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A QUASI-STEADY STATE OPERATION MODE OF ALTERNATIVE REFRIGERANTS FOR R-22 IN WATER CHILLERS

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ABSTRACT:

The zeotropic blends comprise of **R-407C** and **R-404A** refrigerants and the pure **R-134a** were selected as a drop-in substitutes for **R-22** in a water chiller under a quasi-steady state operation. When comparing the performance of the refrigerants with those of **R-22**, the results showed that **R-407C**, **R-404A** and **R-134a** reduced the cooling capacity up to (6) %, (6-12) % and (18) % respectively. **R-134a** and **R-407C** exhibited higher power consumption per ton of refrigeration than that experienced with the **R-22** tests by (8-28) % and (7) % respectively. Whereas, **R-404A** showed a lower corresponding value than that of the **R-22** by (9) %. The **R-407C** and **R-404A** showed a significant reduction in the actual (**COP**) by (19-23) % and (22-55) % respectively. The (**COP**) of the **R-134a** showed oscillation when compared with **R-22** ranging from (22) % lower at high water inlet temperature to (29) % higher at low water inlet temperature.

KEYWORDS: Refrigeration, Alternatives, Zeotrope, R-22, Chiller, Quasi-Steady

ألطور التشغيلي الشبه مستقر لبدائل التثليج لفريون R-22 لوحدات إنتاج الماء المثلج

الخلاصة

بالإضافة إلى المائع R-404A و R-407Cتم استخدام موائع التثليج المكونة من خلائط زيوتروبية والمؤلفة من
في وحدة تبريد الماء لظروف تشغيلية شبه مستقرة من دون إجراء R-22 كبدائل لمائع التثليج R-134aالنقي
أي تحوير على منظومة التبريد. عند مقارنة النتائج فقد أبدت جميع البدائل نقصان لسعة التبريد تصل إلى (6) %
. R-134a في حين كانت حوالي (18) % لمائع التثليج R-404A و (6-12)% للمائع R-407Cللمائع
قد أبديا طاقة مستهلكة لكل طن تبريد بقيمة أعلى عند مقارنتها بتلك المستهلكة R-407C و R-134aالمائعان

لفريون R-22 بحوالي (8-28)% و (7)% على التعاقب. في حين قد أبدى المائع R-404A قيمة أقل للطاقة المستهلكة لكل طن تبريد من مثيلتها المحسوبة لفريون R-22 بمقدار (9)%. المائعان البديلان R-407C و R-404A قد بينا انخفاض كبير لقيمة معامل الأداء (COP) المحسوبة عمليا عند مقارنتها بتلك الناتجة من غاز التبريد R-22 بمقدار (19-23)% و (22-55)% على التعاقب. أما بالنسبة لمائع التبريد النقي R-134a فان معامل الأداء (COP) قد بيّن تذبذب عند مقارنته مع تلك المحسوبة لغاز R-22 بمدى يتراوح بين (22)% أقل عند درجة حرارة دخول الماء العالية إلى (29)% أعلى عند درجة حرارة دخول الماء الواطئة.

INTRODUCTION:

The new R-22 alternatives, zeotrope refrigerants, are introducing difficulties in the prediction of the cooling unit characteristics during smooth operation. The most attractive thermal properties of refrigerant that have significant effect on the performance of the vapour compression refrigeration system are the critical temperature and the molar heat capacity as pointed out by McLinden and Radermacher (1987). Meurer et al. (1999) have found that the variation of the (COP) of the system depends on the condensing operating temperature of the refrigerant circulated for the R-22 and R-410A. Lee et al. (2000) concluded that R-410A exhibited very close (COP) to that of the R-22, it falls within (3) % for the same pressure ratio. R-407C showed a lower (COP) than that of the R-22 by (9) %. Staley, et al. (1992) studied experimentally the performance of a domestic refrigerator using R-12 and R-134a as alternative refrigerant. The refrigerator system operated in two modes, steady state and quasi steady state, cycling mode. A comparison was made between the steady state and cyclic case by looking at a point in time where the instantaneous evaporator air inlet temperature are the same. There was a little difference between the steady state case and this "snapshot" of the cycling case, except for the temperature discharge temperature, refrigerant temperature at the exit of evaporator and the evaporator air discharge temperature. However, the cyclic (COP) was (11%) lower than the steady state.

Motta and Domanski (2000) simulated a number of binary and ternary mixtures as alternatives for R-22. Their results showed that fluids with a low critical temperature experience a large degradation of cooling capacity, while rate of compressor power increase is similar for all fluids. Tarrad and Salim (2009) had showed that the zeotrope alternative R-407C refrigerant blend required higher power consumption than that of the R-22 by about

(19) % and its (**COP**) was lower by (17) % for the air conditioner unit. Tarrad and Abbas (2010) studied experimentally the performance of a drop-in technique for alternatives in air conditioning unit operating with **R-22**. They concluded that the results under steady state operation showed that the drop- in technique of **R-22** by **R-407C** and **R-407A** improved cooling capacity up to (12) % and (25) % respectively. The **R-407C** and **R-407A** showed a significant increase in the (**COP**) by (21) % and (27) % respectively for the operating conditions of the tests. Tarrad et al. (2011) presented experimental work for a steady state operation of a drop in substitutes for **R-22** in a water chiller. Their results showed that the cooling capacities were reduced by (8) % and (18) % for **R-407C** and **R-404A** respectively when compared with those obtained from **R-22** refrigerant.

In the present work, the performance of a cycling quasi-steady state mode for **R-22** and three suggested alternatives were studied as a drop-in substitutes in a laboratory scale water chiller at different operating conditions. Noting that, this strategy of alternative study is focused on the behaviour under the **quasi-steady** state operation but not on the **transient dynamic** behaviour prior to the steady state operation of the unit. Further, it is aimed to provide an experimental test tool for the object of selecting a proper alternative to the **R-22** base unit depending on the cooling rate in a drop-in technique. The suggested alternative refrigerants tested in the present work were the zeotropic mixtures **R-407C** and **R-404A** and pure **R-134a** refrigerant.

EXPERIMENTAL ARRANGEMENT

Apparatus Design:

The experimental rig is comprising of an (0.5) ton of refrigeration water chiller circulating **R-22** as a refrigerant, built for the objectives of the present work by Mahmood (2010). The apparatus arrangement with the instrumentation devices is shown schematically in **Fig. 1**.

The refrigerant side instrumentation are fixed at selected ports around the rig at inlet and exit sides of the components. The water path through the chiller is shown schematically in **Fig. 2** for which the temperature and flow rate were measured at the entering and leaving sides. The evaporator shell, water pump and piping system were completely insulated with a sheet of Armaflex having a thickness of (25) mm and thermal conductivity of ($k = 0.036\ W\ m^{-1}K^{-1}$). The evaporator was made of a stainless steel metal tank shown in **Fig. 3.a**, equipped with immersion tube coil, **Fig. 3.b**. The coil was made of a copper (9.52) mm outside diameter tube, (15) m length consisting of (20) turns and having an external surface area of about (0.45) m^2. The evaporator coil was equipped with a bypass port on the water side, vessel side, to allow for a control of the circulated water through the evaporator, **Fig. 3.c**.

The test condenser was of (25.5 × 28 × 65) cm overall dimensions with a maximum working pressure of (30) bar, **Fig. 4**. It is a louvered finned tube air cooled heat exchanger, with a total surface area of (4.074) m^2 divided into (0.228) m^2 and (3.846) m^2 as a bare tube and finned surface area respectively. The tube layout and the arrangement of refrigerant tubes are shown in **Fig. 5**. It comprises of one circuit accommodating three rows, each consists of (10) copper tubes having (9.52) mm outside and (7.93) mm inside diameters equipped with slit aluminium fins.

The type of compressor used in present work was a reciprocating hermetic compressor. It was manufactured by Copland and has a model "RS43CAE-CAA-201". It was charged with polyolester oil, this type of oil is suitable to be integrated with HCFC such as **R22** refrigerant and working with HFC refrigerant (e.g. **R-407C, R-134a**,……..). It is operating with a single phase voltage (110-115) volt and (50-60) Hz and thermally protect. A copper capillary tube have an internal diameter of (1) mm and external diameter of (2) mm with a length of (90) cm was used.

The load imposed on the refrigeration cycle was provided by water heater. It was made of a carbon steel cylindrical vessel, (40) cm diameter and (68) cm height provided with an electrical heater with a power of (2000) Watt. A centrifugal pump type was used to circulate water between the evaporator tank and external load equipped with an electrical motor of (0.5) hp. The water flow rate of the pump was in the range of (5-30) l/min with a head of the pump (5.5-28) m. The arrangement of the experimental rig permits a bypass for a percent of the pump discharge to control the water flow rate passing through the evaporator as shown in **Fig. 3.c**. The present work has accomplished the tests with water flow rate of (250) l/hr. This flow rate was achieved through recirculation a portion of the water discharge from the pump to the reservoir.

INSTRUMENTATION:

The volumetric flow rate of the circulated chilled water is monitored by a Rota meter having the rang of (120 to 1300) l/hr. A number of temperature and pressure gauges were installed at selected ports all around the test rig as shown in **Figs. 1** and **2**. Further, a mercury thermometers were used to measure the dry bulb and wet bulb temperatures. These instrumentation tools were calibrated against a standard instrumentation devices. The errors in the temperature and pressure measurements were (±0.27 %) and (±0.4 %) respectively.

TEST REFRIGERANTS:

In the present work, the method that was recommended by Kim et al. (1997) will be followed for the liquid charging mode. The compressor must be off and the refrigerant cylinder is in inverted position. Add small amounts of liquid refrigerant carefully and watching the weight of the cylinder. All liquid charge process was done when the compressor was at off condition to avoid its damage. The available refrigerants at the Iraqi local markets were used to conduct the experimental work in the present research. The amount of refrigerants charged in this work were (400), (550), (450) and (450) g for the **R-22**, **R-134**a, **R-407C** and **R-404A** respectively. The thermal properties for **R-22** and its alternative **R-134a** were obtained from ASHRAE Handbook (1997), DuPont Freon **R-22** (2004) and DuPont Suva **134a** (2004). For the mixture refrigerants **R-407C** and **R404A**, the tables of thermodynamic properties published by **Solkane 407C** and **Solkane 404A** were used respectively.

TEST PROCEDURE:

On the commencing of the tests the following instructions must be followed.

1- Allow the water pump to circulate the water through out the system and waiting until the temperature of cold water in the evaporator and hot water in the heater approaches to equalization prior to chiller operating. This was done to avoid temperature maldistribution of water throughout the chiller components and achieving homogenous conditions.

The cycling mode was achieved by turning off the electrical heater and turn on the chiller to -2 cool the water from an initial set temperature. Collect the relevant variable measurements for every (10) minutes time step for refrigerant, water and air sides.

PERFORMANCE CALCULATION:

A typical experimental data collected in the present work are shown in **Table 1** when the unit is circulating **R-22**.

Calculation Assumptions:

The energy balance between fluids on both sides of the heat exchangers is used to estimate the required parameters in the evaporator and condenser of the chiller. Accordingly, the process is assumed to be adiabatic with the surrounding and no heat is lost from the heat exchangers. For the compressor and capillary tube, the process is also assumed to be conducted adiabatically and no heat is rejected out of the components.

Cooling Load Capacity:

The Cooling load capacity is estimated for the refrigerant side by the knowledge of the conditions of refrigerant at inlet and exit of the evaporator. Where adiabatic condition with the surrounding is existed, then

$$Q_{evp} = \dot{m}_{ref} \times \left(h_5 - h_4\right) \tag{1}$$

Since the refrigerant mass flow rate was not measured during the present experimental work. The energy balance on both sides of the evaporator with no heat loss yields to

$$\dot{m}_{ref} = \frac{\dot{m}_w \times cp_w \times \left(T_{in} - T_{out}\right)}{\left(h_5 - h_4\right)} \tag{2}$$

COMPRESSOR WORK:

Actual Compressor Work Rate:

The actual compressor power is defined as that work rate used by refrigerant to change it's conditions. Applying the first law of thermodynamics on the compressor boundary and neglecting the potential, kinetic energies and adiabatic condition is available, this yields to:

$$W_{act} = \dot{m}_{ref} \times \left(h_2 - h_1\right) \tag{3}$$

Power Consumption:

The power consumed by the compressor is estimated from the measured values of the current and voltage of the power supply as:

$$W_{cons} = V \times I \times \cos\theta \tag{4}$$

In which the power factor (**cos θ**) was assumed to be (1).

Condenser Load:

The condenser heat rejection is estimated from the knowledge of the conditions of refrigerant at inlet and exit sides of the condenser with no heat loss, then

$$Q_{cond} = \dot{m}_{ref} \times \left(h_2 - h_3\right) \tag{5}$$

Coefficient of Performance:

Actual Coefficient of Performance:

It is defined as the ratio of the cooling capacity to the actual work of compressor.

$$COP_{act} = \frac{Q_{evp}}{W_{act}} \tag{6}$$

Consumed Coefficient of Performance:

It is defined as the ratio of the cooling capacity to the power consumption

$$COP_{cons} = \frac{Q_{evp}}{W_{cons}} \tag{7}$$

Comparison of Results:

In order to compare the performance for the alternatives with those of the base refrigerant **R-22**, the following expression was used for this purpose:

$$\Delta X_{alternative}(\%) = \frac{X_{alternative} - X_{R-22}}{X_{R-22}} \times 100 \tag{8}$$

Where (X) represents any performance parameter for the cycle.

UNCERTAINTY ANALYSIS:

The uncertainty expected in the final results of the experimental work presented here was predicted by the technique based on the **RMS** uncertainty propagation analysis. This analysis showed that the uncertainty percent of the comparable parameters of the chiller unit were small values. These were (±3.9 %), (±4.5 %) and (±10 %) for the chiller cooling load ($\mathbf{Q_{evap}}$), actual coefficient of performance ($\mathbf{COP_{act}}$) and the consumed coefficient of performance ($\mathbf{COP_{cons}}$) respectively.

RESULTS AND DISCUSSION:

Evaporator Cooling Capacity:

Although these tests showed different mass flow rates for the test refrigerants due to the drop in technique used in the present study for comparison of the performance, they still have a cooling load basis for comparison. The cooling rate variation with time is one of the most important characteristics considered in the present work for the cooling unit under the quasi-steady state operation. Further, the reflection of this parameter on the response of the characteristic behaviour of other operation parameters such as coefficient of performance, power consumption and condenser load were considered. Further, the comparison of the performance measures should be explored under smooth operating condition. The latter requires charging the test unit with different mass charges due to their different physical properties as stated previously.

The evaporator cooling capacity for **R-22**, **R-134**a, **R-407C** and **R-404A** shows a decrease with time due to the reduction of the refrigerant mass flow rate, **Fig. 6**. The cooling capacity of **R-134**a is lower than that of **R-22** by (18) %, mainly because the refrigerant mass flow rate of **R-134**a is lower than that of the **R-22** by about (20) %. Noting that the refrigerating effect of both refrigerants is about the same. The results show that the cooling capacity of the chiller when circulating **R-407C** is close to that of **R-22** and in some points exhibits slightly a lower value by (6) %. This is believed to be partly because of the refrigerant mass flow rate of **R-407C** is lower than that of **R-22** by (2) %.

Comparing the cooling capacity of **R-404A** with that of **R-22**, it is obvious that **R-404A** capacity is slightly lower than that of **R-22** by (6 to 12) %. Although the refrigerant mass flow rate of the refrigerant **R-404A** is higher than **R-22** by (17 to 35) % but the refrigerating effect of **R-404A** is lower than that of the **R-22** by (21 to 28) %.

Compressor Work Rate:

The actual work rate for **R-22**, **R-134**a and **R-407C** exhibited a slight increase with time, **Fig. 7**. The gradual increase of the actual work approaching a maximum in the time range between (30 to 50) minutes after the commencing of the chiller operation. Then, it deceases steadily until the end of the experiment. This is a result of increase in discharge temperature of refrigerant at the beginning of water circulation and then decreases when the discharge temperature of the refrigerant from the compressor reduces.

The actual work rate of **R-134**a is slightly lower than that of **R-22** at high evaporator inlet temperature range approaching a maximum after (30) minutes to a value about (11) %. This can be explained as follows; although the specific work of **R-134**a is higher than **R-22** by (29) %, the mass flow rate of **R-134**a is lower than that of the **R-22** by (20) %. As a result, the mutual effect of both of these parameters shows a reduction in the **R-134**a work than that of **R-22**. At low evaporator temperature range the actual work of **R-134**a is lower than that of **R-22** by (38) % because of the specific work of **R-134**a is lower than that of **R-22** by (23) % and the mass flow rate of **R-134**a is lower than that of **R-22**, **Fig. 7**.

It is obvious that the actual work rate of **R-407C** is higher than that of **R-22** by (32 to 58) % because of the specific work of **R407C** is higher than **R-22** by (63) % and the refrigerant mass flow rate of **R-407C** is close to the **R-22** values. The data reveals that the actual work rate of **R-404A** is higher than that of **R-22** by about twice at high evaporator temperature. This is due to the higher values of the specific work of **R404A** than that of **R-22** by (59) % and the refrigerant mass flow rate of **R-404A** is higher than **R-22**. The actual work of **R-404A** reduces gradually as the water circulating continue until become slightly higher than that of **R-22** by (26) % as a result of reduction of the specific work of the **R-404A**.

The results showed that the power consumption of **R-134**a is slight lower than **R-22** by (5 to 8) %. Whereas it is slightly higher for **R-407C** than that of **R-22** by (6) %, **Fig.8**. The data for the cycling mode also showed that the power consumed by the chiller when circulating **R-404A** was lower than that of **R-22** by (17) % as a maximum, **Fig. 8**.

Condenser Heat Rejection:

Figure 9 expresses the condenser heat rejected for **R-22**, **R-134**a, **R-407C** and **R-404A** variation with time. The result showed that the condenser load for **R-134**a is lower than that of **R-22** by (17) %. This is because of the evaporator cooling capacity of **R-134**a is lower than that of **R-22** by (18) % and the compressor work of **R-134**a is lower than that of **R-22** by a value of (38) %. It is clear that the condenser heat rejected of **R-407C** is slight higher than that of the **R-22** by (5 to 13) % since the compressors work is higher than that of **R-22** by (32 to 58) % although the evaporator cooling capacity of **R-407C** slight lower than that of **R-22**. As show in **Fig. 9**, the condenser heat rejected for **R-404A** is higher than that of **R-22** by (20) % at high evaporator temperature range. This is due to the behaviour of the actual work of **R-404A** when compared to that of **R-22**.

Coefficient of Performance:

The actual coefficient of performance shows a decrease with time until approaches to a minimum after water circulation and then shows a gradual increase, **Fig. 10**. The trend of the actual coefficient of performance curve shows opposite behaviour to the curve of the actual compressor work, Fig. 7. The actual coefficient of performance of **R-134**a is lower than that of **R-22** by (23) % at high evaporator operation temperature range, and higher than **R-22** by (29) % at low evaporator operation temperature range. **R-407C** exhibited lower actual coefficient of performance than that of the **R-22** by the range (19 to 23) %. The corresponding reduction for **R-404A** is in the range (22 to 55) % when compared with that of the **R-22** values.

The refrigerants **R-134**a and **R-407C** showed a lower values for the consumption coefficient of performance than those of the refrigerant **R-22** by (7.8) %, **Fig. 11**. Whereas, **R-404A** exhibited a higher values of consumption coefficient of performance than that of **R-22** by (9.5) %.

The Pressure-Enthalpy Diagram:

It is interesting to represent the experimental data of the test refrigerants on the p-h diagram. A typical measured data during this work are shown on this diagram for the **R-22** and **R-407C** refrigerants in **Fig. 12**. An available commercial computerized package called **Coolpack** was used for this object. It is obvious that the rate of change of pressure with enthalpy for the **R-407C** is flatter than that of the **R-22** refrigerant. Therefore, the **R-407C** exhibited a higher actual work rate than the base **R-22** refrigerant as shown in **Fig. 7**. The same argument may be applied for the other refrigerants and their revealed work rate.

CONCLUSIONS:

The present work for the cycling mode of the chiller showed the following findings according to the working conditions of the unit:

1- The operation parameters concerning the performance assessment of the water chiller showed a time dependence behaviour. These parameters including $\mathbf{Q_{evp}}$, $\mathbf{Q_{cond}}$, $\mathbf{W_{act}}$, $\mathbf{COP_{act}}$ and $\mathbf{COP_{cons}}$ revealed a reduction as the operating time of the chiller proceeds.

2- The actual work rate for the refrigerants **R-22**, **R-134**a and **R-407C** showed an increase with time approaching a maximum in the time range (30 to 50) minutes after the commencing of the chiller operation. On the contrary, the refrigerant **R-404A** showed a decrease of the actual work rate as the time proceeds after the commencing of the water chiller operation.

3- The performance comparison of the drop- in substitutes, **R-134**a, **R-407C** and the **R-404A** with the base design circulated refrigerant, **R-22**, is shown briefly in **Table 2**.

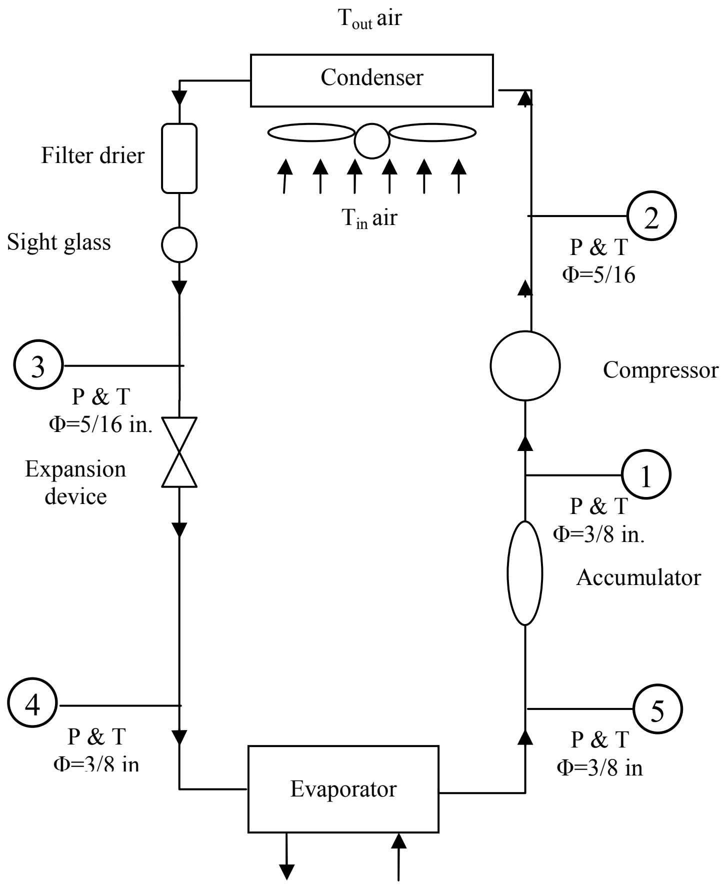


Figure 1: A schematic diagram for the refrigerant side of the water chiller.

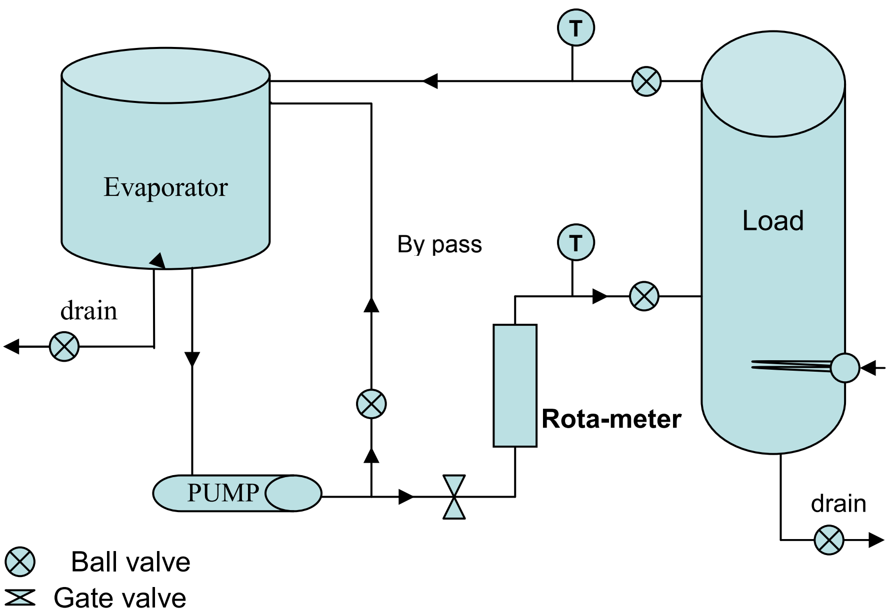
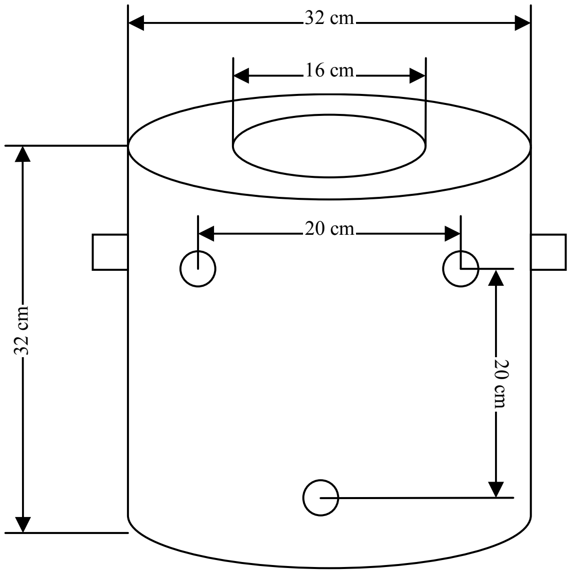
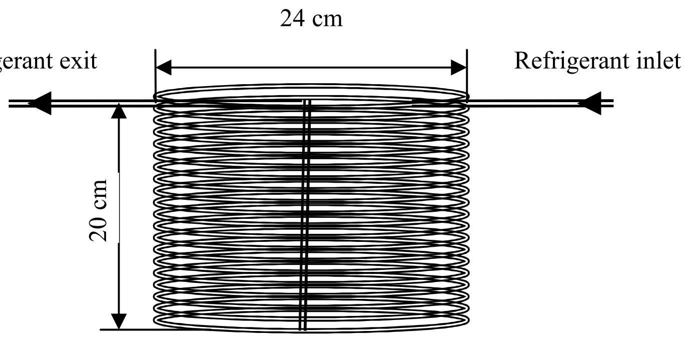


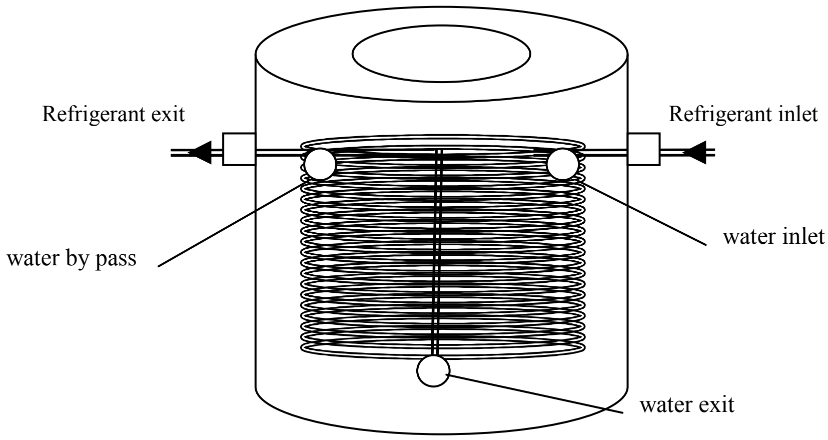
Figure 2: A schematic diagram for the water side path of the test unit.



(a) Shell of evaporator



(b) Coil of evaporator



(c) Shell and coil evaporator

Figure 3: A detailed schematic diagram of the test evaporator.

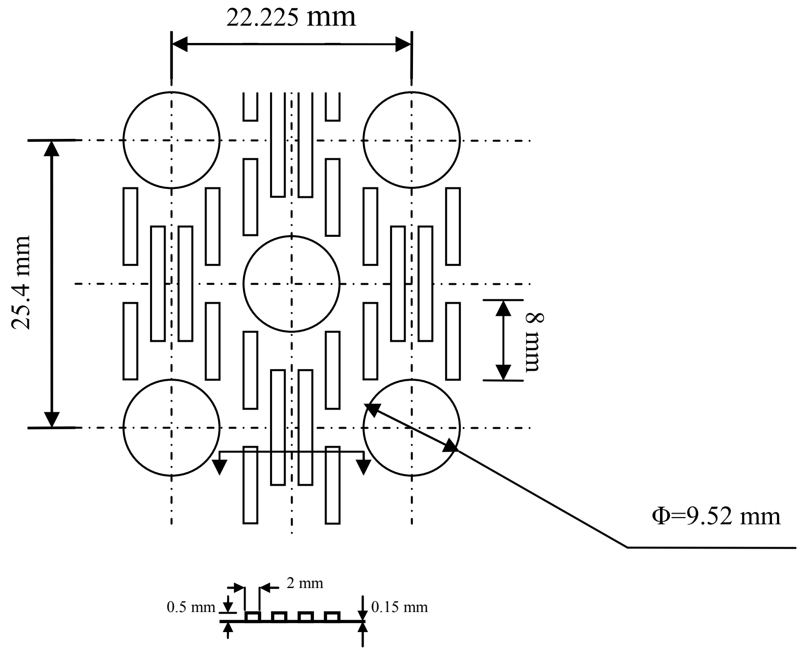


Figure 4: The tube and fins geometry of the condenser.

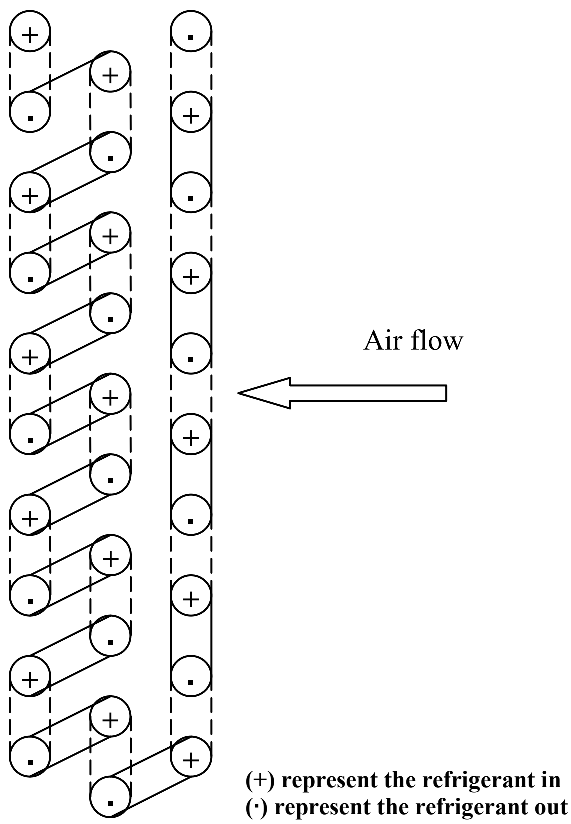


Figure 5: Arrangement of refrigerant tube circuiting of the test condenser.

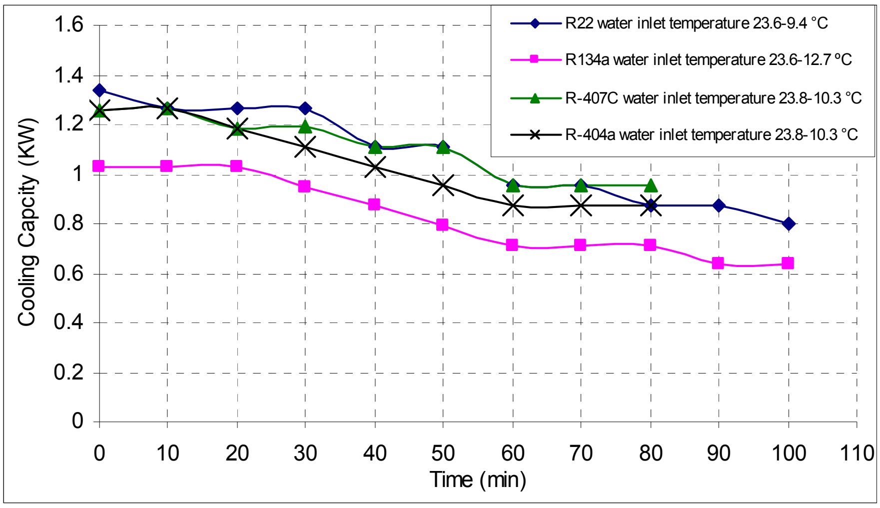


Figure 6: Comparison of the cooling capacity of the test refrigerants.

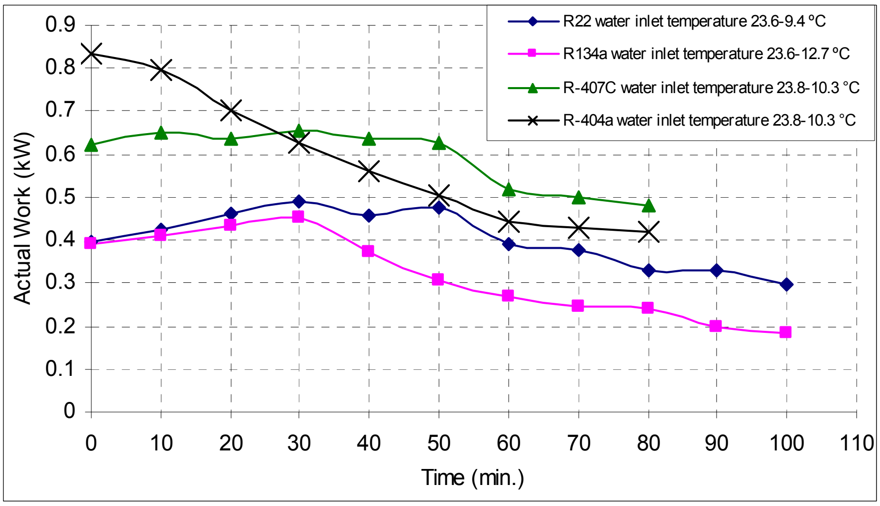


Figure 7: Comparison the actual work rate of the test refrigerants.

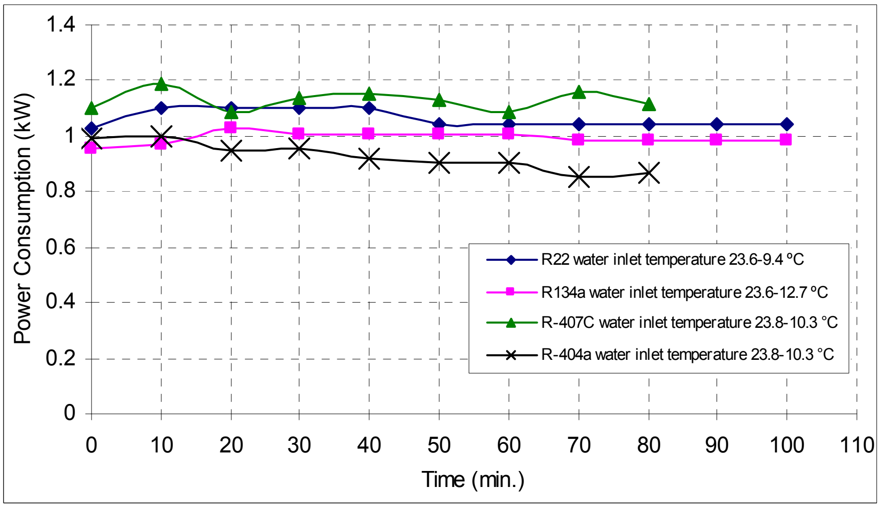


Figure 8: Comparison of the consumed power of the test refrigerants.

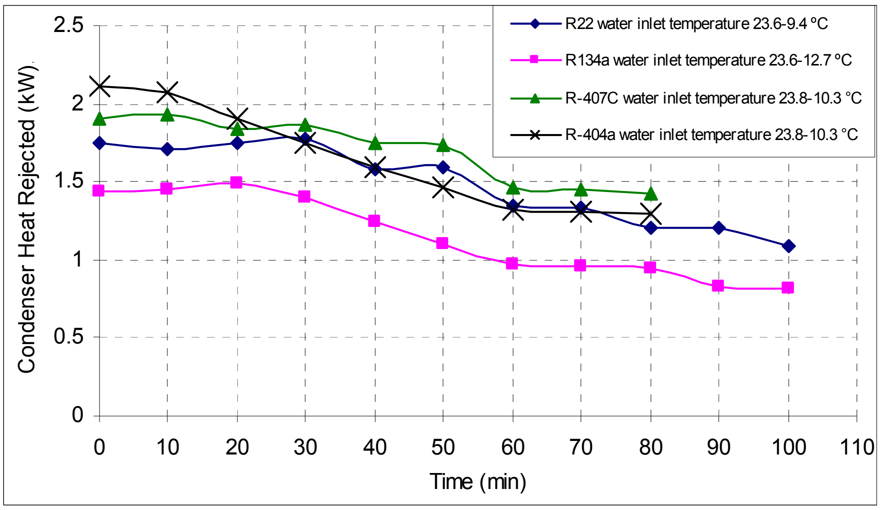


Figure 9: Comparison of the condenser load of the test refrigerants.

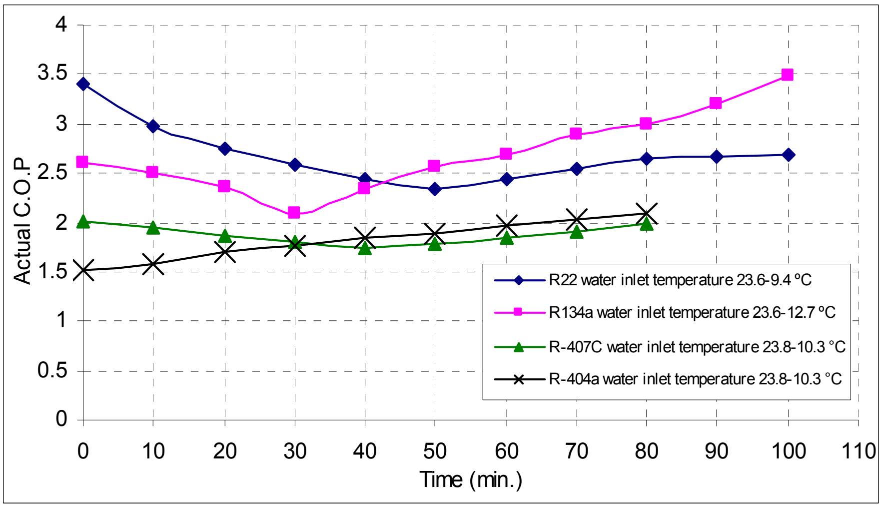


Figure 10: The actual coefficient of performance of the test refrigerants.

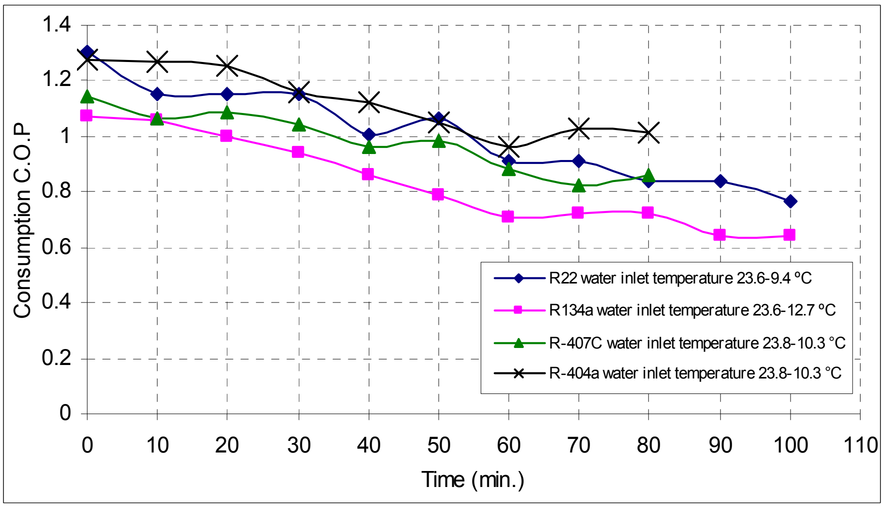


Figure 11: The consumed coefficient of performance of the test refrigerants.

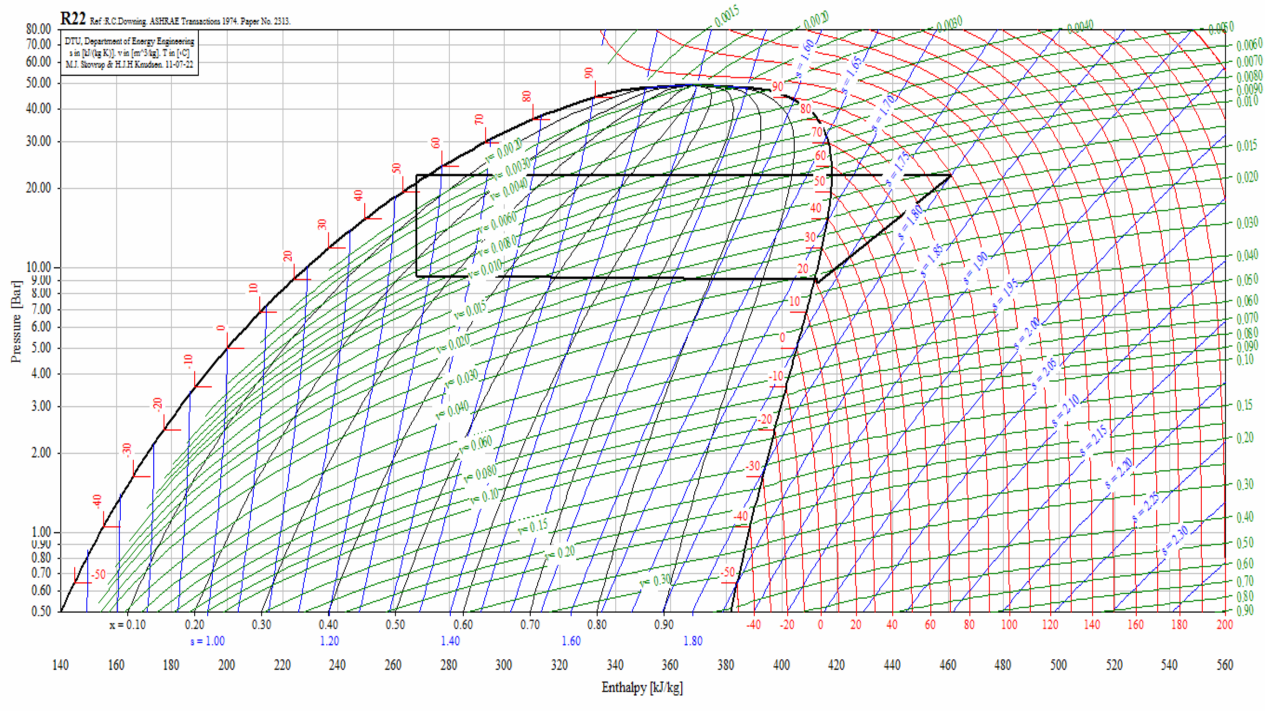


Figure 12.a: A typical *p-h* diagram for the *R-22* refrigerant test.

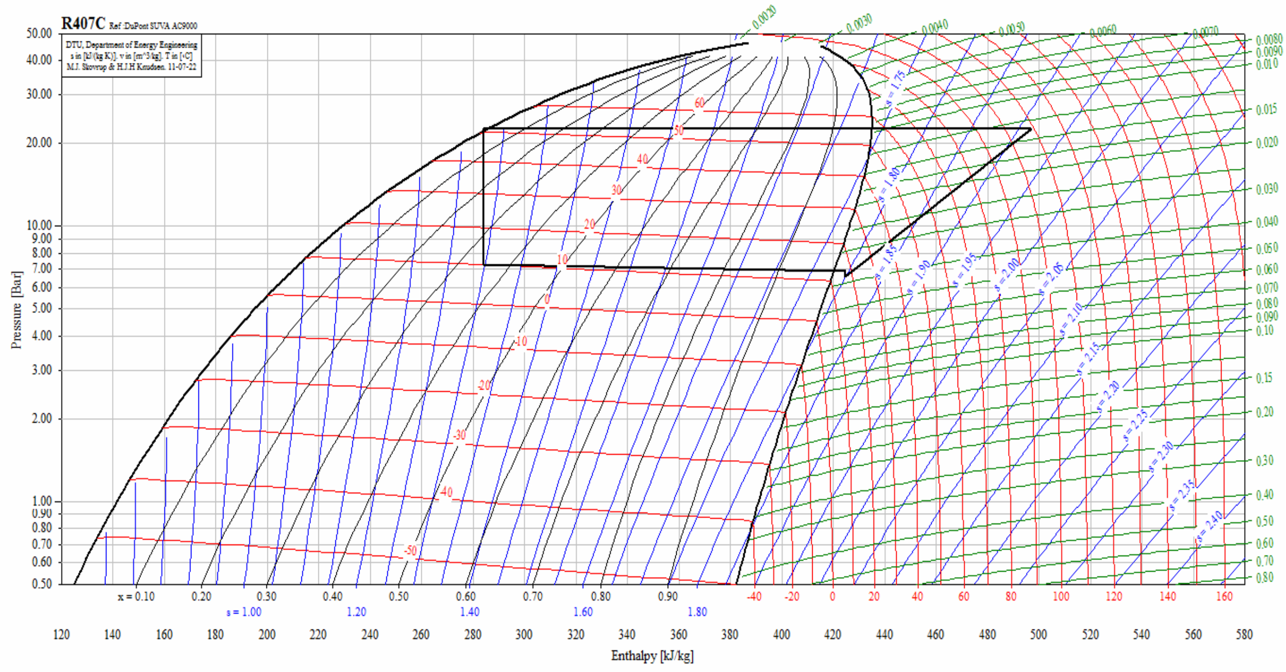


Figure 12.b: A typical *p-h* diagram for the *R-407C* refrigerant test.

Table 1: The experimental data collected when circulating *R-22* in the cooling unit.

Date		20-6-2009			Refrigerant type			R-22		Water side		Air side			Time
T1 (ºC)	T2 (ºC)	T3 (ºC)	T4 (ºC)	T5 (ºC)	P1 (bara)	P2 (bara)	P3 (bara)	P4 (bara)	P5 (bara)	T_{in} (ºC)	T_{out} (ºC)	T_{din} (ºC)	T_{win} (ºC)	T_{dout} (ºC)	
21.1	100	54	10.6	20	6.53	22.53	22.39	7.15	6.81	23.6	18.89	28	21	52	1:40
20.0	105	55	11.1	18.33	6.67	22.74	22.53	7.29	7.01	21.7	17.22	28	21	53	1:50
18.3	108	54	11.1	16.67	6.60	22.53	22.39	7.22	6.94	20.0	15.56	28	21	53	2:00
15.6	109	53	10.6	13.33	6.39	22.26	22.12	7.08	6.74	18.3	13.89	28	21	52	2:10
12.2	109	52	10	10	6.26	21.7	21.57	6.94	6.60	16. 7	12.78	28	21	51	2:20
8.3	108	52	8.3	7.78	6.05	21.29	21.15	6.67	6.39	15.0	11.11	28	21	50	2:30
6. 7	104	51	7. 8	6.11	5.84	20.88	20.74	6.46	6.12	13.3	10.00	27	20	49	2:40
5. 6	101	51	6.67	5.00	5.63	20.88	20.74	6.32	5.98	12.2	8.89	27	20	49	2:50
5.00	98	50	5. 6	4.44	5.50	20.32	20.19	6.12	5.84	10.8	7.78	27	20	48	3:00
3. 9	97	49	5.0	3.33	5.36	20.19	20.05	5.98	5.70	10.3	7.22	27	20	48	3.10
3.3	96	49	4.4	2.78	5.29	19.91	19.77	5.84	5.57	9.44	6.67	27	20	47	3:20

Table 2: Comparison of the quasi-steady performance of the drop-in refrigerants with respect to the *R-22* characteristic data.

Performance Parameter	***R-134a*** **(%)**	***R-407C*** **(%)**	***R-404A*** **(%)**
Cooling capacity	$(18)^{L}$	$(6)^{L}$	$(6\text{-}12)^{L}$
Actual compressor work rate	$(38)^{L}$	$(32\text{-}58)^{H}$	$(59\text{-}110)^{H}$
Actual compressor work rate per ton	$(23)^{L}$ to $(30)^{H}$	$(32\text{-}68)^{H}$	$(23\text{-}125)^{H}$
Condenser load	$(17)^{L}$	$(5\text{-}13)^{H}$	$(20)^{H}$
Compressor power consumption	$(5\text{-}8)^{L}$	$(6)^{H}$	$(17)^{L}$
Compressor power consumption per ton	$(8\text{-}28)^{H}$	$(7)^{H}$	$(9)^{L}$
$(COP)_{act}$	$(22)^{L}$ to $(29)^{H}$	$(19\text{-}23)^{L}$	$(22\text{-}55)^{L}$
$(COP)_{cons}$	$(8)^{L}$	$(8)^{L}$	$(9.5)^{H}$

L: Lower than that of the **R-22** value

H: Higher than that of the **R-22** value

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NOMENCLATURE:

cp: Specific heat of fluid, (kJ/kg. K)

h: Refrigerant enthalpy, (kJ/kg)

I: Electrical current, (Amp)

k: Thermal conductivity, (W/m K)

$\dot{m}_{ref}$: Mass flow rate of refrigerant, (kg/s)

P: Pressure, (bar)

Q: Heat load rate, (kW)

T: Temperature

V: Electrical Voltage (Volt)

W: Work rate, (kW)

Subscript:

act: Actual

cond: Condenser

cons: Consumed

d: Dry bulb

evp: Evaporator

in: Inlet

out: Outlet

w: Wet bulb or water