \text{ x}17 = 3{,}304.8 \text{ Btu/hr}$$

$$Q_{L_{III}} = 0.68\text{x } 900 \text{ x } 0.2 \text{ x } 92 = 11{,}260.811 \text{ Btu/hr}$$

(4) Internal gain

For Lighting,

$$q_{S_{IV\,(lig)}} = 3.4 \text{ x } (8 \text{ x } 40) \text{ x } 1.25 \text{ x } 1 = 1{,}360.00 \text{ Btu/hr}$$

For people,

$$q_{S_{IV\,(peop)}} = 230 \text{ x } 60 = 13{,}800.00 \text{ Btu/hr}$$

$$q_{L_{IV\,(peop)}} = 190 \text{ x } 60 = 11{,}400.00 \text{ Btu/hr}$$

$$Q_{S_{IV}} = q_{S_{IV\,(lig)}} + q_{S_{IV\,(peop)}} = 15{,}160.00 \text{ Btu/ hr}$$

$$Q_{L_{IV}} = 11{,}520.00 \text{ Btu / hr}$$

$(\mathrm{RSH})_{\text{sub-tol}} = Q_{S_I} + Q_{S_{II}} + Q_{S_{III}} + Q_{S_{IV}}$
=29498.121+0+3304.8+ 15160.00
= 47,962.92 Btu/hr

$(\mathrm{RLH})_{\text{sub-tol}} = Q_{L_{III}} + Q_{L_{IV}}$
= 11,260.8 + 11,400.00 = 22,660.8 Btu/hr

(5) Miscellaneous gain

q_{S_V} = 15% of $(\mathrm{RSH})_{\text{sub-tol}}$ = 7,194.438 Btu/hr

q_{L_V} = 10% of $(\mathrm{RSH})_{\text{sub-tol}}$ = 2,266.08 Btu/hr

RSH = $(\mathrm{RSH})_{\text{sub}} + q_{S_V}$ =55,157.358 Btu/hr

RLH = $(\mathrm{RSH})_{\text{sub}} + q_{L_V}$ = 24,926.88 Btu/hr

RTH = RSH + RLH = 80,084.238 Btu/hr

SHF = $\dfrac{\mathrm{RSH}}{\mathrm{RTH}}$ = 0.69

(6)Outside air through apparatus:

$q_{S_{VI}}$ = 1.08 x 900 x 0.8 x 17= 13,219.20 Btu/hr

$q_{L_{VI}}$ = 0.68 x 900 x0.8 x 92 = 45,043.20 Btu/hr

GTH =(RTH)+ $q_{S_{VI}} + q_{L_{VI}}$ = 138,346.64 Btu/hr

Ton of refrigeration = 11.52 tons.

J. Determination of evaporator temperatures and supply air (cfm) for seminar room 3:12

Evaporator parameters and supply air (cfm) were also estimated for these rooms.

$(\mathrm{cfm})_{\text{dehu:}} = \dfrac{55{,}157.86}{1.08 \times 0.8(78-50)}$ = 2,280.00

Outlet temperature difference = $\dfrac{55{,}157.86}{1.08 \times 2280.00}$
= 22.4° F

Supply, $(\mathrm{cfm})_{sa}$ = $\dfrac{55{,}157.86}{1.08\,(17)}$ = 3,004.24

By pass, cfm= $(\mathrm{cfm})_{sa} - (\mathrm{cfm})_{dehi}$ = 727.24

T_{edb}= 78 + $\dfrac{900}{3{,}004.24}$(95 – 78)= 83.09° F

T_{ldp} = 50 + 0.2 (83 – 50) = 56.6° F

For room 108-A

$(\mathrm{cfm})_{\text{dehu:}} = \dfrac{15{,}163.02}{1.08 \times 0.8(78-50)}$ = 626.778

Outlet temperature difference = $\dfrac{15{,}163.02}{1.08 \times 626.78}$
= 22.4° F

Supply, $(\mathrm{cfm})_{sa}$ = $\dfrac{15{,}163.02}{1.08\,(17)}$ = 825.87

By pass, cfm= $(\mathrm{cfm})_{sa} - (\mathrm{cfm})_{dehi}$ = 199.09

T_{edb}= 78 + $\dfrac{150}{825.87}$(95 – 78) = 81.08°F

T_{ldp}= 50 + 0.2 (81 – 50) = 56.2° F

Table 9. Requirements of cooling load and supply air for each room

Room	Room cooling Load (Btu/hr)	Ton of Refrig-eration	Supply air $(\mathrm{CFM})_{\text{calcu:}}$
108-B	27,859.92	2.32	719.52
3:12	138,346.64	11.52	3,004.21
3:13	113,815.64	9.48	2,423.26
3:14	138,346.64	11.52	3,004.21
108-A	29,812.41	2.48	825.87

Table 10. Evaporator entering and leaving temperatures and (SHF) for each room

Room	T_{edb} (°F)	T_{ldb} (°F)	SHF
108-B	81.50	56.30	0.73
3:12	83.00	56.60	0.69
3:13	83.00	56.60	0.68
3:14	83.00	56.60	0.69
108-A	81.00	56.20	0.75

K. Calculation of system capacities for R 22 and R410A at compressor outlet temperature 90 ℃

System capacities are calculated at compressor outlet temperature for R22 and R410A.

37.32 × 200 / 60 × 1.055 = $\omega^{\circ} \times (h_1 - h_4)$

ω° = 0.763 Kg / sec

Power required for outdoor unit= $\omega^{\circ} \times (h_2 - h_1)$

=38.15 KW or 51.139 HP

$(\mathrm{cop})_{refri}$ = (37.32 × 1200) / (2545 × 51.139)= 3.441

For,R410A,

37.32 × 200 / 60 × 1.055 = $\omega^{\circ} \times (h_1 - h_4)$

ω° = 0.7132 Kg / sec

Power required for outdoor unit = $\omega^{\circ} \times (h_2 - h_1)$

= 44.93 KW or 60.23 HP

$(\mathrm{cop})_{refri}$ = (37.32 × 1200) / (2545 × 60.23) = 2.917

Table 11. Comparison of power requirement, $(\mathrm{cop})_{refri}$ and mass flow rate of refrigerant for R22 and R410A

Item	R 22	R 410 A
Power (Hp) for outdoor unit	51.139	60.23
$(\mathrm{cop})_{refri}$	3.441	2.917
Mass flow rate of refrigerant	0.763	0.7132

V. CONCLUSION

Outside design temperature 95° F (DBT) and 65% RH were selected from the meteorological records and inside design temperature 78F (DBT) and 50% RH were chosen from ASHRAE comfort zone [6].

This university was constructed fifteen degrees deviated from the east. Peak time was chosen to obtain maximum heat gain for each room and was selected at 3 pm (15:00 hours) for all SW-walls of these rooms. For southwest-facing glass, maximum solar heat gains occur in the fall in the afternoon. After selecting the peak time, cooling loads were calculated for these rooms. Because selection of indoor unit, cooling capacity, piping and duct size depend on cooling loads.. Sensible heat factors (SHF) were calculated for each room. Weight of refrigerant circulated in the refrigeration system, power required to drive the compressor, and (cop) values for refrigeration system were also calculated for R 22 and R 410A. These values are shown in Table 4.Cooling loads and supply air (cfm) required for each room are within the allowable values of indoor units. Therefore, selection of indoor units model is satisfied with both cooling capacities and supply air (cfm) required for each room. These values are shown in Table 11 and Table 12 for each room. Power required by using R 22 is less than R 410A and (cop) values obtained by using R 22 is higher than R 410A. However, R22 which contains chlorine depletes ozone layer. The choice of refrigerant is greatly influence energy efficiency. Too much green house effect will lead to global warming and greenhouse gases include water vapour, carbon dioxide, methane and nitrous oxide, as well as some refrigerants. When these gases build up in the atmosphere, they trap heat. The natural greenhouse effect is necessary for life on earth and scientists believe that too much greenhouse effect will lead to global warming. Areas 21.2ft^2 and 10.6ft^2 are needed for maximum and minimum outdoor units of 50 HP and 8HP. Therefore, system compact size and effective utilization can be obtained by using city-multi system with refrigerant R410A.

ACKNOWLEDGEMENT

The author wishes to express her sincere thanks and gratitude to Dr.Mi Sandar Mon, Professor and Head of Mechanical Engineering Department, Y.T.U, for the close motivation and encouragement during the entire course of the research period.Next, the author is deeply grateful to Dr. San San Yee, Professor, West Yangon Technological University, for her guidance, willingness to share ideas and helpful suggestion throughout the research.

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