DEVELOPMENT OF ECONOMICAL AIR-CONDITIONING SYSTEM USING DUAL COMPRESSORS FOR DOMESTIC AND SMALL SCALE BUSINESS FACILITY CONSUMERS

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DEDICATION

I would like to dedicate my Thesis to my parents for their love and spiritual support throughout my career.

Names of G.E.C. Members

Dr. Salman Nisar

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NOMENCLATURE

μF= Micro Farad

Amp = Current in ampere

CFM= Air Flow in Cubic Feet per Minute

RT= Refrigeration Ton

Ton= Tonnage of Refrigeration

Hz= Frequency in Hertz

V= Potential Difference in volts

CHW= Chilled Water

℃= Degree Celsius

L/Hr= Liter per Hour

GPM= Gallon per Minute

GPH= Gallon per Hour

AC= Air Conditioner

rpm = Round per Minute

ABSTRACT:

Split, Window & Floor Standing type Air-Conditioners are being used for decreasing inside temperatures of air conditioned spaces. These ACs have a compressor, a condensing unit, an evaporator, a fan for circulating air against chilled coils including an expansion device. These appliances use Electrical energy as input which is becoming expensive gradually and the demand is also increasing at universal level regularly. The compressor consumes maximum proportion of energy in an air-conditioner. In this research primary refrigerant used was R-134a while the secondary refrigerant was chilled water which has been used to produce air-conditioning effect using dual compressors. A domestic room was selected as a testing facility in which Dual Compressor AC and conventional split ACs were installed and their power consumption were compared. It was observed that 12.5% reduction in power consumption was noted using Dual compressors AC as compared to conventional AC.

CHAPTER #1

INTRODUCTION

Air-conditioning is control of temperature, moisture, cleanliness of air and its quality and air circulation in space as required by occupants, process or products [34]. An air conditioning system consists of an air conditioning plant and a thermal distribution system. Air, water or refrigerant are used as fluid media for transferring energy from plant to the conditioned space or from the conditioned space to the plant. A thermal distribution system is required to circulate the media between the conditioned space and the plant [35].

1.1 WHAT IS DONE IN AIR-CONDITIONING

Following are the processes that take place in air-conditioning. They include cooling, heating, humidifying/dehumidifying, cleaning, distributing air to various places, admitting outdoor air and to exhaust air.

1.2 COOLING TECHNIQUES IN AIR-CONDITIONING:

Fluids are cooled using different type of machines and it can be air, liquid or mixture of liquid and vapors and the selection of machine for cooling depends upon the type of fluid used. Like Air is cooled in AHU, Water is cooled in Chillers and Refrigerants are processed by Heat Pumps commonly termed as Compressors.

1.3 TYPES OF REFRIGERATION COMPRESSORS:

A. Reciprocating
   i. Rotary
   ii. Double Rotary
B. Scroll

1.3.1 TYPES OF CHILLERS:

A. Electric Chillers

i. Reciprocating
ii. Screw
iii. Centrifugal

B. Absorption Chillers

i. Natural Gas/Direct fired
ii. Steam Fired
iii. Hot water

1.4 AIR-CONDITIONING APPLICATIONS:

Air conditioning systems are found in many applications like, residences, auditoriums, gymnasiums, offices, schools and universities, health care facilities, hotels, stores and shopping centers, manufacturing facilities, ships, trains, airplanes, submarines etc [34].

1.5 PROBLEM STATEMENT:

Residential and small scale business facility air conditioning is expensive in terms of operational cost since long and serves as a major contributor on utility bills. Considering the energy crisis and increase in utility bills Tariff on regular basis in our country, it is extremely necessary to find ways that suggest ways to reduce energy consumption wherever possible. Especially in summer season, there generates a big gap of demand and supply of electrical energy, while the requirement of air conditioning also increases simultaneously. So, the main focus in this research was lowering operational cost of air-conditioning.

1.6 SUMMARY ON FURTHER CHAPTERS:

In chapter # 2, literature reviews are presented with various aspects related to this research in which different modes of capacity modulation and energy saving methods are encountered. While the experimentation is presented in chapter # 3 in which research methodology and construction of dual compressor AC is expressed along with details of necessary components used in this research. Chapter # 4 contained Collection of Data on various parameters like chilled water temperature and its flow rates including electrical consumption for both Dual Compressor and Conventional AC, followed by Analysis performed on data. In Chapter # 5, conclusion is provided on behalf of experimental study and the directions are provided to future researchers. A nomenclature is also provided before abstract for symbols used in research.

CHAPTER #2

LITERATURE REVIEW

The literature review was carried out on different technologies of air-conditioners available in the market. The review mainly focuses different models of Dual Compressors ACs and Inverter type ACs.

2.1 DUAL COMPRESSOR WORKING BASED ON DUAL CYLINDERS:

Koji at al. [1] created a highly efficient dual compressor working on the principle of dual cylinders in which one cylinder has larger capacity while the other one has smaller capacity. The idea was to use these cylinders according to thermal load i.e. when cooling load was higher, the bigger system was working and smaller system was in operation when the cooling load was reduced. By working of this mechanism, the energy consumption was reduced remarkably as well as the users had better comfort as they found environment according to instantaneous need. They also claimed that the intermittent operation of twin rotary compressor did not only cause power wastage but also affected user comfort. While in their novel design, the operation from two cylinders moved to single cylinder near temperature set point which overcome certain drawbacks. They also optimized winding and stator of the motor and achieved an overall efficiency increase of 3 to 5 % and power consumption by 44%.

Qiu Tu at al. [2] developed an experimental setup using digital scroll compressor for capacity modulation in order to respond promptly against indoor heated loads by variation of refrigerant flow without using inverter technology and at the same time avoided the intermittent start/stop phenomena of conventional compressors. The system worked on the principle that when the pulse width modulation valve was closed, then utmost magnitude and refrigerant mass flow was achieved. This state was termed as loaded state. And in unloaded state, the pulse width modulation valve was opened, hence no magnitude or refrigerant flow was attained. By variation in time of loaded / unloaded conditions, a wide capacity range from 10 to 100% can be attained.

Jeongbae at al. [3] stated multistage capacity by rotary compressor named Digital Rotary Compressor was achieved by newly manufactured design which provided continuous capacity modulation mechanically without using any electronic frequency modulation. Conventional air-conditioner having a single capacity consumed more electric power as compared to a variable capacity air-conditioner. The power consumption of a single capacity air-conditioner didn’t change according to the variation of the temperature around the indoor unit. Capacity modulation of DRC was achieved by controlling the vane motion with PWM control. DRC was composed of two pumps like a conventional twin rotary compressor. The capacity modulation was accomplished by controlling the vane of the upper pump. To control the vane motion, they provided a hermetic pressure chamber at the back of the upper vane by controlling pressure of that chamber. The pressure of the vane chamber was selectively applied by 3 way valve. When the suction pressure (Ps) was applied into the vane chamber, the vane was separated from rolling piston by pressure difference. Consequently, the division between compression chamber and suction chamber in upper pump disappeared and the upper pump idled. In this case, the displacement of lower pump determined the compression capacity of DRC. On the other hand, if the discharge pressure (Pd) was applied to the vane chamber, DRC worked like a conventional twin rotary compressor in which both upper and lower pump worked together. It was designed to achieve minimum 30% of total capacity, by altering operational timing of two pumps having different compressing volume. The intermediate capacities can be obtained by PWM control of each mode. For example, DRC which has total cycle time 20 seconds with 70% capacity was achieved by operating DRC for 11.4 seconds in power mode and 8.6 seconds in save mode. As a whole 30%~100% continuous capacity modulations is possible by manipulating the power and save modes timing.

Koji at al. [4] explained the condition of their country that due to high heat insulation, air-conditioners were usually operated at part load conditions except at the start-up. So, considering the target point of energy saving, the efficiency must be considered at reduced operating loads. But conventional compressor failed to offer these advantages as well as caused temperature fluctuation in the vicinity. In order to accomplish the aforementioned objectives, authors had developed dual stage compressor having two compression stages. The compression stages were controlled by the vane i.e. when you need large capacity then both vanes were pushed and on the either side, only single cylinder raced idle with second cylinder in operation. With this dual compressor, the air-conditioner can provide cooling and heating by minimum 45 watt.

Shih-Cheng at al. [5] developed a multi-type air-conditioner without using inverter technology and performed its testing using five indoor but with single outdoor unit. They proposed that the systems incorporated with inverters caused malfunction of other electronic equipments present in the vicinity because of generation of electronic noise in the circuit and commenced deformation of the electric sine wave. The proposed system illustrated a wider range of capacity modulation according to thermal load i.e. from 17 to 100% as compared to inverter control air-conditioner which modulated between 48 to 104% thus provided energy saving potential in the same manner up to 25%.

2.2 CAPACITY MODULATION USING INVERTER TECHNOLOGY:

Kazu at al. [6] had developed dual stage compressor with Inverter and discussed the use of Air-conditioner according to season that Air-conditioners are operated mostly in spring and autumn in minimal capacity to remove accumulated indoor heat from home electric appliances, lighting equipments and human bodies. This showed that compressor is used mostly in the small capacity region except when air-conditioner is started. Energy efficiency of the conventional compressor is significantly lower in small capacity region. The novel design consisted of variable cylinder system which enabled one of the two compression chambers of two cylinder rotary compressor with one stayed on idle mode while other kept running. Two cylinder operation using two compression chambers was performed when large capacity was needed, such as at the start of the air conditioner while the single cylinder operation was performed in small capacity region such as when the room temperature was near the set temperature. The two cylinder rotary compressor had two compression chambers, and their compression works were performed simultaneously but independently by one shaft connected to the motor rotor. Considering these two independent compression chambers as two small compressors, authors developed new variable capacity mechanism (referred as “variable cylinder system”) that operated two compression chambers in the large capacity region (2-cylinder operation-capacity: 100%) and one compression chamber in the small capacity region (1-cylinder operation-capacity: 50%).

M Igata at al. [7] had developed an IPM Motor (Interior Permanent Magnet) used in compressors of air conditioners which is commercially produced. This IPM Motor fed by an inverter was more efficient than any other type of motor at any rotational speed. The reasons for being high efficient was rare-earth permanent magnets placed inside the rotor and other was appropriate control of motor current vector by inverter. The compressor with these IPM motor had significantly improved performance as compared with the compressor using conventional induction motors. Scientists in this research claimed that by adopting this technology, they had achieved top level energy saving performance since the IPM motor improved efficiency especially at low to middle rotational speeds. They also discussed two major problems faced by the society in compressor system for air conditioner. One was to handle alternative refrigerants corresponding to total abolition of refrigerants using HCFCs in order to prevent ozone layer depletion while the other was to achieve further energy saving to cope with the global warming.

J. M Choi at al. [8] described that capacity modulation was achieved using an inverter driven compressor equipped with two indoor units. Compressor speed adjustment and proper mass flow distribution control using EEVs were essential parts in designing multi-air conditioning system. It was observed that, as the compressor speed increased, the cooling capacities for both indoor units were increased and the capacity balance between each indoor unit occurred at a higher EEV opening. Therefore, it was concluded that it is necessary to control both EEV opening and compressor speed to modulate the capacities of each indoor unit properly.

Qiu Tu at al. [9] presented in this paper, an important operational control characteristic of VRF (Variable Refrigerant Flow) Air-Conditioner in a way that an outdoor unit containing multi standard compressors (say one or two) with one driven by inverter supported various indoor units and the capacity (i.e. the refrigerant flow) was modulated according to the need based on thermal load. Experimentally, it was investigated that if control mode and process were not reasonable, then it lead to frequent ON/OFF phenomenon of the standard compressor, unstable running and bad comfort of the system. Therefore, compressor switch control mode was established.

2.3 TRIBULATIONS WITH INVERTER TECHNOLOGY:

Shuangquan Shao at al. [10] simulated performance of compressor at different frequencies. They identified that lubrication of compressor motor by lube oil accumulated in the oil sump of the compressor become worsen when compressor operated at low frequencies. Hence it resulted in amplification of frictional losses which lead towards undesirable performance operation of inverter compressor. So, they did not agree that the operational performance was consistent with the frequency.

Pongsakorn at al. [11] described various operational characteristics of inverter driven air-conditioners and the relation between operational frequency and COP values. They implied that the attainable frequency range of modulation of conventional inverter driven Air-conditioner was from 0 to 120Hz but the appropriate frequency range was observed between 40 to 80 Hz which avoided complications like compressor lubrication by splashing which encountered below 30 Hz. Also, above 80 Hz, the COP of the compressor declined which made its operation ineffective from energy conservation stand point.

Gomes at al. [12] discussed various operational behaviors of inverter driven air-conditioner. They expressed that, when inverter AC was started, it operated on the maximum frequency in order to attain the set point as shortly as possible. The equivalent practice was observed again when high temperature variations taken place in the vicinity. This max frequency operation mode was referred as In-stability mode in which the devices typically operate on rated power or even elevated power figures. So, in aforementioned circumstances, it was very difficult to attain break-even point of Inverter driven systems in contrast with traditional Air-conditioners.

Qureshi at al. [13] summarized the literature work in this paper which was carried out on variable speed capacity in refrigeration system. Authors had described that inverter technology had so far restricted to small air-conditioning units and their application in medium range appeared insufficient due to various problems like insufficient development or integration of components, insufficient reliability, high initial cost and failure to produce desired results for energy savings. Green house emissions could be reduced by efficient air-conditioning equipment which was one of the major causes of its increase. Two modes for the capacity control method commonly available were multi-compressor control and cylinder unloading. The basic idea behind variable capacity was that the capacity should perceive the load employed on the equipment under various operating circumstances. Some of the advantages of variable speed control over fixed speed system included temperature accuracy, gradual soft start capability of the system and less noise operation at part load conditions.

Dr. Satya [14] tested various Inverter Air-Conditioners under frequency locked and unlocked modes and observed efficiency degradation in unlocked mode.

2.4 MISCELLANEOUS:

Meissner at al. [15] gone through several tests and found that the energy saving could not be achieved in realistic conditions every time by higher energy efficiency rating air-conditioner.

Seung at al. [16] developed two stage rotary compressor to attain the capacity modulation and hence energy saving characteristics.

Hui Pu at al. [17] had investigated the heat transfer co-efficient of aluminum-finned-copper-tube and copper finned-copper-tube evaporator on intermittent operation of air-conditioner and found that heat transfer rate decreased slowly with increase in number of intermittent operation and under the long term influence; the heat transfer of copper finned-copper-tube was higher to aluminum-finned-copper-tube evaporator.

Xiangguo at al. [18] suggested lowering the cooling capacity on achieving set point instead of intermittent operation which lead towards higher humidity as well as discomfort due to an elevated level of skin humidity and inadequate cooling of the mucous membranes in the upper respiratory tract by inhalation of humid or warm air, therefore recommended that the supply air fan speed must be adjusted accordingly. They also discussed the effects of high energy consumption due to continuous operation of compressor and further described that by reducing the system need which was initially operated at full speed and drawn maximum energy consumption, the cooling remained available at all the time, thus provided better cooling comfort to the user. They developed a new control algorithm (i.e. H-L control) for direct expansion air-conditioning versus traditional on-off mechanism and found better control in terms of indoor humidity and better energy efficiency. H-L control enabled compressor and fan to operate at high speeds when temperatures were not justified or else at low speeds.

Abdel-Hamid at al. [19] compared their indigenous fuzzy logic control system with the conventional PID control. It was revealed that the fuzzy logic control satisfied temperature needs more precisely regardless of thermal load type.

Sohair at al. [20] developed the Building Management System for energy efficient operation of air-conditioner with user comfort at the same time. The identified system was able to manage the temperature and humidity ratio of the vicinity. The Building Management System was the proposed system which control HVAC parameters according to the pre-defined category of very important rooms, important rooms and normally operated rooms.

Hajidavalloo at al. [21] proposed increase in co-efficient of performance of condenser by using evaporative cooling. The design was consisting of an evaporative cooler, which was built and coupled to the existing air-cooled condenser of a split-air-conditioner in order to measure its performance under various ambient air temperatures. They simulated on 1.5 Ton Mitsubishi split Air Conditioner and installed a suitable frame at the side where cooling air entered to cool the condenser. A cellulose pad approx. 5 cm thickness was installed on the frame and a water connection was provided with a suitable flow rate using a small pump. It was observed that, in case of very high ambient temperatures, the fluid temperature and consequently the discharge pressure from the compressor was increased to the limit that it made the compressor shut down by pressure safety device provided with the system. Thus, by supplying cool air to the condenser, the efficiency of the condenser increased making more liquid refrigerant available for evaporator which further improved as the ambient temperature was reduced. As a consequence 20% reduction in power consumption was noted and the total system performance was improved by 50%.

S.S. Hu at al. [22] discussed Taiwan Government plan for reduction in emissions from power plants by reducing the consumption of Electrical energy being consumed in domestic air-conditioners by increasing their COP. In order to accomplish the goal, they proposed a water-cooled technology which utilized water to cool down condenser temperature. The designed setup included a pump which circulated cooling water from the condenser to the cooling tower in the form of spray from the top of the cellulose pad (which was used as filling material) and a cross-flow fan pushed fresh air against the cellulosic pad. In this way, the heat generated in the condenser was brought to the cooling tower through circulating water where it was exhaled to the atmosphere. As performance improvement, the system COP increased from 2.96 to 3.45.

Buzelin at al. [23] proposed a system to conserve energy in industrial refrigeration system. They noticed little temperature variations in the cold chamber using proposed closed loop power law controlled system as compared to conventional ON/OFF system.

Shuxue at al. [24] developed a prototype and discussed ratio of high pressure cylinder to the low pressure cylinder and the two stage compression refrigeration/heat pump demonstrated favorable performance under both cooling and heating conditions.

CHAPTER #3

EXPERIMENTATION

3.1 RESEARCH METHODOLOGY:

In this experiment, we have used two sets of condensing unit each having individual compressor of equal capacity. The refrigerant coil of each unit was drowning to Chilled Water Storage Tank of 60 liters capacity and the chilled water temperature was set at 10℃. This chilled water was passing through the Fan Coil (Indoor) Unit using two set of circulating pumps in series and the mass (i.e. chilled water) flow rate was set according to design specifications of Fan Coil Unit which was further regulated for different testing and this FCU was given a set point to maintain 20℃ inside room. The chilled water was running in closed circuit to avoid contamination from environment that could trim down heat transfer rate due to scaling inside copper tubes. The mode of operation was equipped using thermostat that governed compressors on predefined set point of 10℃ as stated above.

3.2 CONSTRUCTION OF DUAL COMPRESSOR:

3.2.1 COMPRESSOR:

The design chiefly consists of two identical compressors whose details are as under:

Compressor manufacturer: Danfoss

LRA Rating: 10

Voltage 220 V

Hz: 50

3.2.2 CHILLED WATER CIRCULATING PUMPS:

Two Pumps were used for circulation of chilled water through indoor (Fan Coil) Unit. Both pumps were installed in series and were also utilized to vary chilled water flow rate throughout the experiment. Fan Coil unit was installed at an approximate height of 6 ft above ground level. Pump # 1 was submersed in water having 3000 L/Hr capacity with max head as 2 to 3 meters while pump # 2 was placed at a height of FCU having 960 L/Hr capacity which was feeding chilled water to FCU. As a whole, 35~65 GPH chilled water flow rate was achieved.

Fig3.1: Chilled water circulating pump-1 with 3000 L/Hr adjustable flow rate

Fig 3.2: Chilled water circulating pump-2 with 960 L/Hr flow rate

3.2.3 CONDENSER:

Two Condensers were used in experiment having half ton capacity, each made in copper tubes supported with Aluminum Fins. The rpm of condenser fan motor was 1300 having 0.35 Amps current.

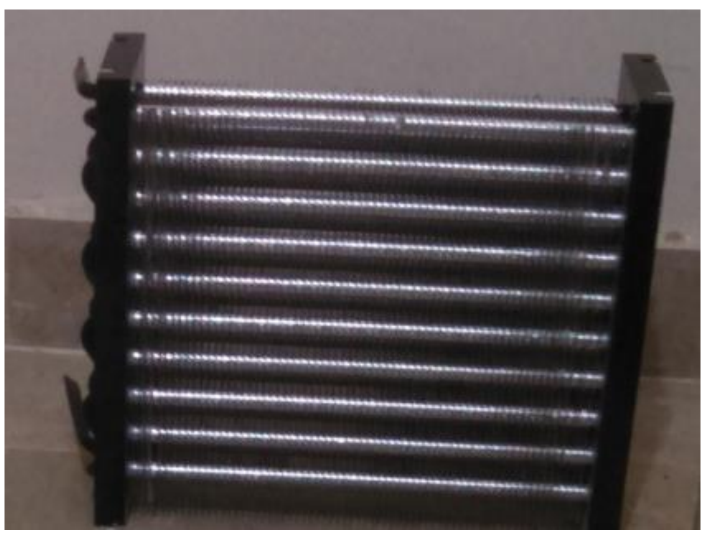


Fig 3.3: Condenser

3.2.4 REFRIGERANT (CHILLED) HELICAL COPPER COIL:

Fig 3.4: Copper Tube used for making Refrigerant (Chilled) Helical Copper coil

Refrigerant coil was made by copper tube with outer diameter as 8 mm and inner diameter as 5.5 mm having wall thickness of 1.25 mm. The size of each coil was 35 feet and the numbers of circular rounds were 19. The height of the coil was more or less 14 inches.

Fig 3.5: Refrigerant (Chilled) Helical Copper Coil

Table 3.1: Surface area calculation of refrigerant coil

Internal Diameter of the copper coil	5.5	Mm	0.217	Inch
Outer Diameter of the copper coil	7.937	Mm	0.312	Inch
Radius with respect to External Diameter	3.969	Mm	0.156	inch
Total Length of Copper Coil	10668	Mm	420	inch
Length is actually the height of coil	10668	Mm	420	inch
Formula for Surface Area:	2 . pi . r . h			
Surface Area for One Coil	412		square.inch	

Hence, the total surface area responsible for heat transfer accumulated was 412 square inch for one coil. Hence for two coils, it turned out to be 824 square inch.

3.2.5 CHILLED WATER STORAGE TANK:

Chilled water storage tank was made by using galvanized iron sheet with thickness of 18 gauges. From decay protection, the tank was coated with bitumen from inside as well as outside. 1.5 inch thick thermo pole sheets were used for the insulation of Tank surfaces facing ambient environment and to prevent energy losses.

Fig 3.6: Chilled water storage tank

3.2.6 IRON FRAME:

An iron frame having 1.25 x 1.25 inches wide angle iron was designed specifically for the installation of about all components except indoor unit. Various necessary holes having diameter of approx 8 mm were generated on all sides on frames for the installation of covering sheets and electrical panel etc.

Fig 3.7: Angle iron frame raw form

Fig 3.8: Angle iron frame after finishing

3.2.7 ELECTRICAL PANEL:

An electrical panel considering internal size of frame was exclusively designed for the installation of Electrical components. It included following items:

1. Two circuit breakers were used, each having 6 amps current. This circuit breaker also served as safety device which could power off system in case of any short circuiting or malfunctioning.
2. A circuit breaker having 10 amps capacity was installed on the main line which was also serving as the secondary safety gadget.
3. Two Contactors were used each having 9 amps current capacity. Each contactor chiefly bearing the load of compressor, condenser fan and the cooling fan installed as optional device for the compressor. The first contactor also bears the load of pumps.
4. Two Multi Timers were used which can be adjusted in Seconds/Minutes/Hours in a wide range.
5. A temperature controller was used having temperature display in Celsius as well as controls stage # 2. Set points are defined in Temperature controller which is responsible for cut-off and cut-in of stage # 2.

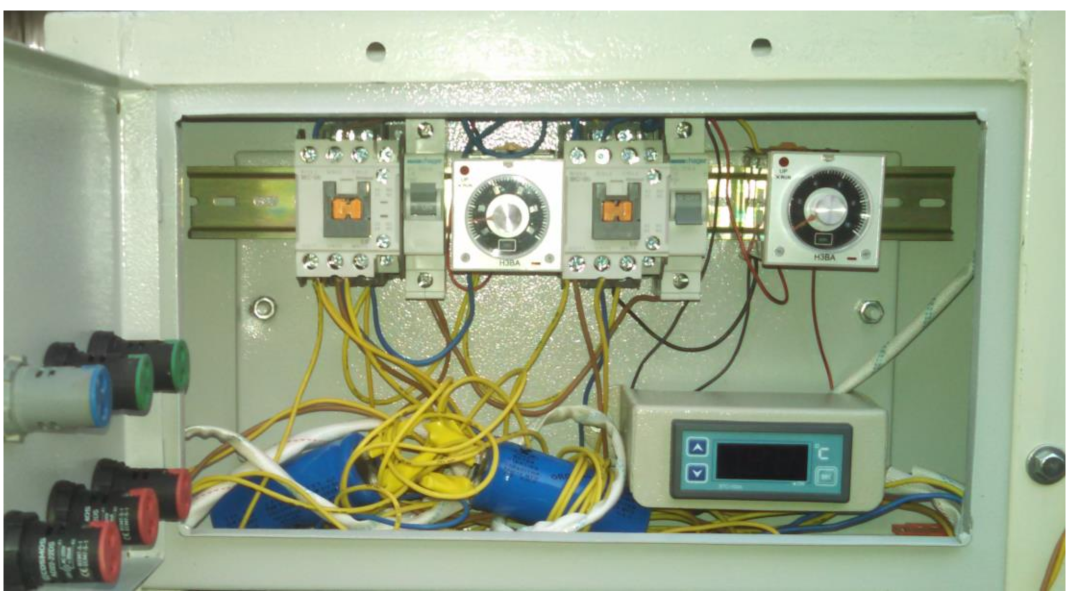


Fig 3.9: Electrical components panel

3.2.8 REFRIGERANT GAS:

The refrigerant used in this system is R-134a by Honeywell USA. This gas is known as less damage to environment as compared to conventional system refrigerant R-22 which is found as harmful for ozone layer and therefore going to BAN its production in entire world in 2020.

3.2.9 ENERGY METER:

Energy Meter is a disc type locally made in casted body which is kept same throughout the experiments.

3.2.10 DUAL COMPRESSOR AIR-CONDITIONER:

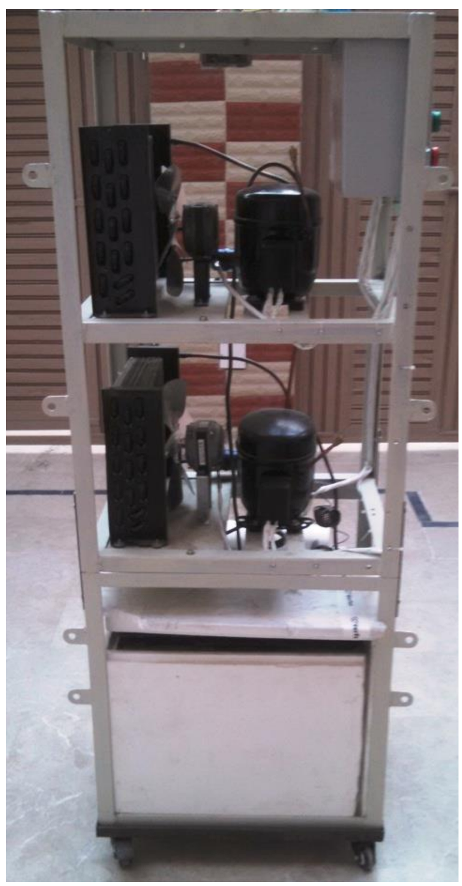


Fig 3.10: Internal View of Dual Compressor

3.2.11 DESIGN:

Fig 3.11: A view of dual compressor air-conditioner

CHAPTER # 4

DATA COLLECTION AND ANALYSIS

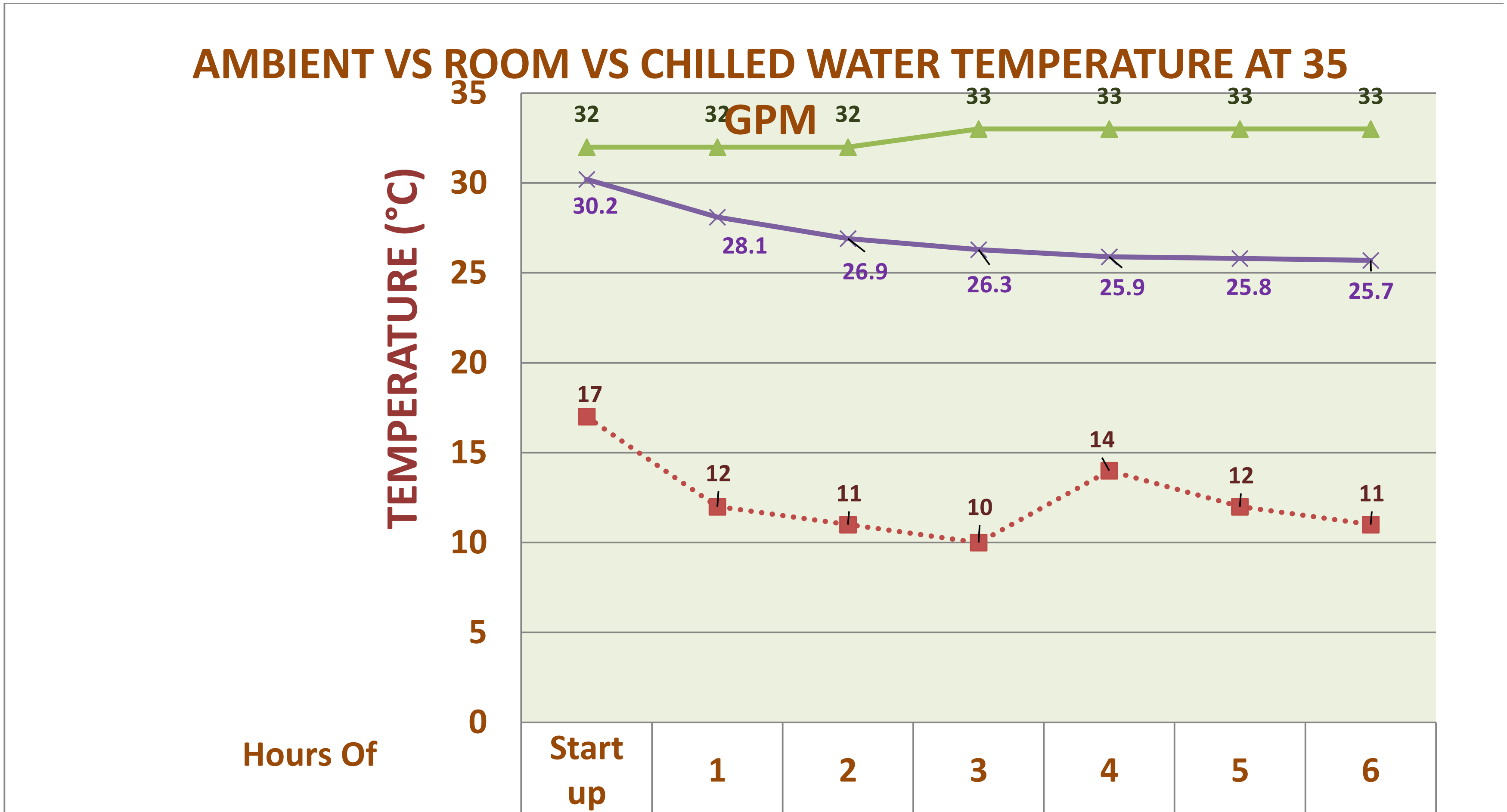
In experimentation, we have conducted four tests in which flow rate of the chilled water was varied. In the end, all the tests were compared on temperatures and power consumption basis.

4.1 OPERATING DUAL COMPRESSOR AT 35 GPH FLOW RATE:

S. No.	Time Interval	Chilled Water Temperature	Chilled Water Flow Rate	Ambient Temperature	Room Temperature	Temp. Diff. per Hour	Percent per Hour	No. of Compressors	Accumu-lated Meter Reading	Net Power Consumption
	Hour	°C	Gallon/Hour	°C	°C	°C	%	#	Counter	KWH
1	**Start-up**	**17**	**35**	**32**	**30.2**	**X**	-	**2**	**40.00**	-
2	1	12	35	32	28.1	2.1	47%	2	41.00	1.00
3	1	11	35	32	26.9	1.2	27%	2	42.00	1.00
4	1	10	35	33	26.3	0.6	13%	1	42.90	0.90
5	1	14	35	33	25.9	0.4	9%	1	43.50	0.60
6	1	12	35	33	25.8	0.1	2%	2	44.35	0.85
7	1	11	35	33	25.7	0.1	2%	2	45.30	0.95
Temperature difference after 6 hours						**4.5**				**5.30**

Chilled Water Minimum Set-point/Cut-off Temperature	10 °C
Chilled Water Maximum Set-point/Cut-in Temperature	15 °C

Table 4.1: Testing At 35 GPH Chilled Water Flow Rate



Hours Of	Start up	1	2	3	4	5	6
Chilled Water Temperature °C	17	12	11	10	14	12	11
Ambient Temperature °C	32	32	32	33	33	33	33
Room Temperature °C	30.2	28.1	26.9	26.3	25.9	25.8	25.7

Fig 4.1 Temperatures Trend with 35 GPH Flow Rates

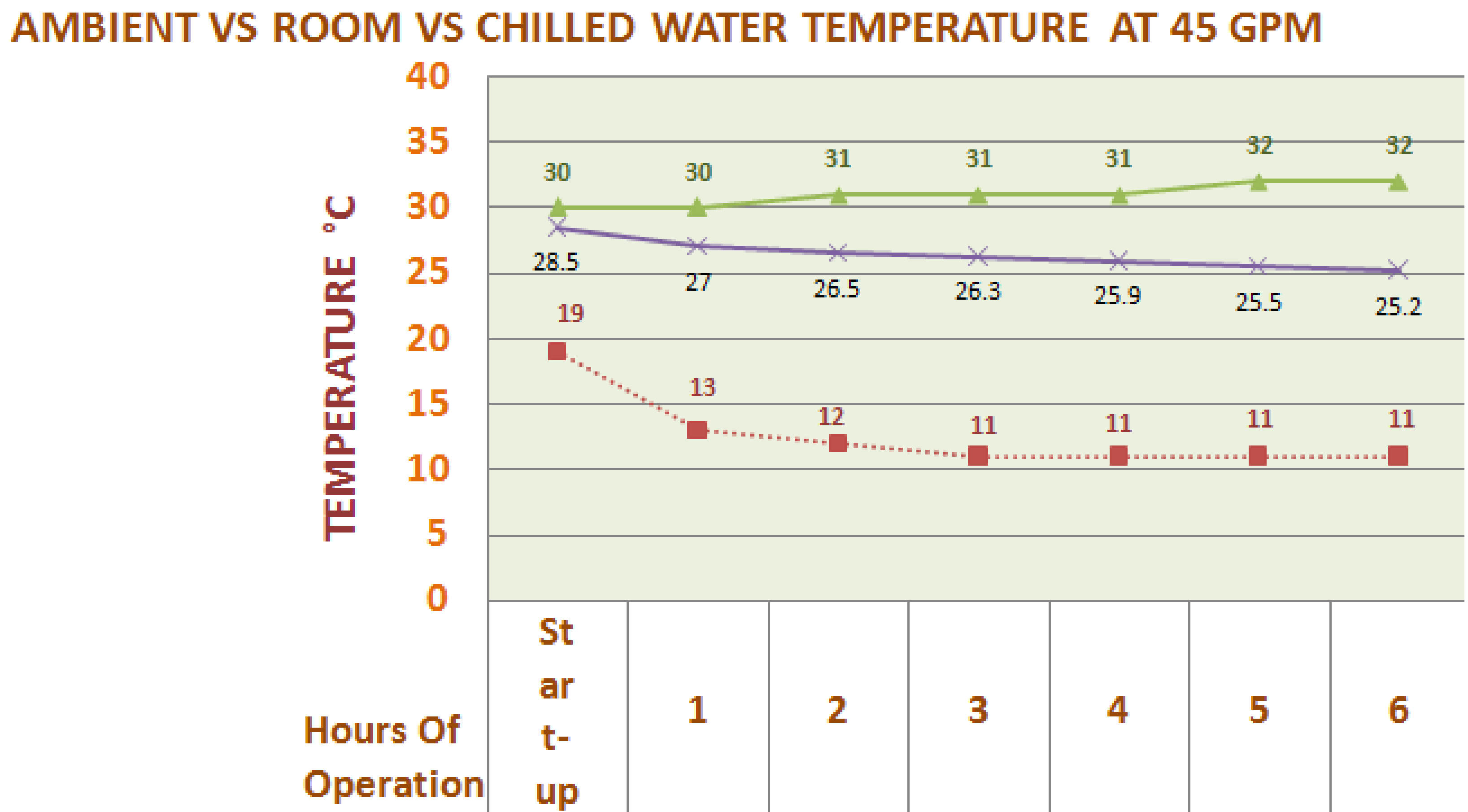
With 35 GPH chilled water flow rate, it was observed in operation of Dual Compressor that both stages of compressors were in operation during first and second hour. But during third hour of operation, the chilled water achieved set point temperature of 10°C which cut-off second stage of compressor while the first stage was in operation along with other accessories like chilled water circulation pump and Fan Coil Unit as it is. During fourth hour of operation, the chilled water temperature rose to 14°C while the cut-in point of second stage compressor was set at 15°C. So, the second stage compressor remained stand-by during fourth hour of operation. Therefore, during third and fourth hours, the power consumed is 10% and 40% lower respectively as compared to first two hours of operation. During fifth hour, the Chilled Water Temperature rose to 15°C and the second stage of compressor was again come into operation which remained operational during sixth hour. Hence, from the data, it is reflected that the total power consumption after six hours of operation is **5.3 KWH**.

Another study is the difference in room temperature achieved after every hour which is 2.1°C i.e. 47% in first hour, 1.2°C i.e. 27% in second hour which further reduced after every hour and summarized a total of 4.5°C after six hours of operation. The operation of the second stage compressor was not found steady that it become OFF within some time during third hour of operation and become ON at some point in fifth hour. The temperature of the chilled water remained vary throughout this test which was observed a maximum of 15°C and the lowest as 10°C at cut-off point, however, the flow rate of chilled water was kept constant at 35 GPH throughout this test.

4.2 OPERATING DUAL COMPRESSOR AT 45 GPH FLOW RATE:

S. No.	Time Interval	Chilled Water Temperature	Chilled Water Flow Rate	Ambient Temperature	Room Temperature	Temp. Diff. per Hour	Percent per Hour	No. of Compressors	Accumu-lated Meter Reading	Net Power Consumption
	Hour	°C	Gallon/Hour	°C	°C	°C	%	#	Counter	KWH
1	**Start-up**	**19**	**45**	**30**	**28.5**	**X**	-	**2**	**30.50**	-
2	1	13	45	30	27	1.5	**45%**	2	31.50	1.00
3	1	12	45	31	26.5	0.5	**15%**	2	32.50	1.00
4	1	11	45	31	26.3	0.2	**6%**	2	33.50	1.00
5	1	11	45	31	25.9	0.4	**12%**	2	34.50	1.00
6	1	11	45	32	25.5	0.4	**12%**	2	35.5	1.00
7	1	11	45	32	25.2	0.3	**9%**	2	36.5	1.00
Temperature difference after 6 hours						**3.3**				**6**
Chilled Water Minimum Set-point/Cut-off Temperature						10 °C				
Chilled Water Maximum Set-point/Cut-in Temperature						15 °C				

Table 4.2: Testing At 45 GPH Chilled Water Flow Rate



	Start-up	1	2	3	4	5	6
Chilled Water Temperature °C	19	13	12	11	11	11	11
Ambient Temperature °C	30	30	31	31	31	32	32
Room Temperature °C	28.5	27	26.5	26.3	25.9	25.5	25.2

Fig 4.2 Temperatures Trend with 45 GPH Flow Rates

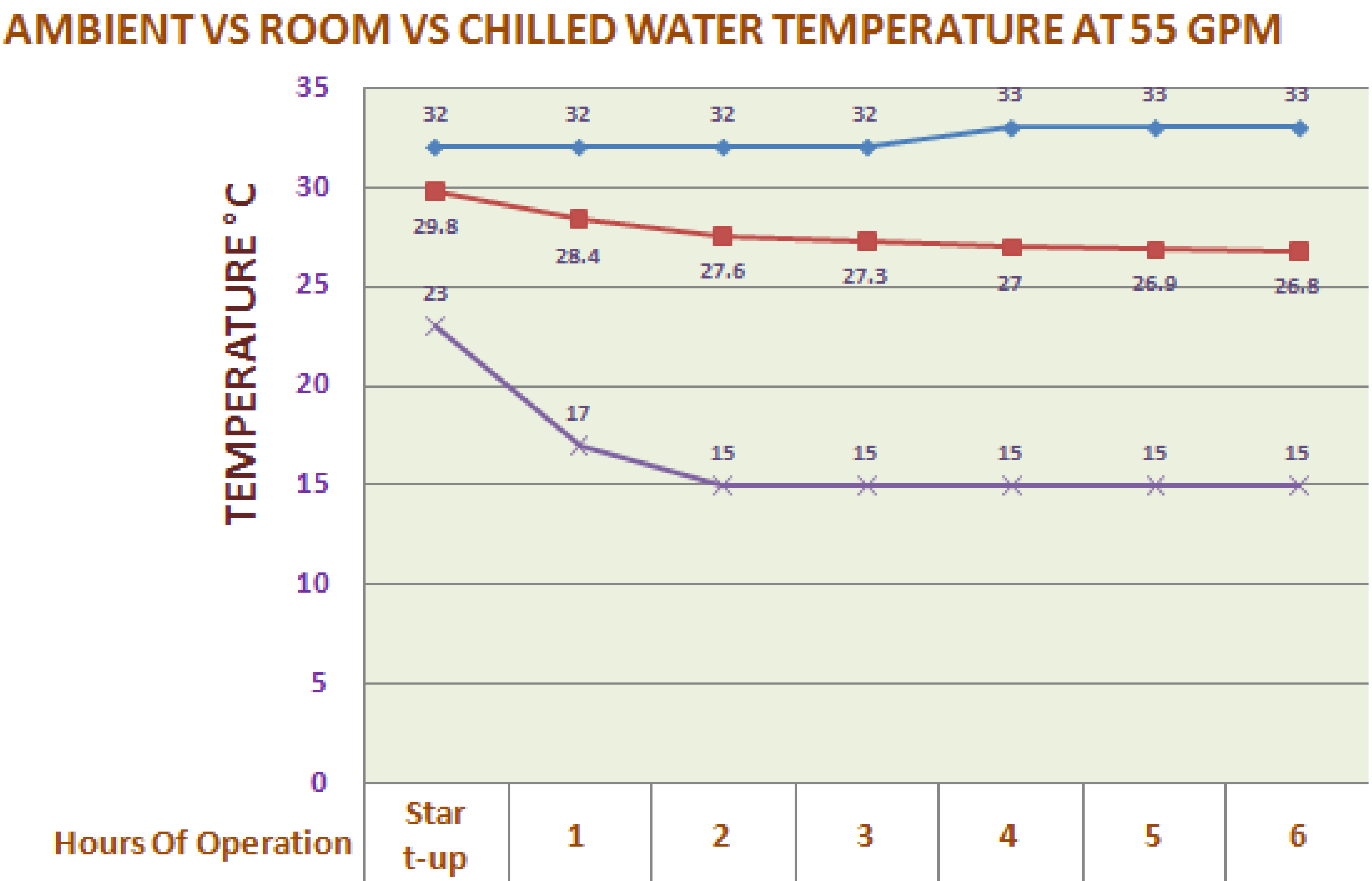
With 45 GPH chilled water flow rate, it was observed that both stages of compressors were in operation during six testing hours and never cut-off, since chilled water did not achieve set point temperature of 10°C. Therefore, both stages were remained in operation along with other accessories like chilled water Circulation Pump and Fan Coil Unit throughout this test. The cut-in point was kept same i.e. 15°C. Hence, from the data, it is reflected that the total power consumption after six hours of operation is 6 KWH.

Another observation was difference in temperature noted after every hour which was 1.5°C i.e. 45% in first hour, 0.5°C i.e. 15% in second hour which further reduced continuously after every hour and summarized a total of 3.3°C after six hours of operation. The temperature of the chilled water observed decreasing till third hour and got equilibrium after three testing hours. It was observed a maximum of 15°C and the lowest as 11°C; however, the flow rate of chilled water was kept constant at 45 GPH throughout this test. The graph reflects maximum temperature difference of chilled water in first hour, means that maximum room temperature difference achieved in first hour while a continuous decreasing trend was observed in subsequent hours.

4.3 OPERATING DUAL COMPRESSOR AT 55 GPH FLOW RATE:

S. No.	Time Interval	Chilled Water Temperature	Chilled Water Flow Rate	Ambient Temperature	Room Temperature	Temp. Diff. per Hour	Percent per Hour	No. of Compressors	Accumu-lated Meter Reading	Net Power Consumptio
	Hour	°C	Gallon/Hour	°C	°C	°C	%	#	Counter	KWH
1	**Start-up**	**23**	**55**	**32**	**29.8**	**X**	-	**2**	**62.50**	-
2	1	17	55	32	28.4	1.4	47%	2	**63.50**	1.00
3	1	15	55	32	27.6	0.8	27%	2	**64.50**	1.00
4	1	15	55	32	27.3	0.3	10%	2	**65.50**	1.00
5	1	15	55	33	27	0.3	10%	2	**66.50**	1.00
6	1	15	55	33	26.9	0.1	3%	2	**67.50**	1.00
7	1	15	55	33	26.8	0.1	3%	2	**68.50**	1.00
Temperature difference after 6 hours						3				6.0
Chilled Water Minimum Set-point/Cut-off Temperature						10 °C				
Chilled Water Maximum Set-point/Cut-in Temperature						15 °C				

Table 4.3: Testing At 55 GPH Chilled Water Flow Rate



Hours Of Operation	Star t-up	1	2	3	4	5	6
Chilled Water Temperature °C	23	17	15	15	15	15	15
Ambient Temperature °C	32	32	32	32	33	33	33
Room Temperature °C	29.8	28.4	27.6	27.3	27	26.9	26.8

Fig 4.3 Temperatures Trend with 55 GPH Flow Rates

With 55 GPH chilled water flow rate, it was observed that both stages of compressors were in operation during six testing hours and never cut-off because the chilled water did not achieve set point temperature of 10°C. Therefore, both stages were remained in operation along with other accessories like chilled water Circulation Pump and Fan Coil Unit throughout this test. The cut-in point was kept same i.e. 15°C. Hence, from the data, it is reflected that the total power consumption after six hours of operation is 6 KWH.

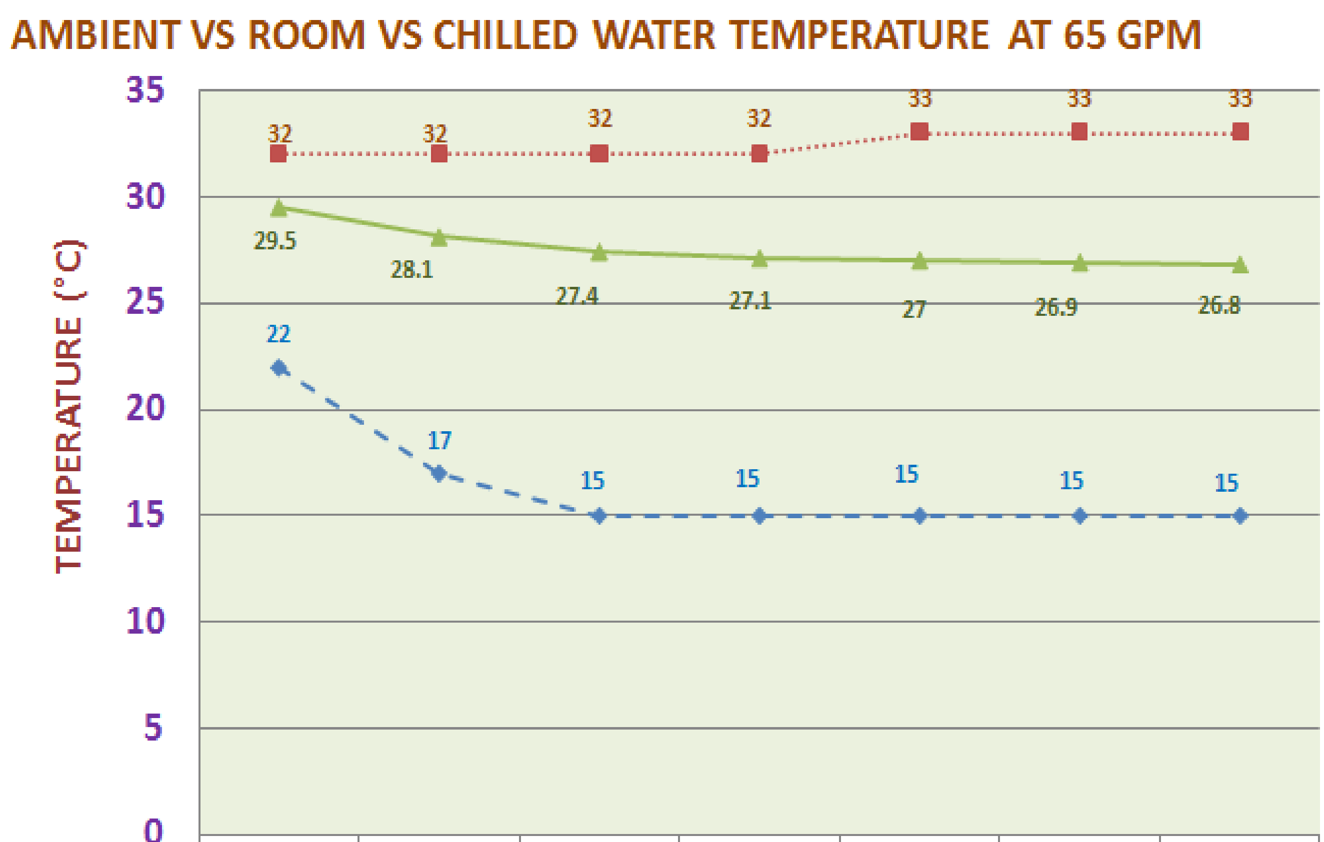
It was reflected from the data that chilled water attained a temperature of 15℃ which retained subsequent hours throughout the test. The temperature difference between chilled water and the room temperature was observed 12℃ at continuous trend. The maximum temperature difference was achieved in first hour i.e. 1.4℃ while a continuous decreasing trend is observed in subsequent hours which summarized a total of 3℃.

4.4 OPERATING DUAL COMPRESSOR AT 65 GPH FLOW RATE:

S. No.	Time Interval	Chilled Water Temperature	Chilled Water Flow Rate	Ambient Temperature	Room Temperature	Temp. Diff. per Hour	Percent per Hour	No. of Compressors	Accumu-lated Meter Reading	Net Power Consumption
	Hour	°C	Gallon/Hour	°C	°C	°C	%	#	Counter	KWH
1	**Start-up**	**22**	**65**	**32**	**29.5**	**X**	-	**2**	**54.95**	-
2	1	17	65	32	28.1	1.4	52%	2	**55.95**	1.00
3	1	15	65	32	27.4	0.7	26%	2	**56.95**	1.00
4	1	15	65	32	27.1	0.3	11%	2	**57.95**	1.00
5	1	15	65	33	27	0.1	4%	2	**58.95**	1.00
6	1	15	65	33	26.9	0.1	4%	2	**59.95**	1.00
7	1	15	65	33	26.8	0.1	4%	2	**60.95**	1.00
Temperature difference after 6 hours						2.7				6.0

Chilled Water Minimum Set-point/Cut-off Temperature	10 °C
Chilled Water Maximum Set-point/Cut-in Temperature	15 °C

Table 4.4: Testing At 65 GPH Chilled Water Flow Rate



Hours Of Operation	Start-up	1	2	3	4	5	6
Chilled Water Temp.	22	17	15	15	15	15	15
Ambient Temp.	32	32	32	32	33	33	33
Room Temp.	29.5	28.1	27.4	27.1	27	26.9	26.8

Fig 4.4 Temperatures Trend with 65 GPH Flow Rates

With 65 GPH chilled water flow rate, it was observed that both stages of compressors were in operation during six testing hours and never cut-off because the chilled water did not achieve set point temperature of 10°C. Therefore, both stages were remained in operation along with other accessories like chilled water Circulation Pump and Fan Coil Unit throughout this test. The cut-in point was kept same i.e. 15°C. Hence, from the data, it was reflected that total power consumption after six hours of operation was 6 KWH.

This operation appears to be worst among all four tests because it provides a temperature difference of 2.7°C with the power consumption of 6 KWH. The temperature difference between chilled water and the room temperature was observed 12℃ at continuous trend.

4.5 MATCHING ATTRIBUTES FOR ALL TESTS:

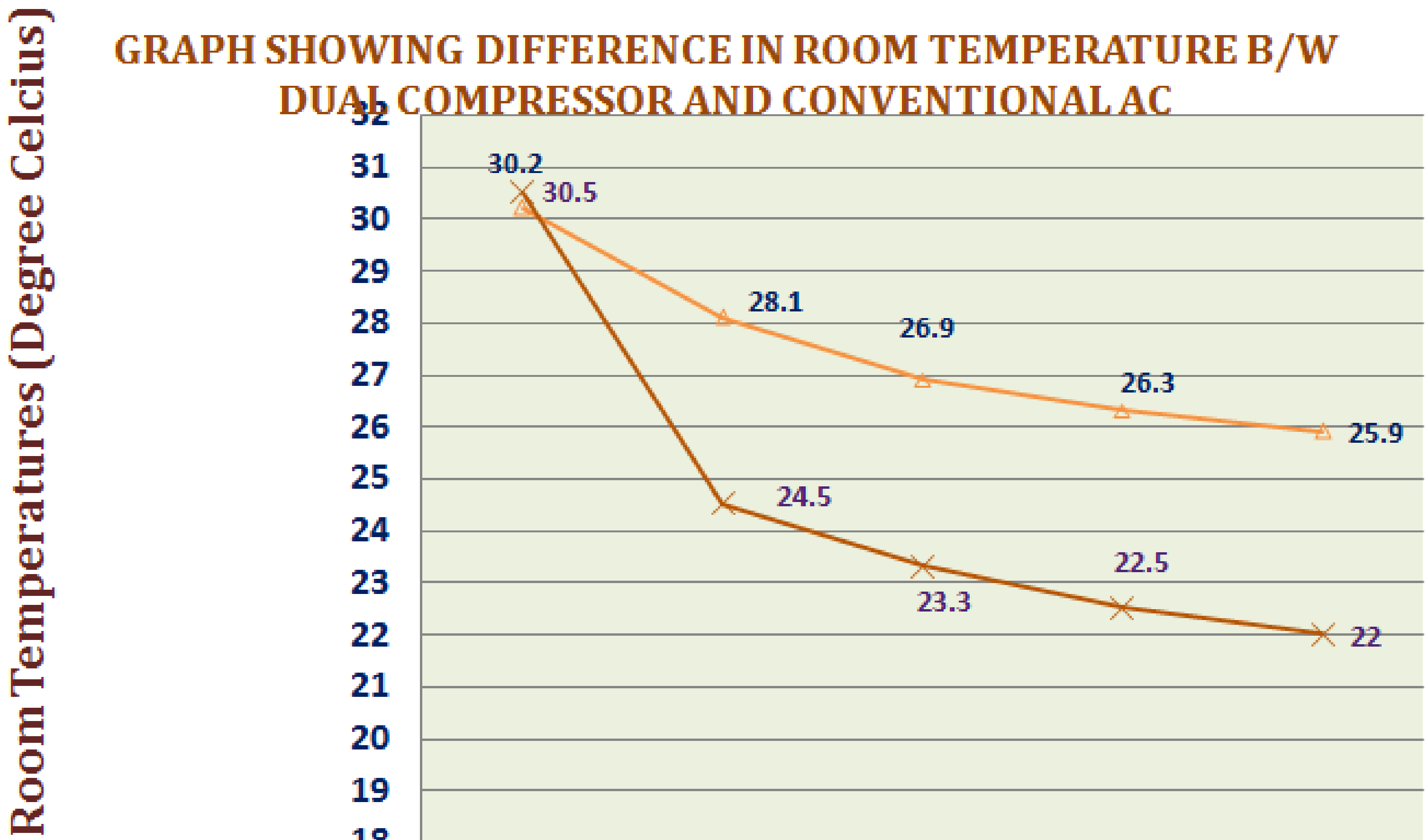
1. Testing room
2. Energy Meter used to measure power consumption
3. Thermal load
4. Indoor/Fan Coil Unit CFM
5. Temperature Sensor and its position used to monitor room temperature
6. Operational Time for testing

4.6 OPERATING DUAL COMPRESSOR VS CONVENTIONAL AIR-CONDITIONER:

S. No.	Time Interval	Chilled Water Temperature	Chilled Water Flow Rate	Ambient Temperature	Room Temperature	Temperature Diff. per Hour	Percent per Hour	No. of Compressors	Accumulated Power	Net Power Consumption
	Hour	°C	Gallon/ Hour	°C	°C	°C	%	#	-	KWH
1	**Start-up**	**17**	**35**	**32**	**30.2**	-	-	**2**	**40.00**	-
2	1	12	35	32	28.1	2.1	49%	2	41.00	1.00
3	1	11	35	32	26.9	1.2	28%	2	42.00	1.00
4	1	10	35	33	26.3	0.6	14%	1	42.90	0.90
5	1	14	35	33	25.9	0.4	9%	1	43.50	0.60
					4.3					**3.50**

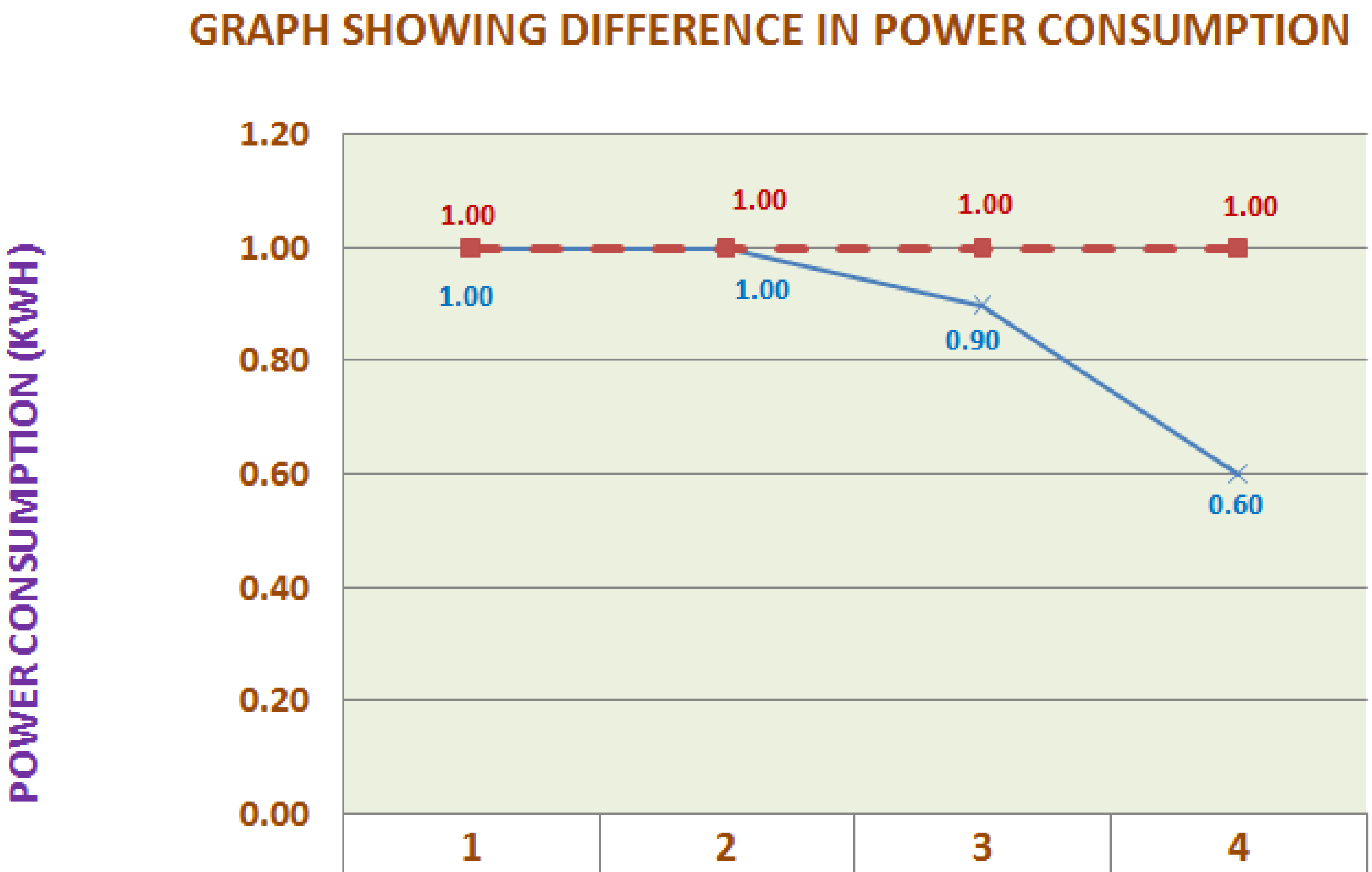
Ambient Temperature	Room Temperature	Temperature Diff. per Hour	Percent per Hour	Accumulated Power Consumption	Net Power Consumption
°C	°C	°C	%	-	KWH
30	**30.5**	-	-	**71.10**	-
29	24.5	6	70.6%	72.10	1.00
29	23.3	1.2	14.1%	73.10	1.00
29	22.5	0.8	9.4%	74.10	1.00
28	22	0.5	5.9%	75.10	1.00
	8.5				**4.00**

Table 4.5: Dual Compressor Vs Conventional AC



Hours Of Operation	Start up	1	2	3	4
Room Temperature (Dual Compressor)	30.2	28.1	26.9	26.3	25.9
Room Temperature (Split AC)	30.5	24.5	23.3	22.5	22

Fig 4.5 Dual Compressor AC versus Conventional AC – Temperatures Trend



	1	2	3	4
Dual Compressor AC	1.00	1.00	0.90	0.60
CONVENTIONAL AC	1.00	1.00	1.00	1.00

Fig 4.6 Electricity Consumption Trend

With 35 GPH chilled water flow rate, it was observed in operation of Dual Compressor that both stages of compressors were in operation during first and second hour. But during third hour of operation, the chilled water achieved set point temperature of 10°C which cut-off second stage of compressor while the first stage was in operation along with other accessories like chilled water Circulation Pump and Fan Coil Unit as it is. During fourth hour of operation, the chilled water temperature increased to 14°C while the cut-in point of second stage compressor was set at 15°C. So, the second stage compressor remained stand-by during fourth hour of operation. Therefore, during third and forth hours, the power consumed is 10% and 40% lower respectively as compared to first two hours of operation. Hence, from the data, it was reflected that the total power consumption after four hours of operation was 3.5 KWH.

Another observation was the difference in temperature achieved after every hour which was 2.1°C in first hour, 1.2°C in second hour which further reduced after every hour and summarized a total of 4.3°C after four hours of operation. The operation of the second stage compressor was not found steady that it become OFF within some time during third hour of operation. The temperature of the chilled water remained vary throughout testing which was observed a maximum of 17°C at start-up and the lowest as 10°C at cut-off point, however, the flow rate of chilled water was kept constant at 35 GPH throughout this test.

Conventional AC was also given a set point temperature of 20°C on thermostat and observed its continuous operation without compressor cut-off for four hour test. It was observed that 6°C and 1.2°C temperature dropped in first and second hours of operation respectively which further reduced in next two hours and summarized a total of 8.5°C after four hours of operation.

4.7 ANALYSIS:

This Analysis is based on the operational comparison of Dual Compressor Air Conditioner (with various flow rates) and Conventional Split Air Conditioner. It was observed that:

(1) At high flow rates of 55 and 65 GPH, the chilled water did not achieve below 15℃ due to reason that chilled water circulation was at the higher rate caused less dwell time with coil and chilled water surface area. While at 45 GPH, the chilled water readily achieved 11℃ temperature in the third hour which was lower than other flow rates and the same was maintained in the remaining three hours of testing and temperature drop inside room was observed as 3.3℃.

(2) As discussed in the introduction regarding second stage compressor cut-off at pre-defined temperature of 10℃, this phenomenon was observed only when the flow rate was set at 35 GPH. At this flow rate, also the lowest chilled water temperature i.e. 10℃ was observed as compared to all other flow rates.

(3) Lowest power consumption as compared to other tests was observed at 35 GPH flow rate which was 3.5 KWH.

(4) Conventional AC was given a set-point of 20℃ inside room, so observed its four hour continuous operation without cut-off with power consumption of 4 KWH.

(5) The designed system successfully attained human comfort temperature of 26℃ with comparative consumption of 12.5% in electrical power.

(6) Since it was chilled water based circuit, therefore it could not be compared apple to apple with direct expansion air-conditioning system. Since, the chilled water based system

is more often used by Industrial sectors for the reason of energy conservation stand point where large scale air-conditioning is required.

CHAPTER # 5

CONCLUSION

In development of dual compressor AC system, it was experimentally evaluated that Dual Compressor AC system has 12.5% less power consumption as compared to Conventional AC even having two separate compressors. Hence, we have developed a high efficient Dual Compressor A/C which is expected to greatly contribute towards development of future AC. The higher will be designed capacity (i.e. unit size) and operational hours in a day, the higher will be conservation of energy. However, following are the observations noted during operational testing on Dual Compressor AC on different chilled water flow rates.

- The second stage compressor cut-off at 10℃ set point only observed at 35 GPH flow rate which is also the lowest chilled water temperature observed throughout entire tests.
- With respect to power consumption, the test with 35 GPH flow leads with lowest power consumption.
- Direct expansion cooling rate with respect to time is faster than chilled water based circuit, so the same is reflected in results.

5.1 RECOMMENDATIONS FOR FUTURE RESEARCH:

Following are the recommendations compiled on the basis of results which can be incorporated in existing design to further decrease power consumption.

1) Use of scroll compressor can lower power consumption due to having better efficiency as compared to reciprocating compressors.

2) Explore different compressors and refrigerant gases which provide relatively fast and extra cooling capacity as compared to Danfoss Compressors for R-134a.
3) The wall thickness of primary refrigerant coil was used as 1.25mm while the wall thickness can be used around 0.60mm. This can improve heat transfer rate and can further reduce chilled water temperature. By decreasing chilled water temperature, the supply air temperature can further reduce which can result in lowering room temperature.
4) Use of larger diameter primary refrigerant coil to increase contact time of chilled water with coil surface. This can result in lowering chilled water temperature and subsequently the supply air temperature inside the room.

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