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Sustainable HVAC in Industries Using TEC and Dry Ice

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Abstract: ***Global warming is caused by the emission of greenhouse gases. Gases that trap heat in the atmosphere are called greenhouse gases. Carbon dioxide is a greenhouse gas, absorbing and emitting infrared radiation at its two infrared-active vibrational frequencies. The world-wide average CO2 emission by capita was about 4 tons per year in 2005. For North America it was about 20 tons and for Europe about 10 tons per year per capita. By 2050, the world-wide average CO2 emission per capita needs to be reduced to 2 tons per year. In the following years, the emissions will need again to be cut by half. On November 12, 2015, NASA scientists reported that human-made carbon dioxide (CO_2) continues to increase above levels not seen in hundreds of thousands of years: currently, about half of the carbon dioxide released from the burning of fossil fuels remains in the atmosphere and is not absorbed by vegetation and the oceans. In recent years, with the increase awareness towards environmental degradation due to the production, use and disposal of Chloro Fluoro Carbons (CFCs) and Hydro Chlorofluorocarbons (HCFCs) as heat carrier fluids in conventional refrigeration and air conditioning systems has become a subject of great concern and resulted in extensive research into development of novel refrigeration and space conditioning technologies. Thermoelectric cooling provides a promising alternative R&AC technology due to their distinct advantages. Carbon dioxide captured from industrial flue gases is used to manufacture dry ice which in turn is used in HVAC for air-conditioning purpose. In the present work, cooling capacity is achieved using dry ice and with one peltier element. The new HVAC system is fabricated and tested for its performance. Theoretical calculations are performed on the test system. Also, carbon life cycle analysis will be performed to establish the sustainability of the proposed system.***

Keywords: Cop – sensible heat factor – Cooling Capacity – Ton of Refrigeration tr – Dry bulb Temperature – Dry ice- Thermo electric cooling modulus - Latent heat - Sensible Heat - Relative humidity - Specific humidity - Mass flow rate

1. Introduction

Global warming is caused by the emission of greenhouse gases. Gases that trap heat in the atmosphere are called greenhouse gases. Carbon dioxide is a greenhouse gas. Carbon dioxide capture, transport, and storage (ccs) from fossil fuel fired power plants is drawing increased interest as an intermediate solution towards sustainable energy system in the long term. Global warming and climate change are terms for the observed century-scale rise in the average temperature of the Earth's climate system and its related effects. Multiple lines of scientific evidence show that the climate system is warming. Although the increase of near-surface atmospheric temperature is the measure of global warming often reported in the popular press, most of the additional energy stored in the climate system since 1970 has gone into the oceans. The rest has melted ice and warmed the continents and atmosphere. Scientific understanding of global warming is increasing. The Intergovernmental Panel on Climate Change (IPCC) reported in 2014 that scientists were more than 95% certain that global warming is mostly being caused by human (anthropogenic) activities, mainly increasing concentrations of greenhouse gases such as carbon dioxide (CO_2). Human-made carbon dioxide continues to increase above levels not seen in hundreds of thousands of years. Currently, about half of the carbon dioxide released from the burning of fossil fuels remains in the atmosphere. The rest is absorbed by vegetation and the oceans.

A number of projects have been carried out to determine the viability of capturing CO_2 before it gets to the atmosphere, compressing it as much as possible and storing it deep in the ground forever. But several companies are now attempting to put the wasted gas to good use instead of simply burying it underground. Broadly, three different configurations of technologies for capture exist: post-combustion, pre-combustion, and oxy fuel combustion. Post combustion technologies available for CO_2 capture such as absorption, adsorption, membrane, cryogenic separation, etc. The main objective of present work is a model cooling unit was fabricated and performance has been compared with air conditioned test rig.

A Coleman company polystyrene foam insulated cooler is taken as a model cooling unit. We are replacing the whole existing HVAC system (Compressor, Electromagnetic clutch, Condenser, Receiver/drier Metering devices, Evaporator, Refrigerants, Hoses and blower) by the TEC System Chamber, Dry ice chamber, one Fan and some electric equipment. Carbon dioxide is captured from industries flue gases which in turn is used to manufacture dry ice is used in model cooling unit for air-conditioning purpose. In the present work cooling capacity is achieved using dry ice and with one Peltier element. Theoretical calculations are performed on the test system.

2. Experimental Setup

A Coleman company polystyrene foam insulated cooler is taken as a model cooling unit and is then equipped with a Thermo electric cooler on top of the unit. A 12x12 cm Fanis placed in front of it for the suction of atmospheric air in to the cooling unit. The cooling unit is divided as two chambers and separated with an Aluminum sheet. Aluminum sheet is used as a heat transfer medium from dry ice to dehumidified cool air by indirect contact and supplied to conditioned space through an elbow pipe.

We are replacing the whole existing HVAC system (Compressor, Electromagnetic clutch, Condenser,

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Receiver/drier, Metering devices, Evaporator, Refrigerants, Hoses and blower) by the TEC System Chamber, Dry ice chamber, one Fan and some electric equipment. The electric source provided to the TEC system by the battery as per the specification of TEC plate. The two heat sinks are attached to the surface of TEC plate. A Fan circulates the atmospheric air from outside to the TEC system chamber and again the cool air is circulating from TEC system chamber to Dry ice chamber. Further cooled air now discharge to the conditioned room through an elbow pipe. A Vent hose pipe is used at the bottom of the cooling unit to evacuate the CO_2 gas from Dry ice chamber to atmosphere.

Specification of conditioned space and description of loads:

Corrugated Card board Space dimensions are

Height of the space = 1.04m
Length of the space = 0.74m
Width of the space = 0.5m

Thickness of the space = 0.002m
Diameter of Outlet to the conditioned space = 0.106m

Loads taken in the conditioned space are,

- Lighting Load
- People Load
- Appliance Load

Outside conditions are 33^0c DBT and 75% RH

MODEL COOLING UNIT

A/C TEST RIG

3. Load Calculations

3.1 Cooling Load Estimate

The components of the cooling load for air-conditioning can now be summerized as follows. The load is classified as the room load that falls on the room directly, and the total load that falls on the apparatus. But, in this we mainly concentrate on room load.

Room load:
A. Room Sensible Heat (RSH)

i. Solar and transmission heat gain through walls, roofs etc.
ii. Solar and transmission heat gain through glass.
iii. Transmission gain through partition walls, ceiling, floor, etc.
iv. Infiltration
v. Internal heat gain from people, power, light, quipment, appliances.
vi. Additional heat gain not accounted above, safety factor, etc.

The sum of all above gives the Room Sensible Heat (RSH) load.

B. Room Latent Heat (RLH)

i. Infiltration
ii. Internal heat gain from people, steam, app;iances.
iii. Vapour Transmission
iv. Additional heat gain not accounted above, safety factor, etc.

The sum of these gives Room Latent Heat (RLH) load.

3.2 Load Calculations for Model Cooing Unit

Calculation of Room sensible Heat

(I). Internal Heat Gain:
1.
2. Occupancy Load
Heat Liberated at 23^0 C and 55%RH is when seated at Rest
Sensible heat = 75W
Latent heat=40W

(II). Lighting Load for the Florescent Bulb Q is
Q= Total Watt x 1.25
= 40 x 1.25
= 50W

(III) Appliance Load for equipment load
In case of iron equipment to be cooled
Sensible heat=2400 kJ / hr.
= 0.666KW

(IV) Solar and Transmission Heat Gain through Walls Roof
`U` for corrugated card Board of thickness 0.002m= K/L= 0.064/0.002=32 W/m-K
Area= (Length of Wall x Height of Wall) =1.04 x 0.74= 0.769m^2
Q=32 x 0.7696 x (32-23)=246.272 W

(V) Transmission Gain through ceiling
U=32 W/m-K
Area of Ceiling = (Length x Width)
A= 0.74 x 0.5 = 0.37m^2
Q= 32 x 0.37 x (33-23) =118.4 W
RSH= occupancy Load +Light load+ appliance load+Solar & Transmission Heatgain through space.
RSH= 75 x 1+50+0.6666 +246.272 +118.4= 490.338 W
Additional heat gain not accounted above is safety factor of 5% RSH

Total room sensible Heat= 490.338 +(0.05 x 490.338)
= 490.338 +24.516
= 514.854W= 0.5148kW

Load Due to infiltration
Outside air sensible heat (OASH)
OASH= 0.0204 x cmm x (T_0-T_1)
= 0.0204 x (15 x π/4 x $(0.106)^2$ x (33-25)
= 0.0204 x 0.132 x 10 =0.0269 kW

Total Sensible Heat (TSH)
TSH = RSH+OASH
= 0.5148+0.0269= 0.5417 kW

Calculation of Room latent heat:
1. Internal heat gain
Occupancy load
Heat liberated at 23^0 C and 55%RH is when sealed at rest
Sensible heat= 75W
Latent Heat=40W
Room Latent heat RLH= 40W
Additional heat gain not accounted above is safety factor 5% RLH
Total Room latent heat RLH=40 x 1+(0.05 x 40)= 42 W=0.042 kW

Load Due to Infiltration:
Outside air Latent Heat (OALH)
OALH = 50 x (cmm) x (W_o-W_i) in kW
Where cmm is cubic meter of air per minute
OALH= 50 x 0.168 x (0.0098-0.024) = 0.09372 kW
Total Latent Heat (TLH)
TLH=RLH + OALH
= 0.042+0.09372=0.13572 kW

Grand Total Heat
GTH=TSH+TLH
= 0.5417+0.13572=0.67742 kW

Sensible Heat Factor
= TSH/ (TSH+TLH) = 0.5417/0.677= 0.80

3.3 Load Calculations for A/C Test RIG

Calculation of Room sensible Heat

(I). Internal Heat Gain:
Occupancy Load
Heat Liberated at 23^0 C and 55%RH is when seated at Rest
Sensible heat = 75W
Latent heat=40W

(II). Lighting Load for the Florescent Bulb Q is
Q= Total Watt x 1.25
= 40 x 1.25
= 50W

(III) Appliance Load for equipment load
In case of iron equipment to be cooled
Sensible heat=2400 kJ / hr.
= 0.666 kW

(1V) Solar and Transmission Heat Gain through Walls Roof
`U` for corrugated card Board of thickness 0.002m= K/L= 0.064/0.002=32 W/m-K
Area= (Length of Wall x Height of Wall) =1.04 x 0.74= 0.769m^2
Q=32 x 0.7696 x (32-25) =197.0176W

(V) Transmission Gain through ceiling
U=32 W/m-K
Area of Ceiling = (Length x Width)
A= 0.74 x 0.5 = 0.37m^2
Q = 32 x 0.37 x (33-25) =94.72 W
RSH= occupancy Load +Light load+ appliance load+Solar & Transmission Heatgain through space.
RSH= 75 x 1+50+0.6666 +197.0176+94.72= 417.4042W
Additional heat gain not accounted above is safety factor of 5% RSH
Total room sensible Heat= 417.4042+(0.05 x 417.4042)
= 417.4042+20.87
= 438.274W= 0.438kW

Load Due to infiltration
Outside air sensible heat (OASH)
OASH= 0.0204 x cmm x (T_0-T_1)
= 0.0204 x (15 x π/4 x $(0.106)^2$ x (33-25)
= 0.0204 x 0.132 x 8 =0.0275 kW

Total Sensible Heat (TSH)
TSH = RSH+OASH
= 0.438+0.0275= 0.4655 kW

Calculation of Room latent heat:

Internal heat gain
Occupancy load
Heat liberated at 23^0 C and 55%RH is when sealed at rest
Sensible heat= 75W
Latent Heat=40W
Room Latent heat RLH=40W

Additional heat gain not accounted above is safety factor 5% RLH
Total Room latent heat RLH=40 x 1+ (0.05 x 40) = 42 W=0.042 kW

Load due to infiltration:
Outside air Latent Heat (OALH)
OALH = 50 x (cmm) x (W_o-W_i) in kW
Where cmm is cubic meter of air per minute
OALH= 50 x 0.168 x (0.024-0.0108) = 0.11088 kW
Total Latent Heat (TLH)
TLH=RLH + OALH
= 0.042+0.11088=0.15288 kW

Grand Total Heat
GTH=TSH+TLH
= 0.4655+0.15288=0.61838 kW

Sensible Heat Factor
= TSH/ (TSH+TLH) = 0.4655/0.61838= 0.75

3.4 Performance

Cooling Unit

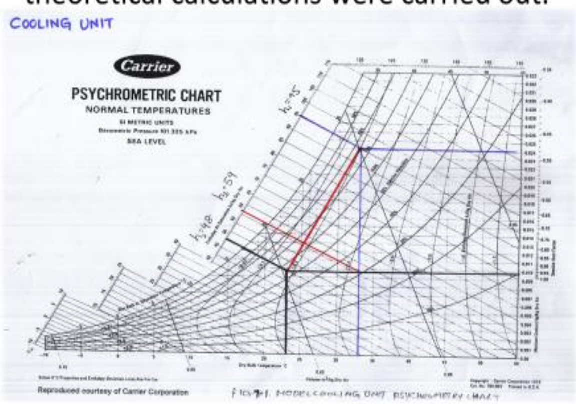
Time (min)	10	20	30	40	50	60
DBT ^{0}C	26.4	25.9	25.7	24	23.8	23
WBT ^{0}C	22.1	20.5	20	19.5	18.2	17

Test RIG

Time (min)	10	20	30	40	50	60
DBT ^{0}C	28.9	28.8	28.2	27.2	26	25
WBT ^{0}C	23.8	23.7	22.5	21.2	20	19

3.5 Psychrometry Charts

Using Pshycrohmetry chart the out door and indoor conditions are indicated and theoretical calculations were carried out.



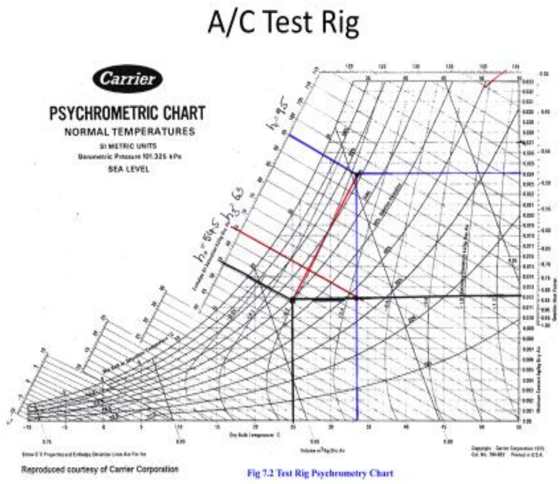


Fig 7.2 Test Rig Psychrometry Chart

3.6 Theoretical Calculations from Psychrometric chart:

3.6.1 Model cooling unit

Specific Humidity of air at outside condition (W_1)= 0.024 kJ / kg of dry air
Specific Humidity of air at inside condition (W_2) = 0.0098 kJ / kg of dry air
Specific volume (V_{S2}) = 0.852 m^3 / kg
Supply air per min. $V_2 = \frac{\pi}{4(0.106)2} * 15$
= 0.0088 x 15=0.132
Mass of air circulated (m_a) = V_2/V_{s2} = 0.132/0.852 = 0.154 Kg /min.
1. Mass of water drained
= m_a (W_1-W_2)
= 0.154(0.024-0.0098) X 60
=0.1312 kg /hr.

2. Capacity of cooling coil

Enthalpy of air at point 1, h_1 = 95 kJ/KG of dry air
Enthalpy of air at point 2, h_2 = 48 kJ/KG of dry air
Capacity of cooling coil = m_a (h_1-h_2)
= 0.154 (95-48) = 7.238 kJ/min
= 7.238/210 = 0.0344 TR

Enthalpy of air at point 3 h_3 = 59 kJ/Kg of dry air

3. Sensible heat Removed (SH) = m_a(h_3-h_2)
= 0.154(59-48)= 1.694kJ / min
= 1.694 x 60 = 101.64kJ/ hr.
4. Latent heat removed (LH) = m_a(h_1-h_3)
= 0.154(95-59)
= 5.544 kJ/min
=5.544 x 60 = 332.64 kJ/hr.

3.6.2 Air Conditioning Test Rig

Specific Humidity of air at outside condition (W_1)= 0.024kJ / kg of dry air
Specific Humidity of air at inside condition (W_2)= 0.0108kJ / kg of dry air
Specific volume (V_{S2})= 0.862m^3 / kg
Supply air volume V_2= 0.132 m^3 / min

Mass of air circulated (m_a) = V_2/V_{s2} = 0.132/0.862 = 0.153 Kg / min.

1. Mass of water drained
= m_a (W_1-W_2)
= 0.153 (0.024-0.0108) = 0.00201 x 60 = 0.1211 Kg /hr.

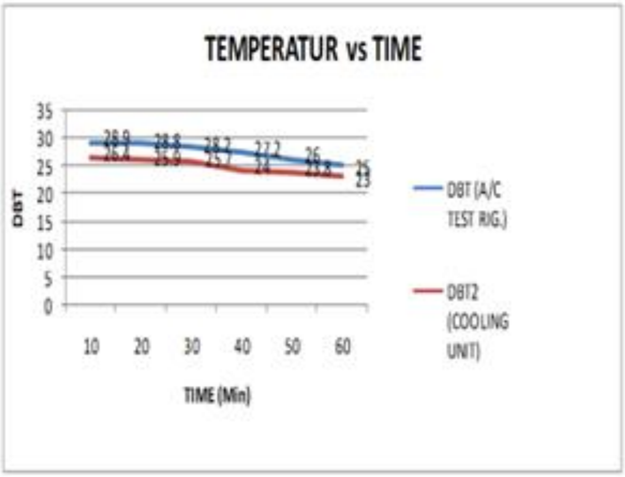
2. Capacity of the cooling Coil
Enthalpy of air at point 1, h_1= 95 kJ/Kg of dry air
Enthalpy of air at point 2, h_2 = 54.5 kJ/Kg of dry air
= m_a(h_1-h_2) = 0.153(95-54.5) = 6.1965 kJ / min
=6.1965/210 = 0.0295 TR
Enthalpy of air at point 3 h_3 = 63 kJ/Kg of dry air

3. Sensible heat = m_a (h_3-h_2)
= 0.153(63-54.5) = 1.3005 kJ/ min.
= 1.3005 x 60 = 78.03 kJ/hr.

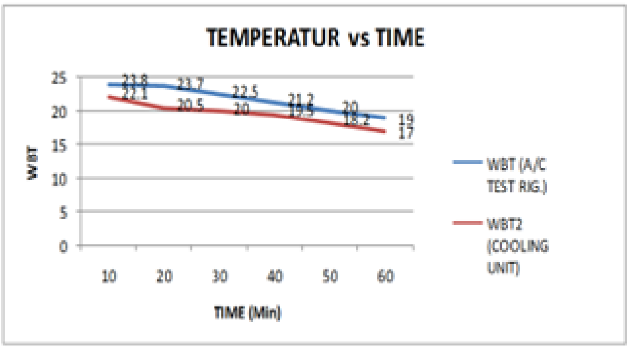
4. Latent heat removed (LH) = m_a(h_1-h_3)
= 0.153(95-63)
=4.896 x 60
= 293.76 kJ/ h

4. Graphs

Dry bulb temperature Vs Time



Wet bulb temperature Vs Time



5. Results

It can be observed that the performance of model cooling unit has been evaluated experimentally with air conditioning test Rig at ambient conditions of 33 ^{0}c DBT and 75% RH. The performance parameter such as capacity of a cooling coil, mass of water drained, sensible heat factor are given below.

Performance parameters	A/C Test RIG.	Model cooling unit
Mass of water drained	0.1211 kg/ hr.	0.1312 kg/ hr.
Capacity of cooling unit	0.0295 TR	0.0344TR
Sensible heat factor	0.75	0.80

Performance Comparison

Graph –I (Dry Bulb temperature V_s Time)

Graphs-I shows the variations in the Dry Bulb temperature to the time of the system for both model cooling unit and A/C test rig and found that better improvement in cooling for the model cooling unit than that of A/C test rig.

Graph –II (Wet Bulb temperature V_s Time)

Graphs-II shows the variations in the Wet Bulb temperature to the time of the system for both model cooling unit and A/C test rig and found that better cooling obtained for the model cooling unit than that of A/C test rig.

6. Conclusions

Experimental investigations have been carried out on a test space of dimensions 0.74 mx 0.5M and height 1.04 the space was alternatively cooled with a regular Air-condition test rig working on Vapour Compression Refrigeration System and with the Air Conditioning system designed with Dry ice and Thermo electric cooling modulus. The following observations are made.
Experiments are conducted on fabricated model cooling unit with TEC and dry ice for a space maintains with 23^{0}c DBT and 55% RH. It is observed that the performance parameters are improved nearly than that conventional air conditioning test Rig. The capacity of cooling coil is 0.0344 TR. The amount of water drained is 0.1312 kg/hr with sensible heat factor for 0.80.

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