COLD STORAGE ROOMPEFORMANCE TESTING WITH AITERNATIVE REFRIGERANTS[+]

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Abstract:

Before selecting any alternative refrigerant for R-12, a performance study for a refrigeration cycle should be carried out using different types of refrigerants. A cold storage room of 3.8 kW capacity is used for this purpose.

Two types of hydrocarbon refrigerants, in addition too R-12, are used to charge the system separately. Namely propane (R-290) and mixture of R-290/ R-600a (68/32 wt.). Indoor loads are simulated by several halogen lambs. While the forced air cooled condenser of the unit is separated from the outdoor condition by a secondary air conditioning unit to control the condensing temperature. The effect of several parameters on the system performance and system matching are investigated. The results are presented for a cold storage room of 3.8 kW cooling capacity with a dimension of $(1.7m \times 1.7m \times 2.25m)$, and show that R-290 and hydrocarbon mixture give comparable thermal performance for the system, but hydrocarbon mixture could be suitable alternative for R-12 with out any modification in the system components that worked on the R-12.

المستخلص

قبل اختيار أي بديل لمائع تثليج **R-12** يجب دراسة أداء دورة التثليج الانضغاطية عند عملها على موائع التثليج المقترحة كبديل. تم استخدام غرفة مبردة بحمل تثليج مقداره **3.8 kW** لغرض دراسة أداء دورة التبريد باستخدام عدة موائع تثليج بديلة ، إضافة الى مائع التثليج $R-12$ ،. تم دراسة مائعي تثليج بديلين لهذا الغرض، مائع التثليج الاول يسمى البروبين $R-290$ و الثاني هو خليط هيدروكاربوني من البروبين و الايسوبيوتان$(R-29/R-600a)$ بنسبة وزنية تساوي $(68/32wt)$، حيث تم اعتماد هذه النسبة اعتمادا على بحوث سابقة تم الاشارة اليها في متن البحث. تم مماثلة الحمل الحراري للغرفة المبردة بواسطة عدد من المصابيح الهالوجينية. و تم أيضا السيطرة على وحدة تكثيف الغرفة المبردة عن طريق عزلها عن المحيط الخارجي بواسطة مجرى هوائي يمكن من خلاله التحكم بدرجة حرارة التكثيف.

درست العديد من المتغيرات التي يمكن ان تؤثر على أداء منظومة التثليج. بينت الدراسة ان مائع التثليج الهيدروكاربوني $R-290$ و الخليط الهيدروكاربوني $(R-290/R-600a)$ يعطيان معامل أداء متماثل لمنظومة التثليج. ولكن الخليط الهيدروكاربوني يمكن أن يكون اكثر ملائمة كبديل لمائع التثليج $R-12$ دون أجراء أي تغيير يذكر على منظومة التثليج التي تعمل على مائع التثليج $R-12$.

[+]Received on ١٦/٥/٢٠٠٦ Accepted on 1/8/2007
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Introduction:

The impending standards, a long will the reductions in the production of CFC-12 brought by Montreal protocol, have created a difficult problem for refrigerants that may increase energy consumption, at the same time the world is searching for many ways to reduce energy consumption. In addition, some requirements like non-toxicity; non- flammability and stability may be eliminated. Hydrocarbon and their mixtures could be suitable alternative refrigerants to replace many existing refrigerants; they don't pose any threat to the ozone layer and have low impact on global warming [1]. The most important issue regarding hydrocarbons as refrigerants is their flammability [2]. But it should be remembered that millions of tones of hydrocarbons are used safely throughout the world every year for cooking, heating, and powering vehicles and aerosol propellant [3].

Hussain [4] presentes, in his work, a computational model for vapour compression cycle of a simple refrigeration system using chlorofluorocarbons (CFCs) and hydrocarbon (HCs) refrigerants. Investigations of different parameters affecting the performance of the refrigerants system were carried out experimentally. The comparison between the experimental data and computational model yielded accurate results of accurate for coefficient of performance (COP) within -11.3 to -0.7% and -29.2 to -22.8% for R-12 and R-290 respectively, the authors showes that the error in COP using pure propane in computational model is due to imparities in propane used. The model predictes that HCs might be considered as alternative substance for CFC, and HCFCs.

Maclaine-cross and Leonard [5] consider the hydrocarbon refrigerants as the most promising alternative. They states hydrocarbon, like, propane, normal butane, and isobutene occur naturally in massive quantities in petroleum gases. The vapour pressure range of hydrocarbon refrigerants matches that of fluorocarbon refrigerant, but their significantly lower molecular mass gives them superior transport properties. They are good electric insulators and compatible with plastic synthetic, lubricants. They are good electric insulators and compatible with plastic insulation and sealant in refrigeration systems. They are non-corrosive and chemically stable.

1. Experimental Rig Layout and Testing Procedure:

A cold storage room of 3.8 kW cooling capacity and dimensions of $\mathbf{(1.7m \times 1.7m \times 2.25m)}$ is used to study the performance of the vapour compression cycle. Indoor loads are simulated by several haloginated lamps, of rated power ranging from 500 to 1250 W, while the outdoor conditions are simulated and controlled by a secondary air conditioner unit which, consists From two electrical heaters of 1500 W rated power each, cooling coil of 1.5 ton capacity and axial fan of 7.6 m^3/min capacity. Thermocouples are used to measure the evaporating, condensing, suction and discharge temperatures, while the mean indoor and outdoor temperatures are measured by several thermocouples which are distributed rationally inside cold storage room and condenser chamber. Fig 1 shows the experimental rig layout.

Three parameters that affect the cycle performance are studied, namely outdoor temperature that ranges from 30 to 45^{o}C in step of 5 ^{o}C, storage temperature that ranges from 30 to 45 ^{o}C in step of 5^{o}C also, and cooling load that ranging from 500 to 1250 W in step of 250 W. In addition too, three types of refrigerant are used to charge the cycle separately, they are, CFC-12, HC-290 and mixture of HC-290/HC-600a of 62/32 wt.

2.Results and Discussion:

Fig 3 shows the condensing pressure as a function of condensing temperature, at 5 ^{o}C storage temperature and 500 W cooling load. It can be seen from the figure that the maximum condensing pressure is for R-290, normal boiling point for R-290 is 42.1 ^{o}C at ambient temperature of 25 ^{o}C. While the minimum condensing pressure is for R-12, normal boiling point of R-12 is -29.8 ^{o}C at ambient temperature of 25 ^{o}C, i.e. a substance of low boiling point has a high vapour pressure. Generally, evaporator should work at a pressure above atmospheric pressure, and the condensing pressure should not be so high [٦]. The figure shows also, that the condensing pressure of hydrocarbon mixture is more closer to that for R-12 than that for R-290.

Fig 4 shows the compression ratio as a function of condensing temperature, at 5 ^{o}C storage temperature and 500 W cooling load. It can be seen from the figure that, the difference in pressure ratio between the refrigerants mentioned above, tends to vanish. This means that R-290 must work at high evaporating pressure.

Fig 5 shows the degree of sub-cooling of liquid refrigerant as a function of cooling load, at 5^{o}C storage temperature and 30 ^{o}C outdoor temperature. It can be seen from the figure that, the degree of sub-cooling of hydrocarbon mixture is less than that for R-12 by 1.3 ^{o}C, while for R-290 it is less than that for R-12 by 1 ^{o}C. As the degree of sub-cooling increases the refrigeration effect per one kg of refrigerant circulated is increased, and the condensation process becomes more efficient [7].

Fig 6 shows the degree of superheating of vapour refrigerant in the evaporator as a function of cooling load at 5^{o}C storage temperature, and 30 ^{o}C outdoor temperature. It can be seen from the figure that the degree of superheating of R-12 is always greater than R-290 and hydrocarbon mixture, the figure shows, also, insignificant difference in the degree of superheating between the three refrigerants mentioned above, namely -0.5 ^{o}C for hydrocarbon mixture and -0.3 ^{o}C for R-290 as compared with that for R-12.

Fig 7 shows the volumetric heat capacity as a function of evaporating temperature at 5 ^{o}C storage temperature and 500 W cooling load. As the volumetric heat capacity increases the physical dimension of compressor decreases. It can be seen from the figure that, the maximum volumetric heat capacity is for R-290, and the minimum is for hydrocarbon mixture. The figure shows also, that the difference in heat capacities, between the refrigerants mentioned above, becomes more closes as the evaporation temperature decreases.

Fig 8 shows the refrigerant volume flow rate as a function of evaporating temperature, at 5 ^{o}C storage temperature and 500W cooling load. It can be seen from the figure that the maximum volume flow rate is for R-12, and the minimum is for R-290, this means that, when using R-290 as an alternative, compressor displacement must be changed and the compressor should be replaced with smaller one.

Fig 9 shows the refrigerant mass flow rate as a function of evaporating temperature, at 5 ^{o}C storage temperature and 500W cooling load. Mass flow rate is associated, at any point, with volume flow rate that is calculated at compressor entry. It can be seen from the figure that the maximum mass flow rate is for R-12, and the minimum is for R-290.

Fig 10 shows the dynamics change of storage temperature against time, at 500W cooling rate and 30 ^{o}C outdoor temperature. The specific latent heat of evaporation depends on the molecular weights, thus, largest enthalpies of evaporation are found for substance with high molecules weight and vice versa [6], From the above, it can be seen from the figure that R-290 needs less time, of about 52 min. to reach the required storage temperature of 5 ^oC, this is due to its high latent heat of evaporation, while hydrocarbon mixture needs more time, of about 81 min. R-12 needs about 65 min. to reach the required storage temperature mentioned above.

Fig 11 shows the compressor motor power as a function of condensing temperature, at 5 ^oC storage temperature, 500W cooling load. It can be seen from the figure, that the compressor motor power consumption increases with increasing of condensing temperature. The compressor motor power consumption for R-12 is 1.1 and 1.25 kW when the condensing temperature are 44 and 52 ^oC, respectively. On the other hand they are 1.15 and 1.23 kW for R-290 and hydrocarbon mixture, respectively. The work compression per kg of circulated refrigerant increases as the condensing temperature increases, thus, from the stand point of power, a low condensing temperature is desirable, and condenser should use the coldest air [8].

Figure 12 shows the COP as a function of condensing temperature, at 500W cooling load and 5 ^oC storage temperature, COP decreases with the increase of condensing temperature at any value of the evaporating temperature [9]. It can be seen from the figure that, although the maximum COP is for R-12, the difference in COPs of the cycle, using the refrigerant mentioned above, tends to vanish. This fact also stated by Gonsey [10],[11]

3.Conclusions:

Many conclusions can be derived from this study, which can be summarized as follows:

1. For the same degree of sub cooling R-290 gives the same increases in the refrigeration effect as when using R-12.
2. Compressor power consumption for hydrocarbon mixture is closes to that for R-12 than that for R-290.
3. Time required for R-290 to reach a given storage temperature is shorter than that for R-12 and that for hydrocarbon mixture.
4. COP for the cycle operating with hydrocarbon mixture is much closer to that for R-12 than that for R-290.
5. Minimum volume flow rate is for R-290; thus smaller compressor should be used when this refrigerant is selected as an alternative for R-12.
6. Hydrocarbon mixture R-290/R-600a of 62/38 wt. gives approximately the same performance R-12 and could be used as an alternative refrigerant out with any modification to the vapour compression system components.

Fig(1) Experimental rig

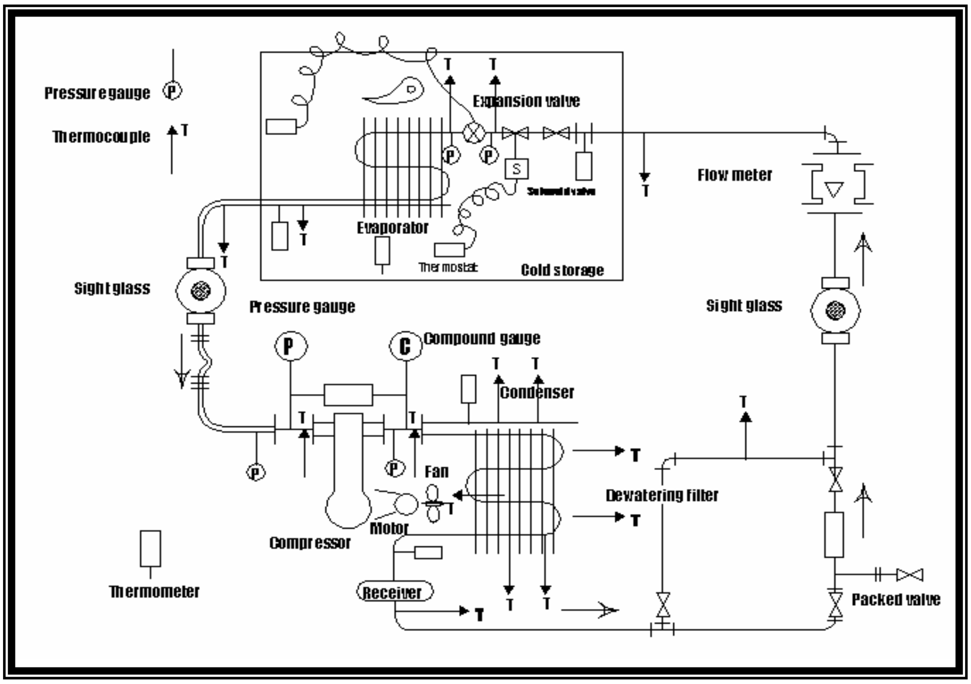
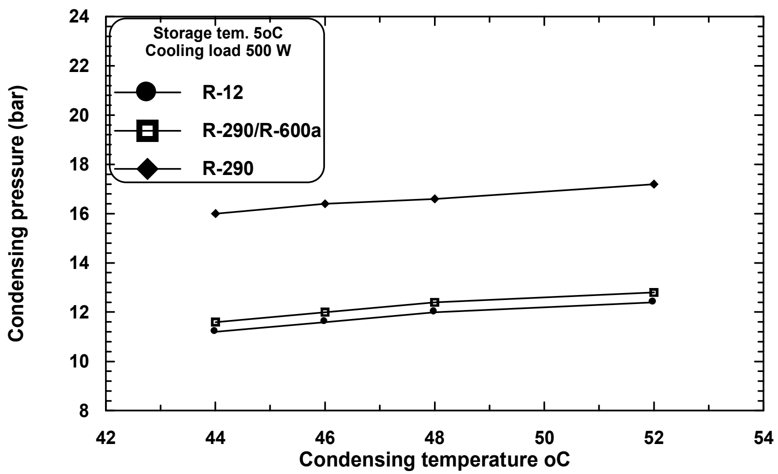
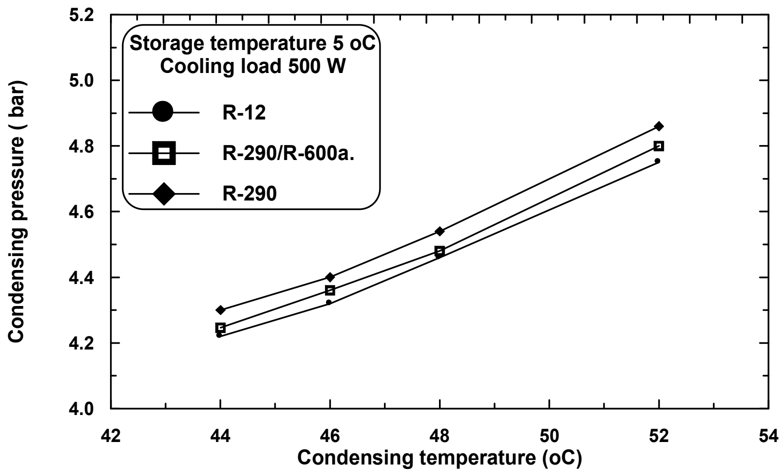


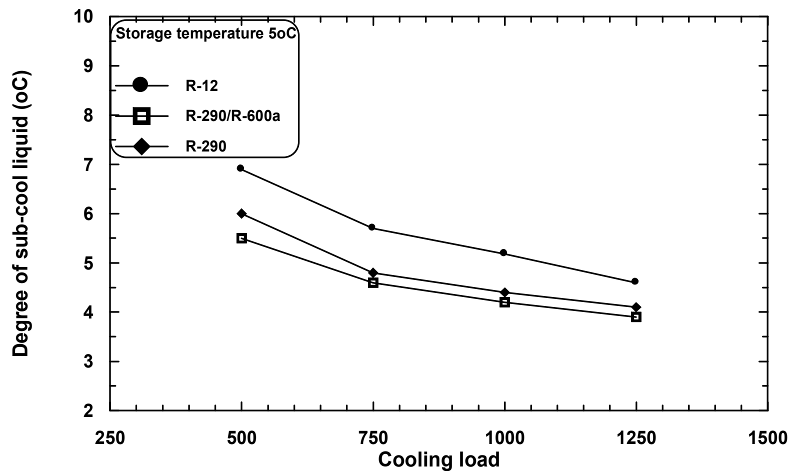
Fig.(2) Experimental rig layout showing the position of temperature and



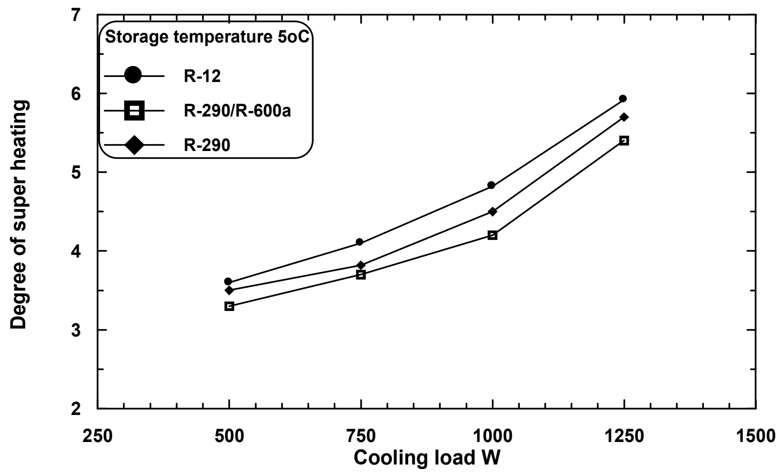
Figure(3) Condensing pressure VS condensing temperature



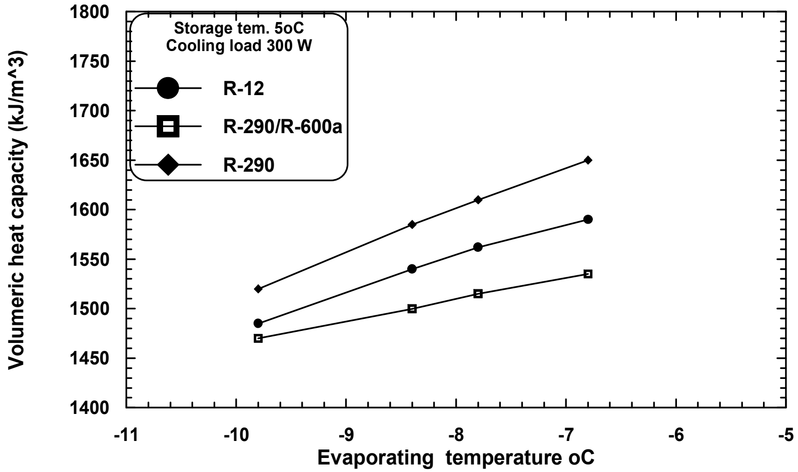
Figure(4) Cpmpression ratio VS condensing temperature



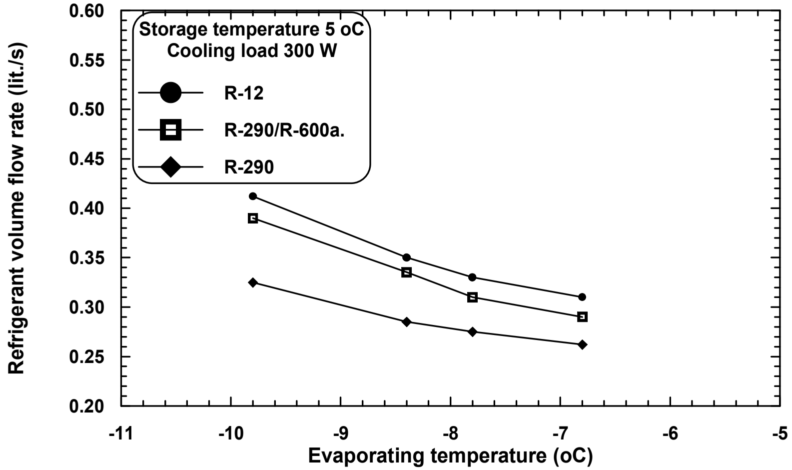
Figure(5) Degree of sub-cooling VS cooling load



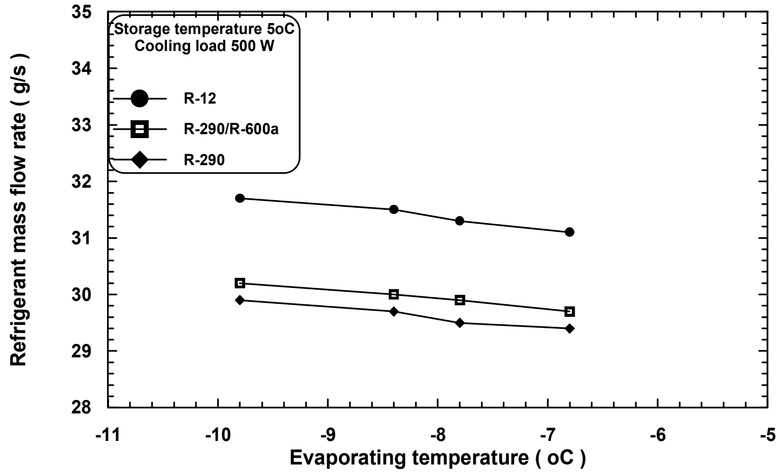
Figure(6) Degree of super heating(in the evaporator) VS cooling load



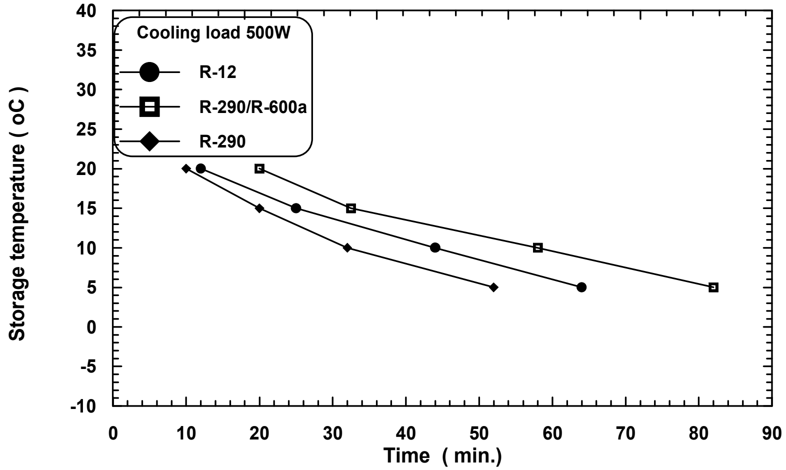
Figure(7) Volumetric heat capacity VS evaporating temperature



Figure(8) Refrigerant volume flow rate VS evaporating temperature



Figure(9) Refrigerant mass flow rate VS evaporating temperature



Figure(10) Storage temperature VS time

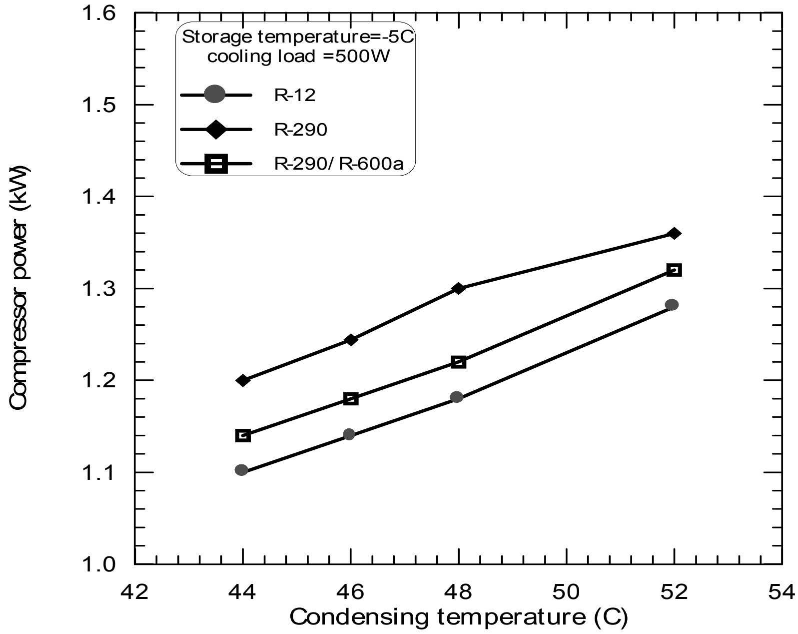


Fig.(11) Compressor power VS condensing temperature

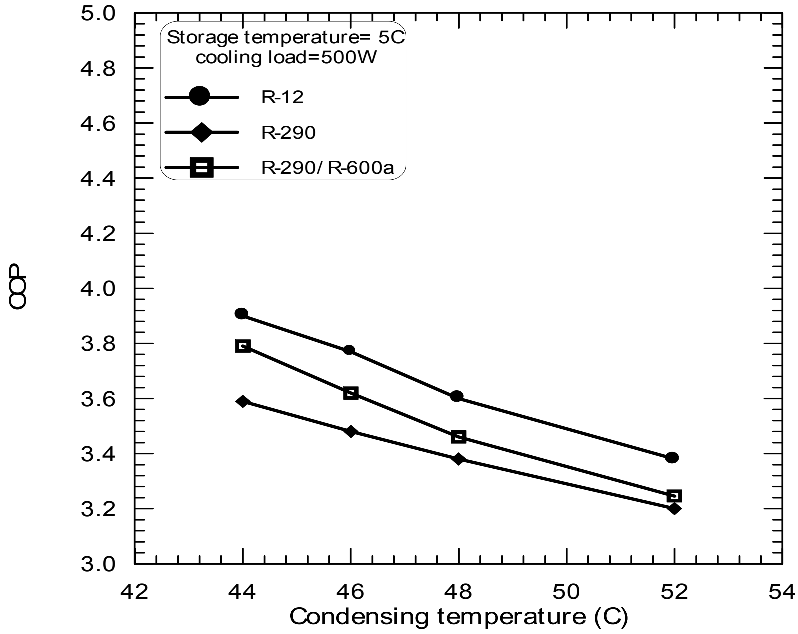


Fig.(12) COP VS condensing temperature

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