Comparative Exergetic Analysis of Vapor Compression Refrigeration Systems in the Superheated and Subcooled Regions

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Abstract

This work evaluated how energy is utilized in two-vapor compression refrigeration systems. These two systems were tagged rig a and rig b. Rig a was constructed from a design model; while rig b was assembled from equivalent components available in the market. Both rigs were of equal capacities (10 kW). The energy usage of these rigs, outside the saturated region – superheated and saturated regions- was compared. Within the experimental limit, rig b required about 20% energy input than rig a. While in the sub-cool region rig b required 25% input energy than rig a for optimum performance. Since energy, in the running cost of mechanical refrigeration systems, takes the largest share. It was shown by this study that the running cost can be reduced by: basing the component design on balance points; not encouraging superheating and limiting sub-cooling to between 3° and 5°C.

Keywords: Exergetic, energy degradation, sub-cooling, superheating, regions, COP, refrigeration systems, anergy.

Introduction

Energy is the single largest running cost of mechanical refrigeration systems, whether it is used for: food preservation and storage; air conditioning; or other industrial uses (Sahin and Kodal 2002a). Many mechanical refrigeration systems installed in air conditioning, food processing and other industrial applications have been design to meet a low capital cost criterion (Cleland, 1999). Many researchers have worked on the modifications of such plants to lower running costs, particularly energy (Akintunde et al., 2004; Chen et al., 2002; Klein 1994; Schlage et al., 1990; Sahin and Kadal 2002b). In these works, simple calculation methods based on empirical formulae for calculation of energy usage were presented. These methods allow the energy and hence the running cost and savings resulting from plant changes to be estimated. All these works were focused on the normal refrigeration cycles without taken into consideration the subcooling and superheated areas. This work compared the energy utilization in two mechanical refrigeration systems (rig a and rig b) of equal capacities based on exergetic point of view. This comparative analysis was done on these systems outside the saturation region were extra energy is required for the running of the system.

The aims of this investigation are: to ascertain the effect of balanced point on the energy consumption of mechanical refrigeration systems; to point out which of these rigs make use of the available energy effectively and which of these devices waste or degrade energy more.

The energy parameters from the mechanical refrigeration systems (rigs a and b) was estimated form the following equations (1) to (4), as suggested by Jabardo et al. (2002):

\[ Q_e = m_e (h_{ce} - h_{c}) , \quad (1) \]

\[ Q_c = m_r (h_{ce} - h_{cl}) , \quad (2) \]

\[ P = m_r (h_{scl} - h_{spl}) , \quad (3) \]

\[ COP = \frac{Q_e}{P} , \quad (4) \]
where:

- $h$ = the enthalpy (kJ/kg K);
- $m$ = the refrigerant mass flow rate (kg/s);

and the subscripts are:

- $ee$ - evaporator exit;
- $ei$ - evaporator inlet;
- $ce$ - condenser exit;
- $ci$ - condenser inlet;
- $cpe$ - compressor exit;
- $cpi$ - compressor inlet;
- $r$ - refrigerant.

According to Adegoke and Akintunde (1999), “Exergy” can be defined as that part of the energy capable of doing work while “Anergy” is the remaining energy. The key exergy parameters for this analysis are: heat exergy ($X$); heat anergy ($Y$); exergetic potential ($\gamma$); and exergetic efficiency ($\eta_x$). These parameters were evaluated from Equations (5) to (13) as suggested by O’Callaghan, (1981) and reported by Adegoke and Akintunde (1999):

- Heat anergy: $Y = \frac{T_o}{T} Q$, \hspace{1cm} (5)
- Heat exergy: $X = \gamma Q$, \hspace{1cm} (6)
- $\gamma = \frac{T - T_0}{T}$. \hspace{1cm} (7)

Thus:

$$Y = (1 - \gamma) Q = Q - X.$$ \hspace{1cm} (8)

The exergetic efficiency of a process ($\eta_x$) is given by Equations (9) and (10):

$$\eta_x = \frac{\text{output energy}}{\text{input energy}},$$ \hspace{1cm} (9)

$$\eta_x = \frac{X_{out}}{X_{in}} = \frac{\gamma_{out}}{\gamma_{in}} \frac{Q_{out}}{Q_{in}}.$$ \hspace{1cm} (10)

The energy efficiency, $\eta_e$, can therefore be defined as shown in equation (11):

$$\eta_e = \frac{Q_{out}}{Q_{in}}.$$ \hspace{1cm} (11)

Then

$$\eta_x = \frac{\gamma_{out}}{\gamma_{in}} \eta_e.$$ \hspace{1cm} (12)

Exergetic potential is a measure of the ability of available energy to produce work (a pure energy form having $\gamma = 100\%$). Exergetic efficiency is an inverse of energy degradation. The lower the value of $\gamma$, the more the quality of energy has been downgraded. O’Callaghan (1981) defined an exergetic degradation factor as:

$$K_x = 1 - \eta_x = 1 - \frac{\gamma_{out}}{\gamma_{in}} \eta_e$$ \hspace{1cm} (13)

Exergetic efficiency is also a measure of effectiveness of the system as it combines effectiveness with thermodynamic efficiency (Adegoke and Akintunde 1999). These exergetic parameters can therefore be utilized to evaluate the vapour compression refrigeration system’s performance, since the system is thermo-dynamically based.

**Materials and Methods**

Two mechanical refrigeration systems were developed as rigs $a$ and $b$. Rig $a$ was constructed using data generated from a computer model designed for vapour compression refrigeration systems by Akintunde (2003). This computer model established “balanced points” between the four major components (compressor, condenser, evaporator and expansion device) of the refrigeration system. Rig $b$ was constructed from bought-out standard components. These two rigs have capacities of 93.25 W, respectively.

The conditions around the two heat exchangers (condenser and evaporator) were varied by circulation of water around them at various rates and different temperatures. This provided “hypothetical variable load” for evaporator and different environmental conditions for condenser. Temperature and pressure gauges were fixed at both high and low pressure sides to measure, simultaneously both pressure and temperature at various load. The temperatures and pressures were used in conjunction with thermodynamics steam table developed by Stoecker and Jones (1982) and Equations (1) to (4) suggested by Jabardo et al. (2002), through which the performance parameters, such as refrigeration capacity ($Q_e$), compression work ($P$), condenser capacity ($Q_c$) and the coefficient of performance (COP) were estimated. These parameters were used to compute exergetic parameters as show in Equations (5) to (13).
Results and Discussion

The total energy of a system is the sum of useful energy or the available energy (exergy) and unavailable energy (anergy). In the systems used for this study, capillary tube was used as the expansion device and it was assumed that adiabatic condition prevailed in this component of the system, (Kim et al., 2002; Wijaya 1992; Chang and Ro 1996; Akintunde 2003a). As a result, three major components were involved in the energy exchange – compressor, condenser and evaporator. Energy from evaporator was estimated using Equation (1) while that of condenser was estimated using Equation (3) and the corresponding compressor power from Equation (2). These results were used to estimate the coefficient of performance using Equation (4).

The exergetic parameters were estimated as follows: heat anergy was estimated from Equation (5 or 8); exergetic potential from Equation (7) and heat exergy from Equation (6). Exergetic efficiency was estimated using Equations (10) or (12) and energy efficiency from Equation (11). The exergetic degradation factors were estimated from Equation (13).

Discussion

Superheated Region

From the evaporator side, Fig. 1 shows the variation of COP with the degree of superheat. This figure shows that as the degree of superheat increases the COP decreases almost linearly with the degree of superheat. This indicated that superheating should not be encouraged at all. It can be observed also that the COP of ‘rig a’ is always higher than that of rig b. This shows that rig a required less energy input than rig b for optimum performance. Fig. 2 explains this further. When the heat anergies were compared, the heat anergy of rig b is higher, hence a lower exergy as compared with rig a. Since less heat is “available” or absorbed by the evaporator of rig b in the superheated region, these results into a reduced COP.
From condenser side, Figs. 3a and 3b show the variation of anergy and exergy, respectively with the degree of superheat. From Fig. 3a, the heat anergy of rig a is more than that of rig b. This shows that at the same atmospheric conditions, condenser of rig a rejected more heat to the atmosphere than that of rig b. This was further corroborated by Fig.3b which shows that the heat gain from the atmosphere by the condenser of rig a is less than that of rig b. In other words, the condenser of rig a rejected heat faster than that of rig b. Fig. 4 shows the total anergy and exergy of the systems. This figure shows that rig a performed better than rig b if the forgone explanations are followed.
Sub-cooled Region

From evaporator side, Fig. 5 shows that the COP of the systems increases with increase in the degree of sub-cooling. At the same time the COP of rig $b$ is about 6% less than that of rig $a$. Fig. 6 compares the variation of anergy and exergy with the degree of subcooling. Both systems showed anomalies between $0^\circ$ and $2^\circ$C. The cause of these anomalies is under investigation as reported by Ibrahim (2001) that condenser performance shows some anomalies between $1^\circ$ and $2.5^\circ$C of subcooling. Between $2^\circ$ and $5^\circ$C of sub-cooling there is a slight increase in the average heat exergy. After $5^\circ$C, heat exergy shows noticeable decrease for both rigs, though rig $b$ was the worst affected. At this same point of $5^\circ$C subcooling, heat exergy shows some noticeable increase. This shows that for optimum conservation of energy, degree of sub-cooling should not be more than $5^\circ$C.
The rate at which the condenser of rig $a$ rejected heat is more than that of rig $b$ by an average of 67% this is as indicated in Fig. 7a. This is further explained by Fig. 7b, which shows that condenser of rig $b$ retained more heat energy than that of rig $a$. Fig. 8 shows the variation of total anergy and exergy of the rigs, with the degree of subcooling. This figure shows that, for overall performance in the subcooled region rig $a$ utilized the available energy more effectively than rig $b$. 

Fig. 7a. Variation of Evaporator Anergy with Degree of Subcooling

Fig. 7b. Variation of Condenser Exergy with Degree of Subcooling

Fig. 8. Variation of Total Anergy/Exergy of the System with Degree of Subcooling
**Exergetic Efficiency**

Fig. 9a shows that the exergetic efficiency of the systems decreases with degree of superheating. This indicated that in systems designed for optimum performance superheating should not be encouraged. While in Fig. 9b, though there are fluctuations, but within 2° and 4°C the exergetic efficiency indicated that subcooling aids the system performance to some extents. If subcooling is to be allowed it should not be more than 5°C for the refrigeration systems of small capacities such as considered in this study.

**Exergetic Degradation**

Fig. 10a shows a general degradation of energy with the increase in the degree of superheat for both rigs. This corroborates the forgone argument that superheating should not be encouraged, with regard to energy conservation and optimum system performance.

Define the relative exergy degradation ($K_r$) as:

$$K_r = \frac{K_{xb}}{K_{xa}},$$

or

$$K_p = 1 / K_r. \quad (14)$$
The general trend of $K_r$ and $K_p$ show that rig $b$ degraded energy more than rig $a$. This shows the effect of balance points upon which the computer-aided design model used to design rig $a$ was based.

From subcooling effect, degradation of energy as indicated in Fig. 10b, shows that the curves flattened out between 3°C and 5°C of subcooling and slightly increase after 5°C. This shows that optimum performance required that the degree of subcooling should range between 3°C and 5°C. Any sub-cooling below this may have no significant effect on energy conservation. In Fig. 10b also since $K_r$ is always less than $K_p$, this shows that rig $b$ degraded energy faster than rig $a$. This indicated that a balance point between the system components is required for optimum performance.

**Conclusion**

In this study it has been clearly shown that, the effect of imbalance between the major components of the refrigeration systems makes the system to degrade the available energy. In the comparison of the two rigs used in this study, it was shown that rig $a$ performed better than rig $b$ because of the fact that the design model used was based on balanced points. It has been identified that as a result of imbalance between the components of rig $b$, any change in the environmental condition affects negatively the system performance.

Since energy in the running cost of mechanical refrigeration systems takes the largest share. It has been shown in this study that this running cost can be reduced by: basing the component design on balance points; not
encourage superheating and limiting subcooling to between 3° and 5°C.

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References


Nomenclatures

COP = Coefficient of performance
$K_p$ = Relative exergetic degradation factor based on rig b.
$K_r$ = Relative exergetic degradation factor based on rig a
$K_x$ = Exergetic degradation factor
$Q$ = Rate of heat rejected or absorbed (W)
$T$ = Absolute temperature of which heat is released (K)
$T_0$ = Prevailing environmental absolute temperature (K)

$X$ = Heat exergy
$Y$ = Heat anergy
$\gamma$ = Exergetic potential
$\eta_e$ = Thermodynamics or Energy efficiency. ($\eta_e$) or $\eta_x$ = Exergetic efficiency.

Other subscripts used:
a - used as subscript to indicate rig a
b - used as subscript to indicate rig b
c - used as subscript to indicate condenser
e - used as subscript to indicate evaporator