An Experimental Investigation of a Hybrid Home Air Conditioner Using R134a Refrigerant

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Abstract: ***Now a day's Air conditioning has become the basic necessity for human comfort from the last decade due to global warming. Innovation in air conditioning technologies continues, with much recent emphasis placed on energy and efficiency. Reducing the power consumption is the main criteria because the cost of electricity is increasing day by day. In most of the air conditioners refrigerants such as R11, R12, R22 etc. are used which contributes depletion of the ozone layer. This thesis proposes that by using low temperature eco friendly refrigerants such as R134a can reduces the cooling time of the room. Experimental results also show the comparison of COP at various loads and different condenser fan speed regulations. In general for a 1 tr room dimensions the load in the equipment is not more than 10000KJ/hr but in our experimental analysis we tested our 1 tr equipment with full load conditions i.e., 12960KJ/hr. Hence proposed system could be a new option for performance improvement of a room air conditioner by enhancing heat transfer of the evaporator and useful for domestic establishments.***

Keywords: Cop – Refrigerant – Cooling Capacity – Ton of Refrigeration tr – Dry bulb Temperature – R134a – R22 – Carnot efficiency - Latent heat - Sensible Heat - Relative humidity - Specific humidity - Mass flow rate - volumetric efficiency.

1. Introduction

Innovation in air conditioning technologies continues, with much recent emphasis placed on energy and efficiency. Reducing the power consumption is the main criteria because the cost of electricity is increasing day by day so we have to reduce the power and coefficient of performance of the system should be increased or maintained the same. Selection of refrigerant is another criterion. Most refrigerants such as R11, R12, R22etc, used for air conditioning contributes depletion of the ozone layer as these refrigerants consists of Chlorine and Carbon atoms. It is very difficult to replace any other refrigerant with the existence product by the companies even though the new refrigerants are eco-friendly. In most countries manufacturing and use of CFCs has been banned or severely restricted due to concerns about ozone depletion.

The present thesis proposes that by using low temperature eco-friendly refrigerants can reduces the cooling time of the room and thereby reducing the power consumption. Working fluid selection for the refrigeration and air conditioning applications is based on three factors: safety (toxicity and flammability), environmental impact (stratospheric ozone), and performance. Considering these three, we used R134a as refrigerant.

Experimental results also shows the preheat of the refrigerant i.e., R134a to 5^0 C before entering into compressor which gives the maximum coefficient of performance and the comparison of COP at various loads and different condenser fan speed regulations. In general for a 1 tr room dimensions the load in the equipment is not more than 10000KJ/hr but in our experimental analysis we tested our 1 tr equipment with full load conditions i.e., 12960KJ/hr. Hence proposed system could be a new option for performance improvement of a room air conditioner by enhancing heat transfer of the evaporator and useful for domestic establishments.

2. Experimental Setup

The vapour-compression uses a circulating liquid refrigerant as the medium which absorbs and removes heat from the space to be cooled and subsequently rejects that heat elsewhere. All such systems have four components: a compressor, a condenser, a thermal expansion valve, and an evaporator. Circulating refrigerant enters the compressor in the thermodynamic state known as a saturated vapour and is compressed to a higher pressure, resulting in a higher temperature as well. The hot, compressed vapour is then in the thermodynamic state known as a superheated vapour and it is at a temperature and pressure at which it can be condensed with cooling air. That hot vapour is routed through a condenser where it is cooled and condensed into a liquid by flowing through a coil with cool air flowing across the coil. This is where the circulating refrigerant rejects heat from the system and the rejected heat is carried away by either the air (whichever may be the case).

The condensed liquid refrigerant, in the thermodynamic state known as a saturated liquid, is next routed through an expansion valve where it undergoes an abrupt reduction in pressure. That pressure reduction results in the adiabatic flash evaporation of a part of the liquid refrigerant. The auto-refrigeration effect of the adiabatic flash evaporation lowers the temperature of the liquid and vapour refrigerant mixture to where it is colder than the temperature of the enclosed space to be refrigerated.

Our outdoor unit specifications are compressor – 0.75hp of emerson climate technology's limited model KCJ498HAG having 2.591×10^{-5} m^2 displacement and 2800 rpm speed, condenser pipe perimeter 2.5cm condenser pipe length 1.35m condenser fan motor rated speed 1175rpm. We want to know the thermal performance that's why we set the pressure and temperature gauges at suction and discharge of the compressor i.e., at evaporator outlet and condenser inlet respectively. We did 3 regulations of condenser fan speed to determine the optimum speed.

Our indoor unit comprises of capillary tube as expansion device and evaporator coil, blower. We used 0.66'' dia and 5 feet length of capillary. We used napoleon AS- 12c53f150I4 model indoor unit. Mass flow rate of air coming from our evaporator blower is 633.87 kg/hr it is an observed value by using density of air and volume.

We fabricated an insulated cabin having dimensions of 8 × 4 × 8 feets of size and we applied sensible heat through electrical bulbs.

Load calculations for 1tr equipment:

- Dimensions for 1tr AC = 120 sq feet
- Room Dimensions = L × W × (4 × H) considering dimensions = 8 × 4 × 8 feets
- Heat flow due to the conduction (wall):
  Q = UA (To – Ti)
  U = overall heat transfer coefficient
  To = Outside Temp.
  Ti = Inside Temp.
  A = Area through which heat is transferred
  Q = 1.45 × 8.95 × (32 – 19)
  Q = 168.7 W
  Q = 0.168 KW

➢ Heat flow through Window:

➢ Size 4 × 4 Area = 1.218 × 1.218 m^2
Q = UA (To – Ti)
U = 5.5 w/m2 k
Q = 5.5 × (1.218 × 1.218) × (32 – 19)
Q = 106.07 W
Q = 0.106 W

➢ Heat flow through the door:
Thickness = 3.75 cm
Door size = 76 × 4 feets = 2.59 m^2
U = 2.55 W/m2 k
Q = UA (To – Ti)
Q = 2.55 × 2.59 (25 – 19)
Q = 39.6 W = 0.0396 KW

➢ Heat flow in bedroom:

➢ Total Heat = 4 occupants + 2 windows + door + 1 TV + 2
➢ Bulbs + 2 Walls + 1 ceiling
1 TV = 100 watts
1 occupants = 0.15 KW for 4 occupants = 0.15 × 4 = 0.6 KW
for 1 window = 0.106 KW
door = 0.0396 KW
for 1 bulb = 0.2 KW for 1 wall = 0.168 KW
Total Heat = 0.6 + 2 × 0.106 + 0.0396 + 0.100 + 2 × 0.2 + 2 × 0.168 + 0.168
Total Heat = 1.85 KW

➢ Heat flow in Hall:
Total Heat = 6 occupants + 2 bulbs + 1 TV + 1 System + door + 2 windows + 2 walls + 1 ceiling
Total Heat = 6 × 0.150 + 2 × 0.2 + 0.100 + 0.080 +0.0396 + 2 × 0.106 + 2 × 0.168 + 0.168
Total Heat = 2.235 KW

For 1 tr room dimensions we calculated load for hall and bedroom. The maximum load on the equipment for 1 tr room dimensions is not more than 10000 Kj/hr. But we test our equipment with 12960 Kj/hr i.e., full load conditions. We put 18 electrical bulbs of 200 W and each bulb gives 720 KJ/hr heat, the below figures shows our fabricated model.

3. Observations and Results

We set indoor temp as 19^0C

Condenser Fan Speed 1175 rpm, full Load 12960 KJ/hr

Table 3.1: Condenser Fan Speed 1175 rpm, 3/4th Load 10080 KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp °C	Discharge Temp °C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	28.3	40	29	15	7	42	214	448	5.33	1.276	0.327	0.429	47	7.717
2	10:00	29.5	35	24.6	17	8	46	215	446	6.48	1.728	0.40	0.51	50	7.025
3	11:00	29.7	46	24.4	16	8	48	212	464	6.48	1.29	0.60	0.51	49	7.021
4	12:00	30.2	31	24.4	17	8	48	175	464	6.48	1.8	0.32	0.42	40.5	7.025
5	01:00	30.8	34	23.4	19	8	50	191	480	6.4	1.32	0.35	0.46	44	6.69
6	02:00	33.1	36	25	26	9	50	216	480	6.4	1.72	0.40	0.52	50	6.69
7	03:00	32	30	28	26	9	50	245	480	6.4	1.38	0.45	0.59	46	6.69
8	04:00	30.5	29	30	18	8	50	233	480	6.28	1.26	0.412	0.56	54	7.39
9	05:00	30	30	34	18	6	46	375	464	6.4	1.08	0.68	0.9	86	7.56
10	06:00	29	30	36	18	6	44	493	440	3.87	1.07	0.54	0.74	71	6.05

Table 3.2: Condenser Fan Speed 1175 rpm, 3/4th Load 10080 KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	30.9	50	30	16	8	48	328	448	6.4	1.12	0.60	0.78	65	6.69
2	10:00	30	41	28.9	14	8	50	206	448	6.4	1.4	0.37	0.49	40	6.69
3	11:00	32.4	49	25.4	16	8	48	330	464	6.1	1.2	0.57	0.7	65	6.69
4	12:00	32.5	59	25.4	28	8	48	316	480	6.1	1.42	0.55	0.7	62	6.69
5	01:00	32.5	52	24.3	17	8	50	224	448	6.1	1.72	0.39	0.53	44	6.69
6	02:00	32.7	57	28	20	8	49	229	480	6.1	1.68	0.40	0.55	45	6.69
7	03:00	30.3	44	28.2	15	8	50	282	480	6.1	1.21	0.49	0.67	50	6.69
8	04:00	30	42	28	15	8	50	285	480	6.1	1.29	0.50	0.68	50	6.69
9	05:00	29	40	28	20	8	46	392	464	5.17	1.2	0.57	0.78	60	7.39
10	06:00	28	42	28	20	8	46	338	440	5.17	1.2	0.50	0.6	50	7.39

Table 3.3: Condenser Fan Speed 1175 rpm Load ½ Load 6480 KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	25.8	46	22.9	22	4	42	391	432	5.37	0.589	0.60	0.78	70	7.2
2	10:00	29.4	47	23.3	22	4	44.5	412	432	5.09	0.648	0.60	0.79	71	6.7
3	11:00	29.8	44	21.1	25	5	46	399	432	5.09	0.81	0.60	0.79	71	6.7
4	12:00	29.8	44	20.4	24	4	47	328	432	5.09	0.732	0.47	0.63	61	6.4
5	01:00	30.5	37	20.1	26	4	47	378	448	5.09	0.74	0.55	0.73	70	6.4
6	02:00	31.5	43	20.0	28	4	47	412	456	5.09	0.81	0.60	0.79	76	6.4
7	03:00	32.0	41	19.6	31	4	47	277	440	5.09	0.691	0.45	0.53	50	6.4
8	04:00	29.8	42	19.1	28	3	46	412	432	4.7	0.48	0.55	0.72	70	6.4
9	05:00	27.7	44	18.5	30	3	46	449	424	4.7	0.72	0.60	0.79	76	6.4
10	06:00	29	42	16.8	39	3	46	449	450	4.7	0.648	0.64	0.79	76	6.4

Table 3.4: Condenser Fan Speed 1175 rpm Load 1/4th Load 3600 KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	25.8	82	18.6	77	-1	44	460	412	4.52	0	0.60	0.79	87	6.044
2	10:00	25.9	79	18.6	78	-1	44	484	410	4.52	0	0.62	0.83	91.5	6.044
3	11:00	28.6	78	18.8	75	-1	44	460	410	4.52	0	0.603	0.79	76	6.044
4	12:00	28.7	73	19	72	2	45	431	410	4.46	0	0.55	0.739	80	6.395
5	01:00	28.7	69	18.5	64	2	45	549	415	4.46	0	0.704	0.94	90	6.395
6	02:00	28.6	71	18.6	66	2	45	507	415	4.46	0	0.65	0.869	95.2	6.395
7	03:00	28.3	61	18.6	67	-1	44	519	410	4.54	0	0.67	0.89	98	6.044
8	04:00	27.6	64	18.6	65	-1	44	500	410	4.52	0	0.65	0.85	93	6.044
9	05:00	26.7	70	18.5	73	-1	44	465	410	4.52	0	0.603	0.796	87.7	6.044
10	06:00	26.1	69	18.3	72	-1	44	348	410	4.52	0	0.452	0.596	65.7	6.044

Table 3.5: Condenser Fan Speed 1175 rpm No Load

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	26.1	68	26.1	19.5	5	46	136	432	5.46	0.5	0.25	0.27	26	6.7
2	10:00	27.4	68	19.3	14	5	46	136	432	5.46	0.5	0.25	0.27	26	6.7
3	11:00	27.1	64	18.5	14	5	46	136	432	5.46	0.45	0.30	0.38	37	6.7
4	12:00	30.2	54	22.7	15.5	5	45	194	440	5.46	0.366	0.55	0.71	68	6.7
5	01:00	28.2	56	22	13.9	5	45	356	432	5.46	0.42	0.45	0.55	53	6.7
6	02:00	29.1	56	19	12.8	5	45	277	448	5.46	0.375	0.25	0.32	28	6.7
7	03:00	29	58	20.0	13.9	5	45	162	432	5.18	0.45	0.60	0.77	35	6.7
8	04:00	28	59	19.9	13	5	46	388	432	5.18	0.5	0.25	0.32	31	6.7
9	05:00	27.4	66	22	15.2	5	46	164	432	5.18	0.36	0.30	0.38	37	6.7
10	06:00	25	56	21	12	5	46	194	416	5.18	0.42	0.50	0.64	62	6.7

Table 3.6: Condenser Fan Speed 846 rpm 3/4th Load 10080KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	29.3	41	27.9	26	6	48	469	432	4.47	1.26	0.60	0.78	75	6.6
2	10:00	30.3	34	27.7	21	6	50	339	432	3.60	1.176	0.35	0.47	45	6.3
3	11:00	31.7	35	27.4	19	6	50	488	440	3.60	1.05	0.50	0.68	65	6.3
4	12:00	31	41	27.3	24	6	50	533	440	3.60	0.924	0.55	0.74	71	6.3
5	01:00	31.2	35	28.2	17	6	50	534	440	3.60	1.155	0.55	0.74	71	6.3
6	02:00	30.9	42	26.6	23	6	50	534	438	3.6	1.15	0.55	0.74	71	6.3
7	03:00	29.5	34	28	18	6	50	436	440	3.6	0.945	0.45	0.60	58	6.3
8	04:00	28.5	34	27.6	15	6	48	391	432	4.47	1.2	0.50	0.65	62	6.6
9	05:00	27.6	29	28.5	18	6	48	313	432	4.47	1.68	0.40	0.52	50	6.6
10	06:00	27	36	27	15	6	48	353	432	4.47	1.26	0.45	0.58	56	6.6

Table 3.7: Condenser Fan Speed 846 rpm ½ Load 6480 KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	25.3	48	20.6	27	4	48	352	416	4.8	0.90	0.50	0.6	63	6.29
2	10:00	27.5	49	21	28	4	48	352	424	4.8	0.60	0.50	0.6	65	6.29
3	11:00	27.9	46	20	29	4	48	352	424	4.8	0.48	0.50	0.6	65	6.29
4	12:00	27.8	42	20	27.5	4	48	316	424	4.8	0.40	0.45	0.59	57	6.29
5	01:00	28	39	19.8	28	4	48	387	424	4.8	0.60	0.55	0.72	69	6.29
6	02:00	28.7	39	20	29	4	48	383	424	4.1	0.7	0.45	0.59	56	6.69
7	03:00	28.3	39	19.8	31	4	48	352	424	4.8	0.5	0.50	0.66	60	6.29
8	04:00	26.6	39	19.2	29	4	48	277	416	4.1	0.4	0.40	0.52	50	6.29
9	05:00	24.9	40	18.9	31	4	48	211	416	4.1	0.25	0.30	0.39	38	6.29
10	06:00	25	41	19	35	4	48	246	416	4.8	0.41	0.35	0.46	44	6.29

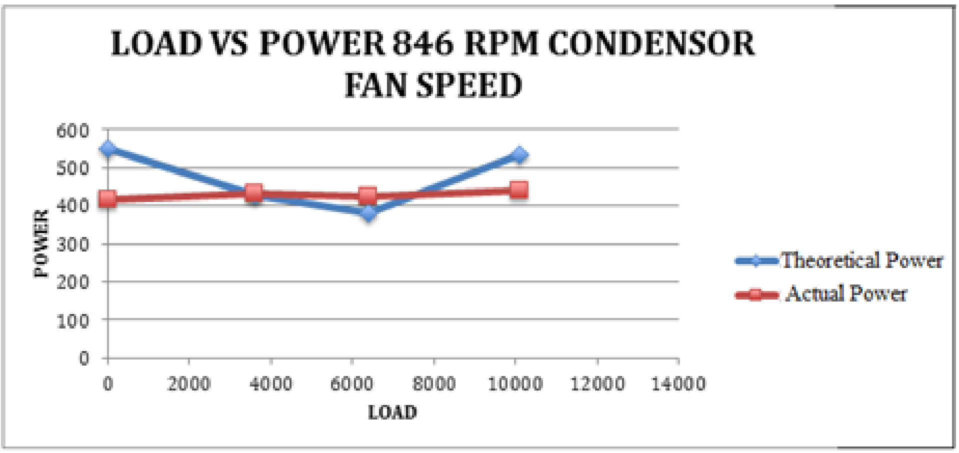
Table 3.8: Condenser Fan Speed 846 rpm 1/4th Load 3600 KJ/hr

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	25.7	65	19.5	56	3	46	206	432	5.08	0.90	0.30	0.38	37	6.41
2	10:00	26.9	54	20.6	47	3	46	206	424	5.08	0.60	0.30	0.36	37	6.41
3	11:00	28.6	50	21.9	44	3	46	274	428	5.08	0.48	0.40	0.5	49	6.41
4	12:00	27.6	51	21.5	38	4	47	430	432	5.18	0.40	0.60	0.8	77	6.18
5	01:00	28.2	51	19.3	41	4	47	274	440	5.18	0.60	0.40	0.5	49	6.41
6	02:00	27.5	49	19.9	46	4	47	251	456	5.18	0.70	0.35	0.47	45	6.18
7	03:00	27.8	48	19.5	47	4	47	179	424	5.18	0.50	0.25	0.33	31	6.18
8	04:00	26.6	53	20.5	43	3	46	274	424	5.18	0.40	0.4	0.5	49	6.41
9	05:00	24.4	48	20.3	34	3	46	171	416	5.18	0.25	0.25	0.32	30	6.18
10	06:00	22.7	56	20	43	3	46	377	416	5.18	0.41	0.55	0.70	69	6.18

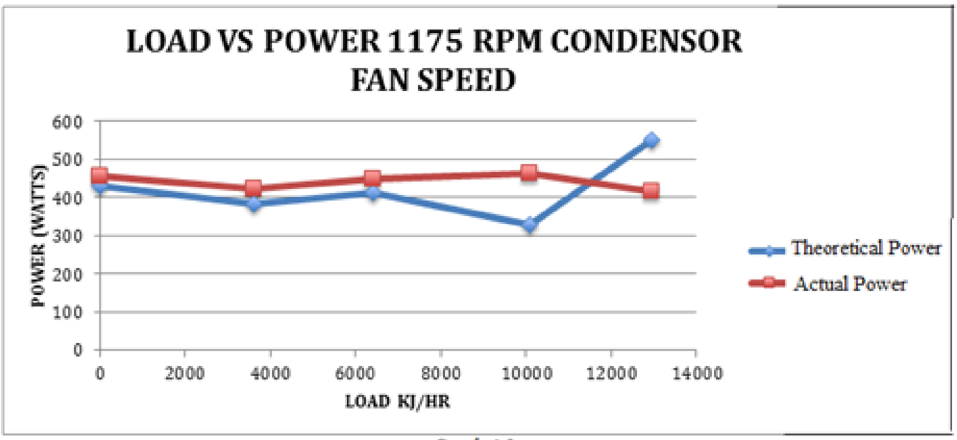
Table 3.9: Condenser Fan Speed 846 rpm No Load

S.No	Time	Outdoor Conditions		Indoor Conditions		Suction temp ^0C	Discharge Temp ^0C	Power Consumption		COP	Theoretical Cooling Capacity Tr	Practical Cooling Capacity tr	Mass Flow Rate of refrigerant Kg/min	Volumetric Efficiency of compressor %	Carnot η
		Ambient Temp. (T_{d1}) °c	Relative Humidity (ϕ_1) %	Dry Bulb Temp. (T_{d2}) °c	Relative Humidity (ϕ_2) %			Theoretical Power Watts	Practical power consumption Watts						
1	09:00	25.8	82	18.6	77	-1	44	460	412	4.52	0	0.60	0.79	87	6.044
2	10:00	25.9	79	18.6	78	-1	44	484	410	4.52	0	0.62	0.83	91.5	6.044
3	11:00	28.6	78	18.8	75	-1	44	460	410	4.52	0	0.603	0.79	76	6.044
4	12:00	28.7	73	19	72	2	45	431	410	4.46	0	0.55	0.739	80	6.395
5	01:00	28.7	69	18.5	64	2	45	549	415	4.46	0	0.704	0.94	90	6.395
6	02:00	28.6	71	18.6	66	2	45	507	415	4.46	0	0.65	0.869	95.2	6.395
7	03:00	28.3	61	18.6	67	-1	44	519	410	4.54	0	0.67	0.89	98	6.044
8	04:00	27.6	64	18.6	65	-1	44	500	410	4.52	0	0.65	0.85	93	6.044
9	05:00	26.7	70	18.5	73	-1	44	465	410	4.52	0	0.603	0.796	87.7	6.044
10	06:00	26.1	69	18.3	72	-1	44	348	410	4.52	0	0.452	0.596	65.7	6.044

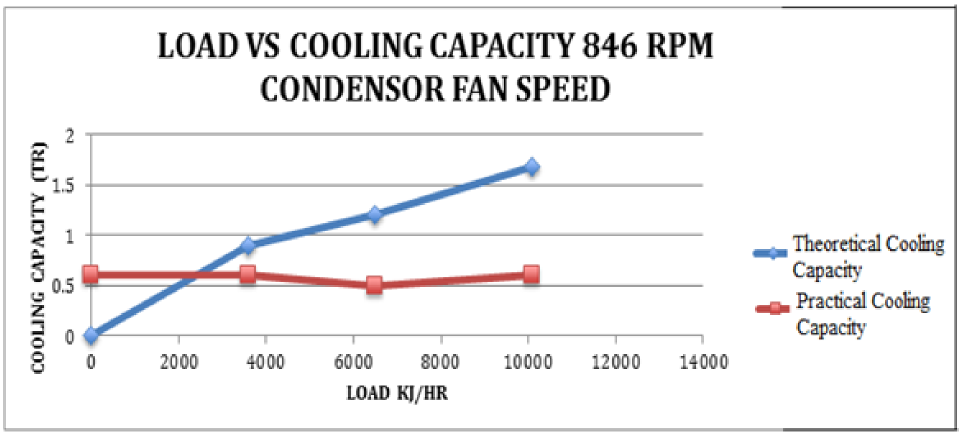
4. Graphs



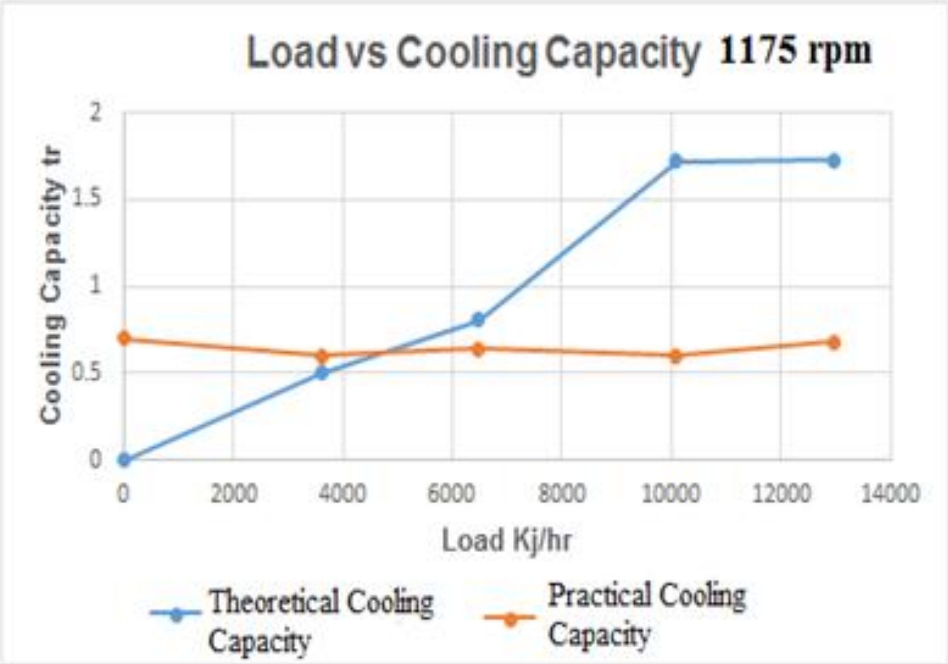
Graph 4.1



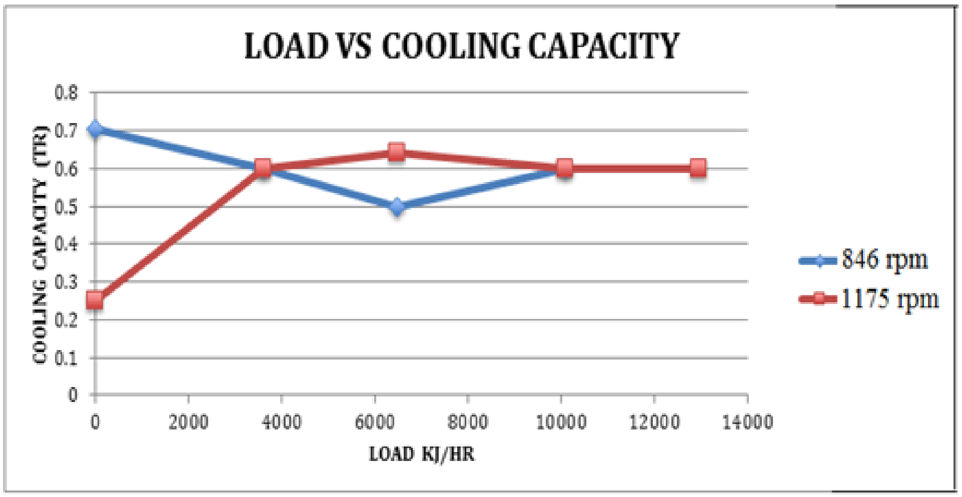
Graph 4.2



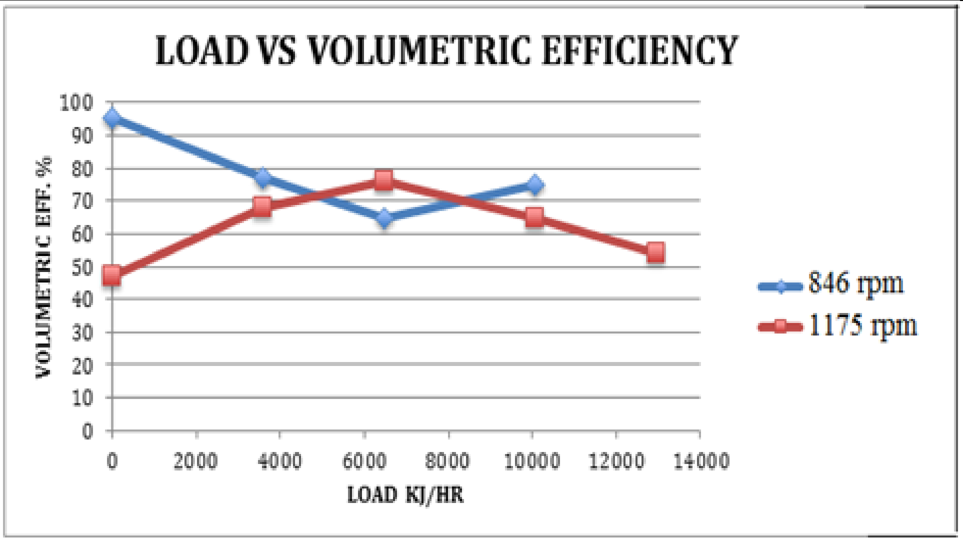
Graph 4.3



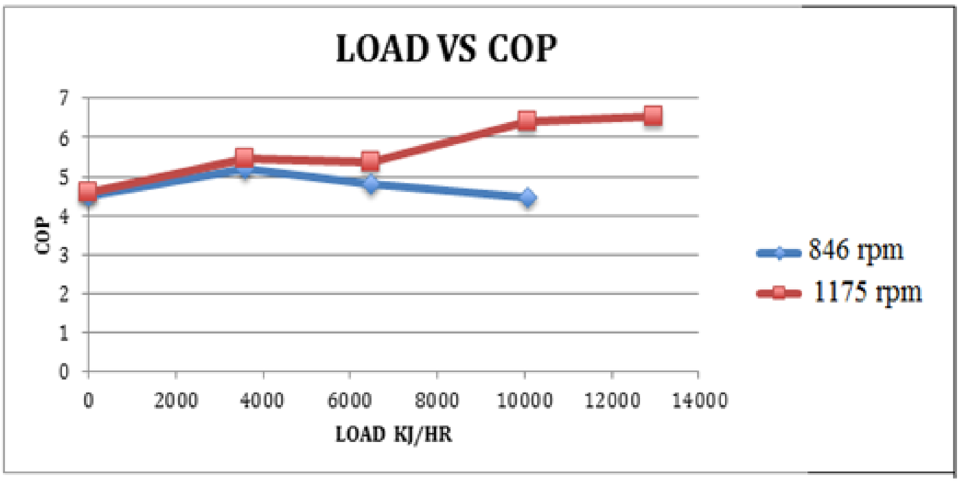
Graph 4.4



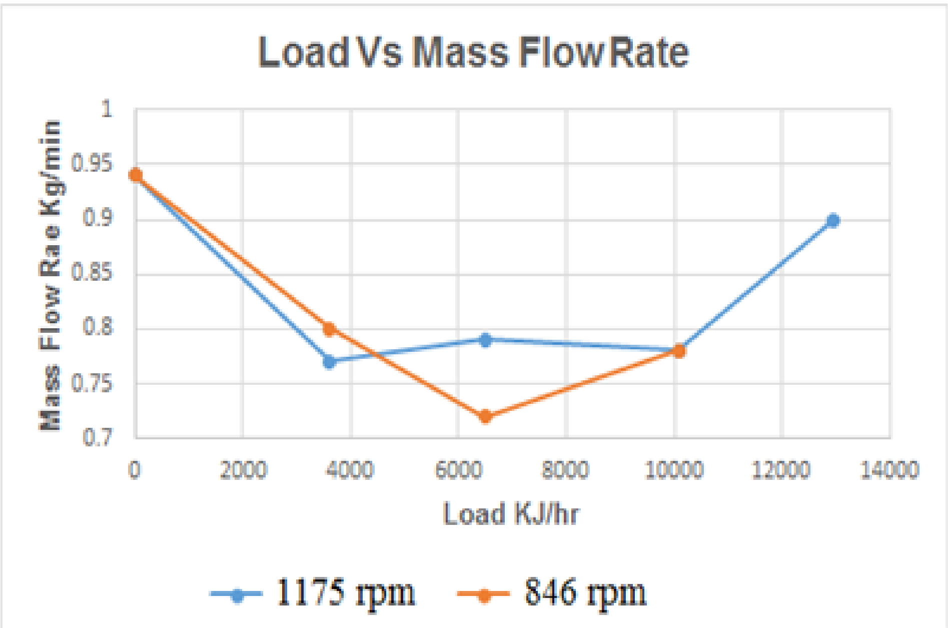
Graph 4.5



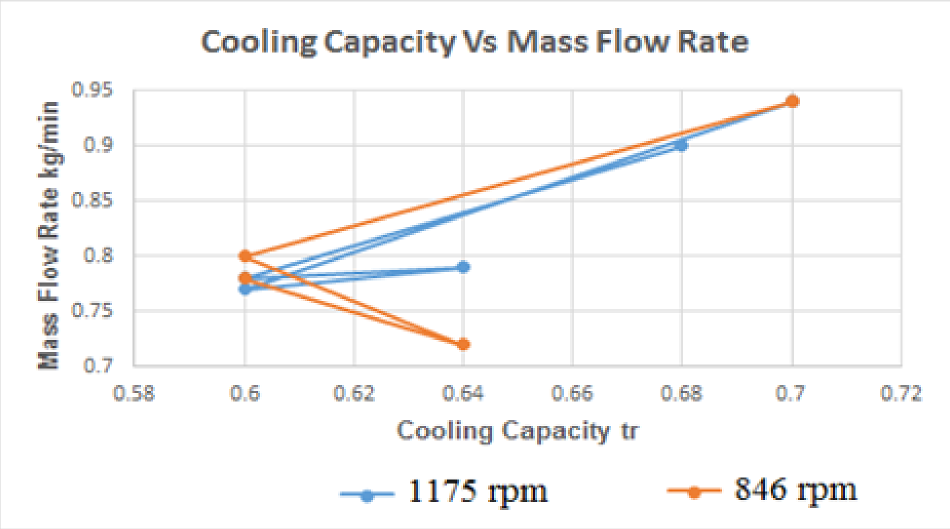
Graph 4.6



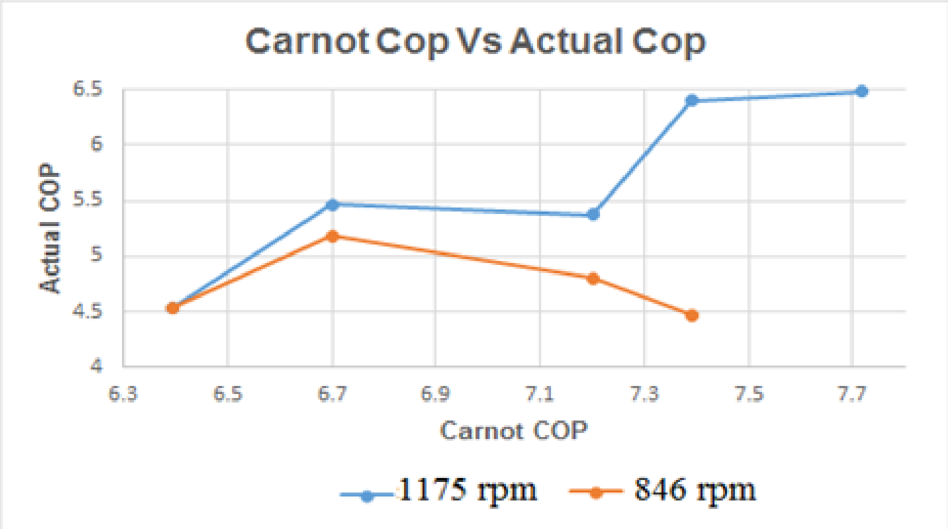
Graph 4.7



Graph 4.8



Graph 4.9



Graph 4.10

5. Model Calculation

Model Calculations ¾ th load value 1

From Psychometry:

Dry Bulb Temp. Ambient T_{d1}=27.7^{0}C
Relative Humidity Φ_1=32%
Wet bulb Temp. T_{w1}=16.1^{0}C
Specific Humidity ω_1=0.007 KJ/KG of dry air

Dry Bulb Temp. Inside the cabin T_{d2}=25.8^{0}C
Relative Humidity Φ_2=26%
Wet Bulb Temp. Inside the cabin T_{w2}=13.9^{0}C
Specific Humidity ω_2=0.005 KJ/KG of dry air

Cooling coil temp. i.e.; Suction temp T_{d6}=2^{0}C
Bulb Temp. at entry condition of air T_{d3}=26^{0}C
Wet Bulb Temp.at entry condition of air T_{w3}=14.5^{0}C

From Psychrometry Chart

By Pass Factor (BPF) =Length 4-6/Length 3-6=0.74Dry

Room sensible Heat=10080 KJ/HR Room Latent Heat=504 KJ/HR

Room Sensible Heat Factor=R.S.H / R.S.H+L.S.H=0.95 from B.P.F=T_{d4}-T_{d6}/ T_{d3}-T_{d6}

DBT at exit condition of air=T_{d4}=19.2^{0}C
WBT at exit condition of air T_{w4}=12^{0}C

From Psychrometry Chart:

h_1=46 KJ/KG h_2=39 KJ/KG h_3=41 KJ/KG

$h_{4=}$34 KJ/KG h_5=20 KJ/KG h_6=13 KJ/KG

Total Mass of air flowing M_4=

Room Latent Heat/h_2-h_4=10584/39-34=2116.8 KJ/HR

Mass of fresh air $m_{r=}$5% of m_a=105.84 KG/HR

Theoretical Cooling Capacity=

Q=m_a(h_3-h_4)/ 60x120=1.17

Practical Cooling Capacity=

tr Q=m_a(h_3-h_4)/60x210=0.35 tr

From Refrigerant Properties:-

Suction pressure P_1=1.84 bar Suction temp. T_1=2^{0}C h_1=610 KJ/KG h_2=640 KJ/KG

Discharge Pressure P_2=10.53 bar Discharge Temp. T_2=46^{0}C h_{f3}=460 KJ/KG

COP=h_1-h_{f3}/h_2-h_1=5

$Q_{practical}$=m_r(h_1-h_{f3})/210 mass flow of refrigherant m_r=0.49 KG/MIN

Theoretical Power Consumption P=m_r(h_2-h_1)/60 =245 watts

Volumetric Efficiency of Compressor η_v=m_rxV_1/$_{\text{Displacement X speed}}$ = m_r x V_1/2.591x10^{-5}x2800

0.49x0.07/2.591x10^{-5}x2800= 47%

Carnot Efficiency η_{carnot} = T_1/T_2-T_1 = 6.25

6. Conclusion

Experimental test have been carried out to investigate the performance improvement of room AC. For experimentation we have used R134a refrigerant which is eco-friendly and the properties of R-134a will satisfies the requirements of AC. Experiment were carried out at different load and at different condenser fan speed to investigate the performance of the room AC. The following conclusions were drawn based on experimental results.

1) Experimental results show that, at the 3/4th load we got the maximum cop of 6.4 which is very closer to the carnot cop.
2) We achieved the better cooling capacity of 0.7 tr at the utmost 3/4th load.
3) The volumetric efficiency of the compressor is in between 75 % - 90% at all loads which is a better indication of working of a compressor and therefore the power consumption is just above 130 watts at the maximum load i.e., 3/4th Load.
4) Experimental results shows that the cop of the refrigeration system is increasing with the load which indicates the rate of cooling capacity is also increasing proportionately with load.
5) Since the rate of cooling capacity in the conditioned space is high therefore the power consumption decreases per unit cooling.

From the results R134a achieves the required room temperature in very fast manner. It is very good advantage in R134a room AC.

Government of India and Supreme Court has ordered to replace the present refrigerant R22 because of ozone depletion. Based on our results we concluded that R134a is best alternative for R12 and it achieves all the properties of R22. So R134a becomes one of the alternatives in future. R22 refrigerant is a single hydrochlorofluorocarbon (HCFC) compound that contains hydrogen, chlorine, fluorine and carbon. R134 refrigerant is a single hydrofluorocarbon (HFC) refrigerant that contains hydrogen, fluorine and carbon. It does not contain chlorine, which makes it more

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environmentally-friendly than R22 refrigerant. R22 is less stable than R134 because, when the hydrogen compound breaks down in the atmosphere, it releases chlorine before it reaches the stratosphere. The chlorine then reacts with the oxygen molecules in the ozone to create new molecules that result in ozone depletion.

References

[1] Zubair, S.M., Yaqub, M. and Khan, S.H., Second law based thermodynamic analysis of two stage and mechanical sub-cooling refrigeration cycles, International Journal of Refigeration, Vol. 19(8), (1996), pp. 506-516
[2] Park, K.J. and Jung, D., Thermodynamic performance of R502 alternative refrigerant mixtures for low temperature and transport applications, Energy Conversion and Management, Vol. 48, (2007), pp. 3084-3089
[3] Qureshi, B.A., Zubair, S.M., The effect of refrigerant combinations on performance of a vapor compression refrigeration system with dedicated mechanical sub-cooling, International Journal of Refrigeration, Vol. 35(1), (2012), pp. 47-57
[4] http://www.peakmechanical.ca/
[5] http://en.wikipedia.org/wiki/Refrigeration
[6] http://nptel.ac.in/

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