American Journal of Engineering Research (AJER)
e-ISSN: 2320-0847 p-ISSN : 2320-0936
Volume-6, Issue-7, pp-58-69
www.ajer.org
Research Paper Open Access

The Effect of Condenserfluid Quality on the Performance of a Vapour Compression Refrigeration Cycle "A Simulation Model Approach"

[*1]Njoku Cosmos Chibuzo, [2]KinigomaBoma Samuel, [3]Iliyasu Nacha
Department of Petroleum and Gas Engineering, University of Port Harcourt, Port Harcourt, River state, Nigeria

Abstract: *The paper investigated the effect of incomplete condensation / condenser fluid quality on the performance of vapour compression refrigeration cycles. Simulation model was developed using ASPEN HYSYS version 7.1 to determine the relationship between condensers fluid quality and some refrigeration system performance parameters such as compressor duty, cooling effect (CE) and the ratio of cooling effect and compressor duty for a simple propane pre cooled loop. The simulation was carried out at condensation temperature range of 0 to 60℃ and fluid quality reduction interval of 5% using three different condensers of capacities 300mj/hr, 500mj/hr and 700mj/hr respectively and plant process data obtained from an LNG facility in Nigeria. The analysis of the results shows that decrease in the condensers fluid quality (increase in the degree of incomplete condensation) leads to decrease in the performance of a vapor compression refrigeration system and an increase in the condenser fluid quality leads to increase in the performance of the system and the fluid quality failure point of the systemis dependent on the condensation temperature but independent on the size or heat capacity of the condenser.Hence at condensation temperatures of 0℃, 20℃,40℃ and 60℃, the fluid quality failure points are 7%, 14%, 22% and 31% respectively irrespective of the size or heat capacity of the condenser.*

Keywords*;condenser.fluid quality .vaporcompression refrigeration cycle. Condensation.Simulation model.*

I. INTRODUCTION

Refrigeration cycles based on vapor compression refrigeration cycle (VCRC) are the world most recommended and generally accepted refrigeration cycle because of their high thermodynamic efficiency, safe and cost effective nature. They include the pure cascaded refrigeration system, propane precooled mixed refrigerant cycle (PPMR/C3MR), Philips optimized cascaded LNG process (POCLP), mixed fluid refrigerant process (MFR) etc. the vapor compression refrigeration cycle operates on the principle of using a refrigerant (working fluid) to extract heat from one part of the cycle and reject it to another part by mean of a compressor, heat exchangers (condenser and evaporator) and expansion system.

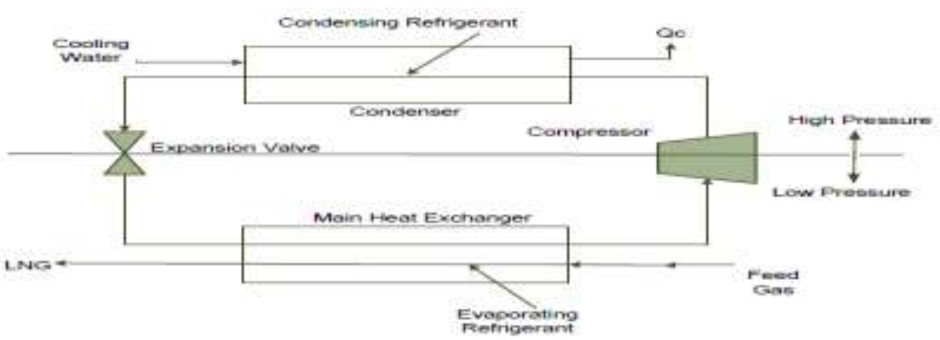

Figure 1 a simple closed loop vapor compression refrigeration cycle for natural gas liquefaction.

The condenser is one of the key units of the vapor compression refrigeration cycle According to Ajeet et al (2013), the condenser is a heat transfer unit designed to condense fluid from gaseous to liquid state through losing of their latent heat of vaporization to another fluid (condenser coolant). In condensers the refrigerant vapor condenses by rejecting heat to an external fluid, which acts as a heat sink. Condensation set in whenever a saturated vapor comes in contact with a surface whose temperature is lower than the saturated temperature corresponding to the vapor pressure. The main purpose of the condenser in a vapor compression refrigeration cycle is to extract the heat lost by the cooling fluid to the refrigerant during cooling and transfer it to the environment via an indirect heat flow contact between the refrigerant and the cooling water.

Incomplete condensation is a situation in a condenser when the outlet fluid coming out of the condenser has some traces of vapor that are not condensed during the process of condensation. Thermodynamically, this situation implies that these uncondensed vapors coming out of the condenser still retained their latent heat of vaporization and as such circulates through the system without rejecting heat to the coolant hence constituting a negative performance effect to the cycle in a closed system. This uncondensed vapor constitute a negative performance affect because the vapour flow round the loop without rejecting or extracting heat from the cooling fluid but consumes the work of the compressor that is used in circulating it through the loop.

According to Arora (2010),the condenser fluid quality is the ratio of volume of liquid to volume of entire fluid at the outlet of the condenser. A high fluid quality means a high volume of liquid in the fluid and a low fluid quality means a low volume of liquid in the fluid. The quality of a condenser outlet fluid depends on the state or vapor fraction of its fluid on leaving the condenser. A condenser is teamed highly effective if its fluid leaves as sub-cooled liquid (zero vapor fraction with additional temperature drop below inlet temperature), effective if its fluid leaves as saturated liquid (zero vapor fraction with zero temperature drop) and ineffective if its fluid leaves as double phase fluid (liquid and vapor).Therefore, the higher the vapor fraction of fluid leaving the condenser, the lower the effectiveness of the condenser. Hence, the relationship between fluid quality and vapor fraction is given as

Fluid quality = (1-vapor fraction) $\times$100 (1)

The performance of a refrigeration cycle is usually expressed using performance estimation parameters which include compressor duty, refrigeration effect (RE), cooling effects (CE), Coefficient of performance (COP), and volumetric refrigeration effect (VRE**).**

Compressorduty. This is the amount of shaft power input in the refrigeration system that is used to drive the compressor. It is usually expressed in kilowatt (kw). In a refrigeration system, the higher the amount of compressor duty, the lower the performance and the lower the compressor duty, the higher the performance (Rajput 2006)

Coefficient of performance (COP). This is the ratio of the heat capacity or the heat absorbed by the evaporator and the compressor duty. It has no unit since it express ratio of two energy variables. The higher the coefficient of performance the higher the performance and vice versa (Michael & Howard 2000)

COP=heat rejected by evaporator (kw) / compressor duty (kw)

Refrigeration effect (RE). This is the amount of heat energy absorbed by the evaporator in the cause of cooling the confined area. It is expressed (kw) or kj/hr. the higher the refrigeration effect the higher the performance and vice versa (Rajput 2006)

Cooling effect (CE) this is the temperature difference between the inlet natural gas and outlet pre cooled gas. It is expressed in$℃\ or ℉$. The higher the cooling affect the higher the performance and vice versa.

Volumetric refrigeration effect (VRE). This is the refrigeration effect per mole of the refrigerants. It is expressed in kw/mol or kj/hrmol. The higher the volumetric refrigeration, the higher the performance and vice versa(Garbon& Roger 2011)

VRE = heat absorbed by evaporator (kW) / number of mole of refrigerant (mol)

There are many factors that affect the performance of a vapour compression refrigeration system. They include condensation temperature, evaporation temperature, sub cooling, superheating. Eigendy (2013) carried out a research to investigate the effect of replacing the valve with an ejector in the vapour compression refrigeration system in an attempt to improve the cycle performance. Condensation temperature, evaporation temperature, nozzle size, subcooling, and superheating were parameters investigated. Analysis of the result revealed that condensation temperature, evaporation temperature, subcooling and superheating has the performance ratio effect of 129%, 84 %. 14% and 4.3% respectivelyThis means that condensation temperature has the greatest essence in the design of vapour refrigeration systems, while superheating has the least. Ashish et al (2013) carried out a research to investigate the thermodynamic performance of three different refrigerants. Performance parameters such as coefficient of performance (COP), volumetric refrigeration effect (VRE), and refrigeration effect sub cooling and superheating were used. Condensation temperature, evaporation temperatures subcooling and superheating were some of the system parameters used. Analyses of the results show that increase in condensation temperature lowers the cycle performance while reduction in the condensation temperature increases the cycle performance, increased evaporation temperature increases system performance while reduced condensation temperature decreases system performance, increase in degree of sub cooling increases the cycle performance while reduction in the degree of sub cooling decreases the cycle performance, while increase in degree of superheating increases the cycle performance while reduction in the degree of superheating decreases the cycle performance.

Ahmed et al (2015) in a separate research carried out to investigate the effect of mixed refrigerant of different proportion using propane (R22) and isobutene (R66a). Coefficient of performance, refrigeration effect, and volumetric cooling capacity are performance parameters while condensation temperature, evaporation temperature were system parameters. Analyses of the result confirm that increased evaporation temperature increases system performance while reduced condensation temperature decreases system performance. Austin .N, Senthel Kumar P, Kenthavelkumaran N. (2012) carried out a research to determine the best refrigerant for house hold refrigerators. They used condensation temperature for system parameter and cooling effect and coefficient of performance as performance parameter. They confirmed that condensation temperature has s great effect on the performance of a vapour compression refrigeration system. They also concluded that increased condensation temperature lowers the performance of refrigeration cycle while lower condensation temperature increases the performance of vapour compression refrigeration system.

Kharagpur (2008) in his textbook "Refrigeration and Air conditioning" stated that useful superheating increase the refrigeration effect as well as the compressor work therefore the performance of the refrigeration cycle may or may not be increased with superheating depending on the nature of the working fluid. He also mentioned that even though useful super heating may or may not increase the performance of vapour refrigeration cycle a minimum level of superheating is desirable to avoid the entry of liquid into the compressor. He illustrated the effect using a T-S diagram as shown in figure 2

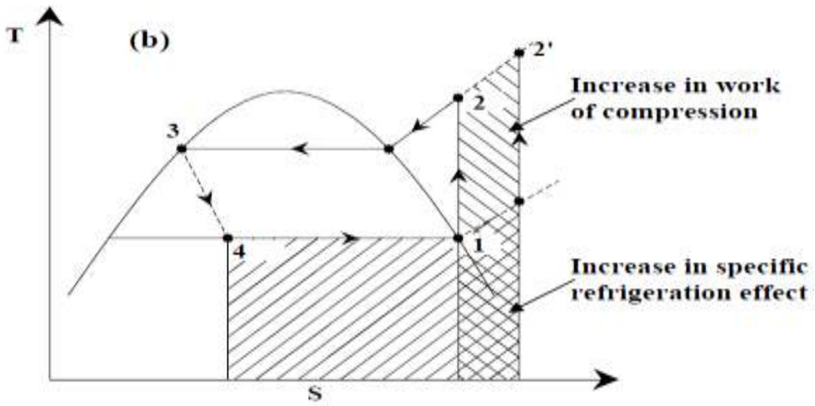

Figure 2 Comparison of refrigeration cycles with and without superheating in a T-S diagram

Statement of problem

Under ideal and optimum operation of the condenser unit of the vapor compression refrigeration cycle, the refrigerant fluid always enters the condenser as vapor but leaves as complete liquid (fully condensed). But in real operational condition, the outlet fluid leaves with some vapor fraction (partially condensed). This condition of incomplete condensation is as a result of many factors ranging from handling problem to the problem of heat transfer situation of corrosion, fouling effect etc.

Theoretically, the temperature/entropy diagram of a closed loop vapor compression refrigeration system show that incomplete condensation affects the overall performance of the cycle negatively ranging from increase in the power required to run the compressor, to the reduction of the cooling effect on the natural gas which is capable of posing an operational and economic problem to the refrigeration operation business. But the pattern or nature of these effects are not known with respect to the sizes of the condenser and the condensation conditions (temperature or pressure). Hence we are posed with these two main questions

1. What is the relationship between incomplete condensation and the performance of a closed loop vapour compression refrigeration cycle at different condenser size and condensation conditions?
2. Is there a particular degree of incomplete condensation of the refrigerant fluid at which a closed loop vapour compression refrigeration cycle will fail at different condensation condition and does this point dependent on the sizes of the condenser?

Nevertheless, the research below was designed to answer the questions comprehensively and explicitly because it is extremely important to understand the factors that affect condensation process in a condenser unit and also the relationship between condenser effectiveness/incomplete condensation and performance of the refrigeration system so as to run refrigeration systems optimally and effectively. And also to help people in LNG production business such as NLNG in specifying the appropriate unit operation (compressor) based on condition of the condenser.

Aim and objectives of the research

The principal aim of this research is to determine the effect of condenser fluid quality or incomplete condensation on the performance of vapor compression refrigeration system using a simulated simple propane pre cooled vapour compression refrigeration loop. The major objectives of the study are;

1. To determine the relationship between incomplete condensation and the performance of a vapor compressed refrigerant cycle using ASPEN HYSYS version 7.1 simulation software and three performance parameters
2. To determine the maximum level of incomplete condensation at which a refrigeration cycle fail at different condensation temperature and find out if this said level of incomplete condensation at which the refrigeration cycle fail is dependent on the condenser capacities.

Scope of the research

The research work will be limited to a simulated simple propane precooled loop of the propane precooled mixed refrigeration cycle (C3MR) design using plant data collected from an LNG facility in Nigeria. . This study is also limited to the results of the flash calculation of the simulation software used (ASPEN HYSYS VERSION 7.1)

II. MATERIALS AND METHOD

Data required for the simulation like inlet gas composition, inlet gas conditions (temperature, pressure, flow rate etc) condenser pressure drop, LNG heat exchanger pressure drop, valve opening, were obtained from operation manual of an LNG facility in Nigeria.Other necessary simulation data were sourced from ASPEN HYSYS training manual (2011)

The main assumptions of the simulation are

1. The system is operating on a steady state condition
2. The vapor is saturated at inlet and outlet of the compressor
3. Vaporization in the evaporator is at constant temperature
4. There are no pressure drops along the connections joining the units
5. The adiabatic and polytrophic efficiency of the compressor are constant.

ASPEN HYSYS simulation software version 7.1 was used to simulate three different propane pre cooled refrigeration loop with condenser capacities of 300mj/hr, 500mj/hr and 700mj/hr under Peng-robinsons equation of state. On each of the loops, the refrigerant vapor fraction at the condensers outlet is varied at interval of 0.05 at different condensation temperatures of 0℃ to 60℃ at interval of 20 and the corresponding compressor duties and pre cooled gas temperatures recorded

Microsoft excel will be used to plot the graph of compressor duty and cooling effect against vapor fraction at condenser heat exchange duty of 300mj/hr 500mj/hr and 700mj/hr on different graph at different outlet stream temperatures Also Microsoft excel will be used to plot the graph of compressor duty and cooling effect on the same graph and axis against the vapor fraction at different outlet stream temperatures for condenser heat exchange capacity of 300mj/hr 500mj/hr and 700mj/hr

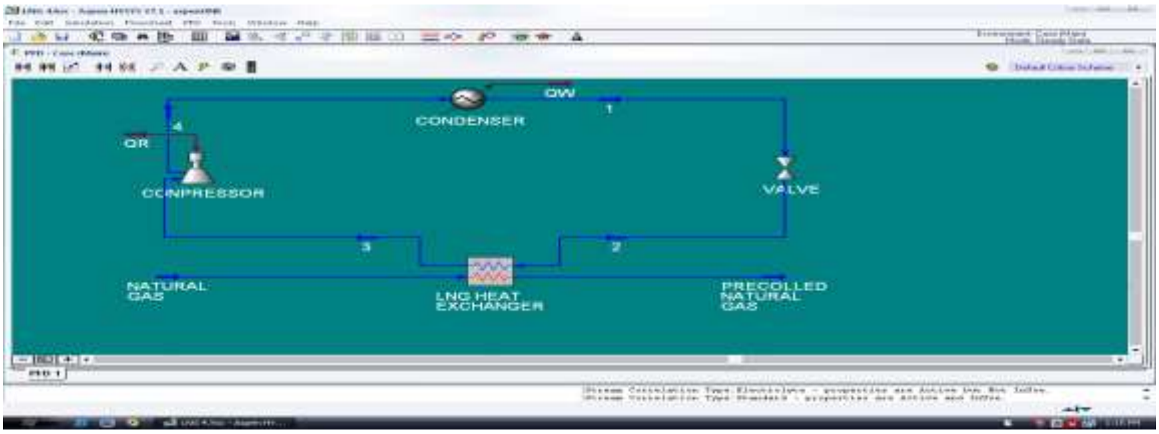

Figure 3 HYSYS simulation environment showing a converged window of complete propane pre cooled loop.

III. RESULTS AND DISCUSSION

3.1 Discussion of the relationship between compressor duty and vapor fraction at different condensation temperatures and condenser capacities

Figure 4-7 show the result of the simulation as regards to compressor duty and vapour fraction.

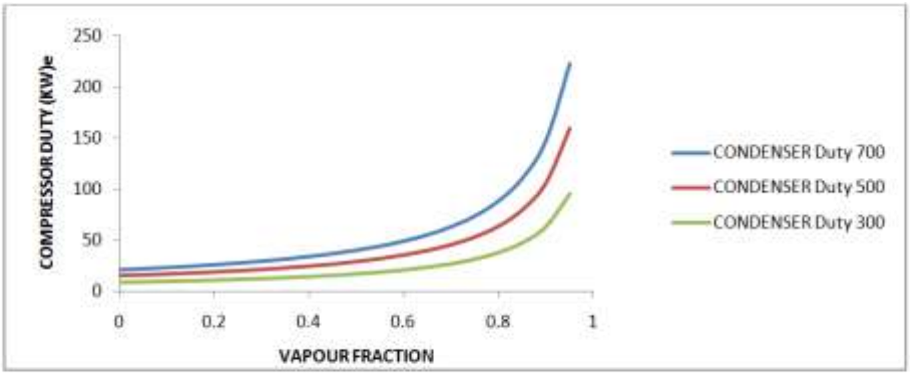

Figure 4 Graph of compressor duty against vapor fraction at temperature 0℃ and condenser duties of 300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 0℃, the compressor duties tend to be constant when the fluid vapor fraction increases linearly from 0.00 to 0.20. The compressor duties then increase linearly with the corresponding linear increase in the fluid vapor fractions within the range of 0.20 and 0.60and increase geometrically with linear increase in fluid vapor fractions from range 0.60 to 0.95. The loop finally fails due to heat flow based temperature cross between vapor fraction somewhere around 0.9 and 0.95. This behavior is similar to the entire three condensers regardless of their sizes or capacities.

Operationally, the above trend means that at condensation temperature of 0℃, the cycle performance is not affected by the drop in the fluid quality within the fluid quality range of 100% to 80%. The performance reduces gradually with the correspondinmg gradual drop in the fluid quality within the fluid quality range of 80% to 40%. The cycle performance then reduces rapidly with a gradual drop in the fluid quality within the range of 40% to 5%. Finally the loop fails due to reversed heat flow in the heat exchanger somewhere within the fluid quality range of 10% to 5%. Moreover this trend is independent on the size and capacity of the condensers. See figure 4.

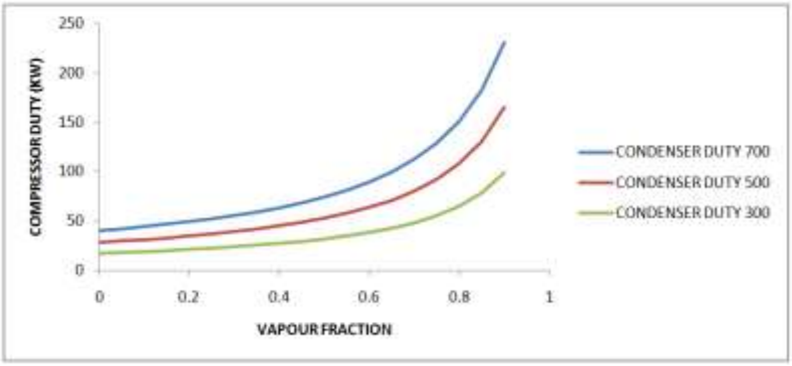

Figure 5;Graph of compressor duty against vapor fraction at temperature 20℃ and condenser duties of 300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 20℃, the compressor duties increase linearly with the corresponding linear increase in vapor fraction within the vapor fraction range of 0.00 to 0.70. The compressor duties then increase geometrically with the linear increase in the vapor fractions within vapor fraction range of 0.70 to 0.85. The loop finally fails due to heat flow based temperature cross at vapor fractions range somewhere between 0.85 and 0.90.

These operationally imply that at condensation temperature of 20℃, the cycle performances reduce linearly with the fluid quality within the range of 100% to 30%. The cycle performances then reduce geometrically with the linear decrease in the fluid qualities within the quality range of 30% and 15% and then the loop finally fails due to reversed heat flow in the heat exchanger between the refrigerant and natural gas when fluid quality drop somewhere between 15% and 10%. However, the trend is still independent on the sizes or the capacitiesof the condensers. See figure 5.

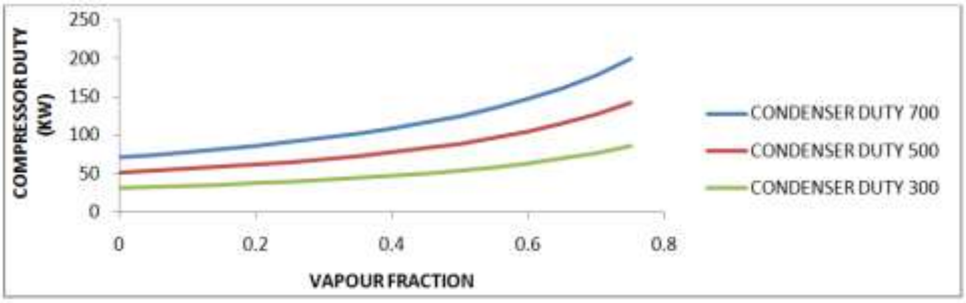

Figure 6;Graph of compressor duty against vapor fraction at temperature 40℃ and condenser duties of 300mj/hr, 500mj/hr and 700mj/hr

At the condensation temperature of 40℃, the compressor duty increase linearly with the corresponding increase in the fluid vapor fraction within the vapor fraction range of 0.00 to 0.61. The compressor duty then increase geometrically with linear increase in the vapor fraction within vapor fraction range of 0.61 to 0.78. The loop finally fails due to heat flow based temperature cross in the heat exchanger at vapor fraction somewhere between 0.78 and 0.80Physically these mean that within the condenser fluid quality of 100% to 39%, the cycle performance reduces linearly with the corresponding linear drop in the fluid quality. The cycle performance then reduces geometrically with linear drop in fluid quality within the fluid quality range of 39% to 19%. The loop finally fails due to reversed heat flow between refrigerant and natural gas in the heat exchanger at fluid quality somewhere between 22% and 20%. The trend above is still not dependent on the sizes or capacities of the condensers. See figure 6.

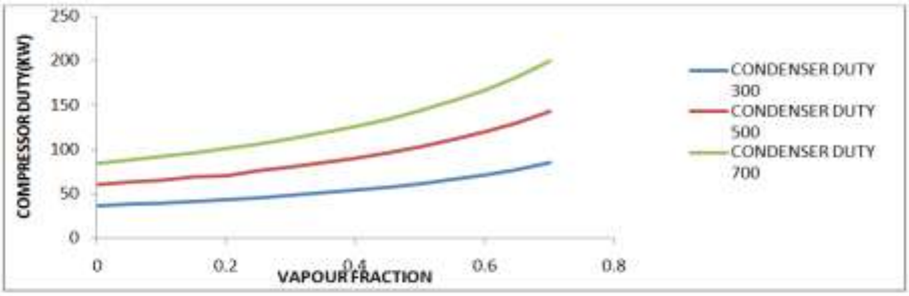

Figure 7;Graph of compressor duty against vapor fraction at temperature 60℃ and condenser duties of 300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 60℃, the compressor duty increase linearly with corresponding linear increase in the fluid vapor fraction within the vapor fraction range of 0.00 to 0.67. The loop then fails due to heat flow based temperature cross of the heat exchanger at vapor fraction somewhere between 0.67 and 0.69.
Physically these trends above imply that at condensation temperature of 60℃, the cycle performance reduces linearly with the corresponding linear drop in the fluid quality within the fluid quality range of 100% to 33%.the loop fails due to reversed heat flow between refrigerant and natural gas in the heat exchanger at fluid quality somewhere between 33% and 31%. Still the trends above are independent on the sizes and capacities of the condensers. See figure 7.

3.2 Discussion omf the relationship between cooling effect and vapor fraction at different condensation temperatures and condenser capacities

Figure 8-11 show the result of the simulation as regards to cooling effect and vapour fraction.

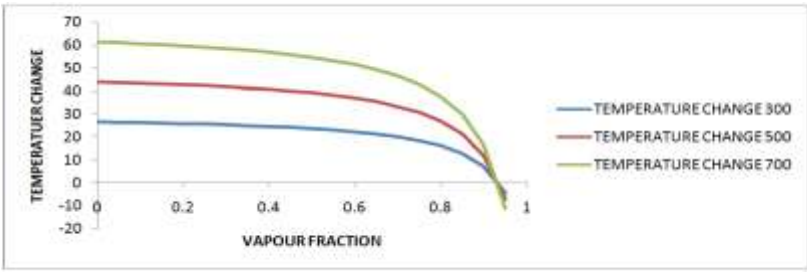

Figure 8; Graph of temperature change against vapor fraction at temperature 0℃ and condenser duties of300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 0℃, the temperature difference between the inlet natural gas and the precooled gas is relatively constant within the condenser outlet fluid vapor fraction range of 0.00 to 0.20. The temperature difference then reduces gradually and linearly with the corresponding linear increase in the vapor fraction within the range of 0.20 to 0.60. The temperature difference finally reduces geometrically from vapor fraction of0.60 to 0.93. At vapor fraction of 0.93, the temperature difference equals zero. Beyond 0.93, the temperature difference is negative and the loop fail due to heat flow based temperature cross.

Operationally, the analysis above imply that condensation temperature of 0℃,the cooling or refrigeration effect on the natural gas is not affected by the drop in the refrigerant fluid quality within the fluid quality range of 100% to 80%. The cooling effect then reduce s slowly and linearly with corresponding gradual reduction in the refrigerant fluid quality within the fluid quality range of 80% and 40%. The cooling effect further reduces geometrically with linear drop in fluid quality from range 40% to 8%. At fluid quality of 7%, there is no cooling effect on the natural gas due to thermal equilibrium between the natural gas and the refrigerant. Below 7% fluid quality, the loop fails due to reversed heat flow between the natural gas and the refrigerant. These phenomena are completely independent on the sizes or capacities of the condensers. See figure 8..

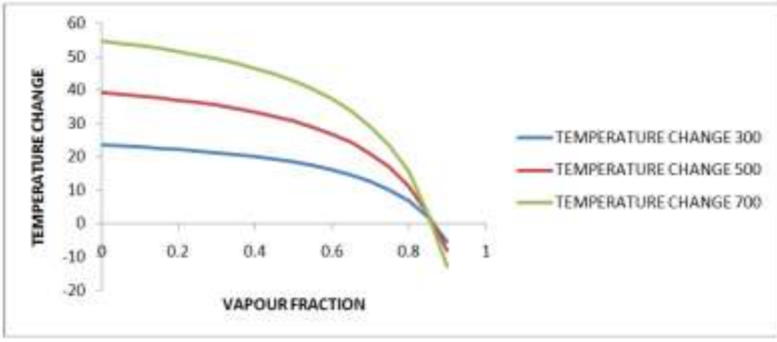

Figure 9;Graph of temperature change against vapor fraction at temperature 20℃ and condenser duties of 300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 20℃**,** the temperature difference between the inlet natural gas and the precooled gas decrease linearly with corresponding linear increase in the condenser refrigerant fluid vapor fraction within the vapor fraction range of 0.00 to 0.74. The temperature difference further decrease geometrically with linear increase in the fluid vapor fraction within the fluid vapor fraction range of 0.74 to 0.85.Atvapor fraction of 0.86, the temperature difference equal zero. Beyond the vapor fraction of 0.86, the temperature difference enter negative region and the loop fail due to heat flow based temperature cross.

Operationally these analyses above imply that at condensation temperature of 20℃**,** the cooling effect on the natural gas reduces gradually with ma corresponding gradual drop in the refrigerant fluid quality within the fluid quality range of 100% to 34%. The cooling effect then further decreases spontaneously with gradual drop in the fluid quality within the fluid quality range of 34$ to 15%. At fluid quality of 14%, there is no cooling effect on the natural gas due to thermal equilibrium that exists between natural gas and the refrigerant. Below fluid quality of 14%, the natural gas experience heating effect hence the loop fail due to reversed heat flow between the natural gas and the refrigerant. The processes above are still not dependent on the sizes or capacities of the condensers. See figure 9.

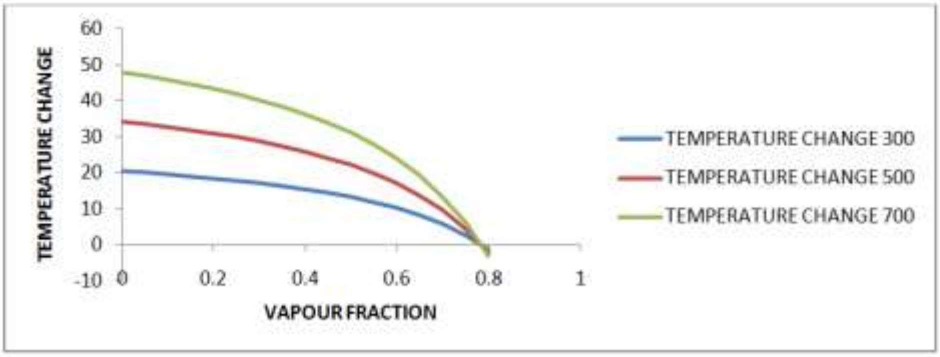

Figure 10; Graph of temperature change and vapor fraction at temperature 40℃ and condenser duties of 300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 40℃, the temperature difference between the inlet natural gas and outlet percooled gas decreases linearly with corresponding linear increase in the condenser refrigerant fluid vapor fraction within the vapor fraction range of 0.00 to 0.61. The temperature difference further decreases geometrically with a linear increase in the fluid vapor fraction within the vapor fraction range of 0.61 to 0.77. At vapor fraction of 0.78, the temperature difference equal zero. Beyond the vapor fraction of 0.78, the temperature difference enter negative region and the loop fail due to heat flow based temperature cross.

Operationally, these analyses above means that at condensation temperature of 40℃, the cooling effect on the natural gas drops gradually with corresponding gradual drop in the condenser outlet refrigerant fluid quality within the fluid quality range of 100% to 39%. The cooling effect then further drop spontaneously with gradual drop in the fluid quality within the fluid quality range of 39% to 23%. At fluid quality of 22%, there is no cooling effect on the natural gas due to thermal equilibrium that occurs between the natural gas and the refrigerant. Below 22%, the natural gas experience heating effect causing the loop to fail due to reversed heat flow between the natural gas and refrigerant. The processes above are still not dependent on the capacities or sizes of the condenser. See figure 10.

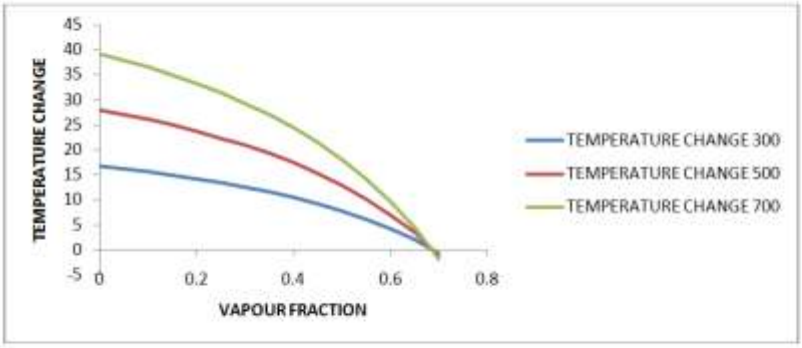

Figure 11;Graph of temperature change against vapor fraction at temperature 60℃ and condenser duties 300mj/hr, 500mj/hr and 700mj/hr

At condensation temperature of 60℃, the temperature difference between the inlet natural gas and outlet precooled gas decreases linearly with linear increase in the condenser outlet fluid vapor fraction within the vapor fraction range of 0.00 to 0.68. At vapor fraction of 0.69, the temperature difference equals zero and beyond 0.69, the temperature difference enter negative region and the loop fails due to heat flow based temperature cross.

Operationally these analyses imply that at condensation temperature 0f 60℃, the cooling effect on the natural gas decreases gradually with gradual drop in the condenser fluid quality within the fluid quality range of 100% to 32%. At fluid quality of 31%, there is no cooling effect on the gas below fluid quality of 31%, the gas experiences a heating effect causing the loop to fail cue to reversed heat flow between the natural gas and the refrigerant. These analyses are applicable to all sizes of condenser. See fig 11.

3.3 Discussion of the relationship between the ratio of compressor duty and cooling effect and vapour fraction

For simplicity and clearer analysis in this section, the following terminologies are first explained

COOLING EFFECTIVE REGION (C.E.R), This is the region in the graph of relationship between the temperature change, compressor duty and vapor fraction within which the ratio of temperature change and compressor duty is greater than one

At C.E.R, TEMPERATURE CHANGE / COMPRESSOR DUTY > 1

In this region, the temperature change line is above the compressor ratio line; this is the region on which the vapor compression refrigeration system is operating effectively and optimally. Hence, within this region of vapor fraction, to every one unit work input to the system, there is more than one unit cooling effect on the natural gas

CRITICAL COOLING POINT (C.C.P) this is the point on the graph of relationship between temperature change, compressor duty and vapor fraction at which the ratio of temperature change and compressor duty equals one.

At C.C.P, TEMPERATURE CHANGE / COMPRESSOR DUTY = 1

It is the point that marks the end of cooling effective region and the beginning of cooling ineffective region. Hence at this vapor fraction point, to every unit work done on the system, there is a corresponding unit cooling effect on the natural gas

COOLING INEFFECTIVE REGION (C.I.R), This is the region on the graph of relationship between the temperature change, compressor duty and vapor fraction within which the ratio of temperature change and compressor duty is less than one

At C.I.R, TEMPERATURE CHANGE / COMPRESSOR DUTY < 1

On this region, the line of temperature change is below the line of compressor duty. Within this region of vapor fraction, there is cooling but the cooling effect does not correspond to the amount of work input to the system. Meaning that at this vapor fraction region, to every unit work done on the system, there is less than one unit cooling effect on the natural gas.

THERMAL TRANSITION POINT (T.T.P). This is the point on the graph of relationship between temperature change compressor duty and vapor fraction at which the ratio of temperature change and compressor duty equals zero

At T.T.P, TEMPERATURE CHANGE / COMPRESSOR DUTY = 0

At this point there is no temperature change between the inlet natural gas and the precooled gas. Hence the natural gas and the refrigerant are in the state of thermal equilibrium. This point marks the end of cooling region and the beginning of heating region on the graph. A vapor compression refrigeration system should not be allowed to operate at the vapor fraction corresponding to this point to avoid failure.

HEATING POINT (H.R). This is the region within which the ratio of temperature change and compressor is negative

At H.R, TEMPERATURE CHANGE / COMPRESSOR DUTY < 0

At this region the natural gas is at lower heat energy potential hence serving as the refrigerant while the refrigerant is at higher heat energy potential hence serving as the cooled gas.The proper analysis of the combined relationship between temperature change compressor duty and vapor fraction are given as follows

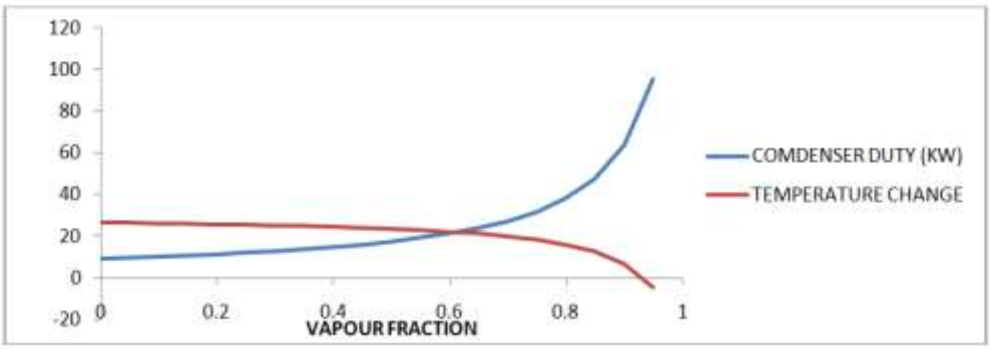

Figure 12;Graph of temperature change and compressor duty against vapors fraction at condenser temperature of 0℃ and condenser capacity of 500mj/hr

At condensation temperature of 0℃, the line of temperature change is above the line of compressor duty within the vapor fraction range of 0.00 to 0.61. The two lines intersect at vapors fraction of 0.62. From vapors fraction range of 0.63 to 0.92, the temperature change line is below the compressor duty line. At vapors fraction of 0.93, the temperature line hit the horizontal axis. Beyond vapors fraction of 0.93, the line of temperature change enter negative region.

Operationally, these trends above imply that at condensation temperature of 0℃ , cooling effective region prevailed within the fluid quality range of 100% to 39%. Fluid quality of 38% is the critical cooling point at condenser outlet fluid temperature of 0℃, within the fluid quality range of 38% to 8%, cooling ineffective region prevailed. Fluid quality of 7% is the thermal transition point at the condenser outlet fluid temperature of 0℃. Below the fluid quality of 7%, the system enter heating region which means the total failure of the system. See figure 12.

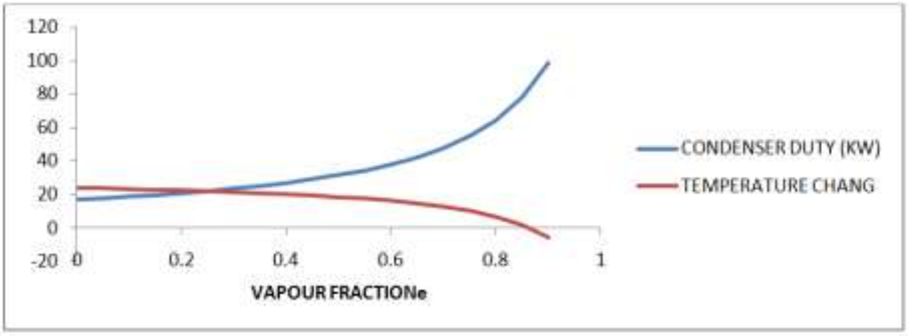

Figure 13;Graph of temperature change andcompressor duty against vapor fraction at condenser temperature of 20℃ and condenser capacity of 500mj/hr

At the condensation temperature of 20℃, the line of temperature change is above the line of compressor duty within the vapors fraction range of 0.00 t0 0.23. The two lines intersect at vapors fraction point 0.24. Fromvapors fraction of 0.24 to 0.85, the line of temperature change is below the line of compressor duty. At vapors fraction 0.86, the temperature change line meets the horizontal axis. Beyond the vapors fraction of 0.86, temperature change enter the negative region

Operationally the above trends imply that at condensation temperature of 20℃, cooling effective region prevailed within the fluid quality range of 100% to 77%. Fluid quality of 76% is the critical cooling point corresponding to condenser temperature of 20℃. Within the fluid quality range of 76% to 15%, cooling ineffective region prevailed. At fluid quality of 14%, thermal transition occurs and below this fluid quality, the systems enter into heating region hence fails. See figure 13.

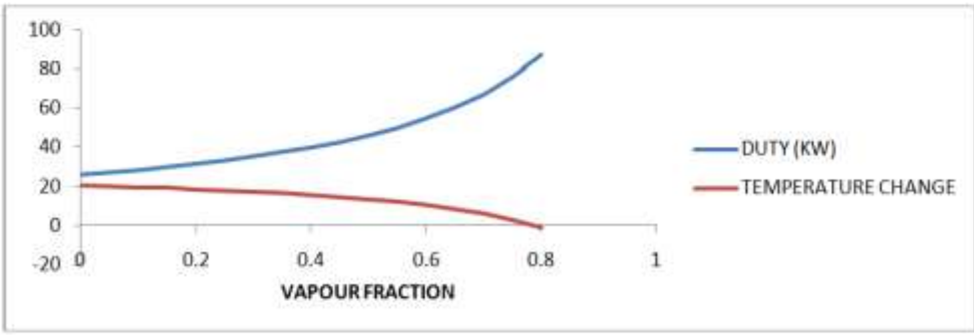

Figure 14;Graph of temperature change and compressor duty against vapor fraction at condenser temperature of 40℃ and condenser capacity of 500mj/hr

At condensation temperature 40℃, the line of temperature change is below the line of compressor duty within the vapors fraction range of 0.00 to 0.78. At vapors fraction of 0.79, the temperature change line meets the horizontal axis. Beyond 0.79, the temperature change line enters the negative region.Physically these trends imply that at condensation temperature of 40℃, cooling effective region does not exist. Within the fluid quality range of 100% to 22%, cooling ineffective region prevailed. Fluid quality of 21% is the thermal transition point corresponding to the condenser temperature of 40℃. Below the fluid quality of 21%, the system enter heating region hence failure occur

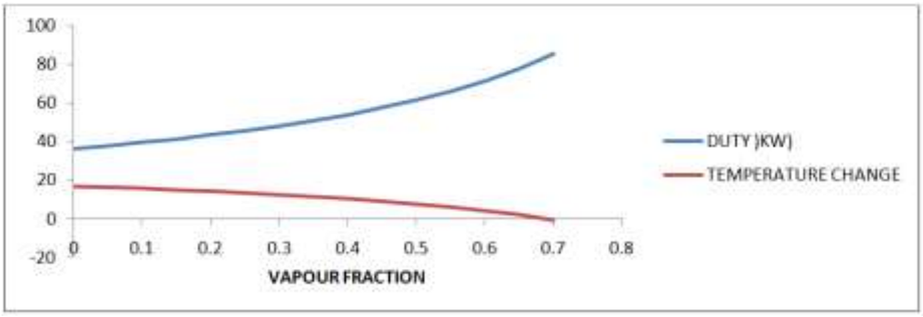

Figure 15;Graph of temperature change and compressor duty against vapors fraction at condenser temperature of 60℃ and condenser capacity of 500mj/hr

At condensation temperature of 60℃, the temperature change line is below the compressor duty line at all the vapor fraction interval specifically within the vapors fraction range of 0.00 to 0.67. The temperature change line meets the horizontal axis at vapors fraction point of 0.68. Beyond the vapors fraction point of 0.68, the temperature change line enters the negative region.

Physically the trends above signify that at condensation temperature of 60℃, cooling effective region does not exist. Within the fluid quality range of 100% to 33%, cooling ineffective region prevailed. At fluid quality of 32%, thermal transition point occurs. Below the fluid quality of 32% heating region occur and the system fails. See figure 15.

3.4 Discussion of the relationship between thermal transition point and condensation temperature

There is a direct linear relationship between thermal transition point and condensation temperature with percentage deviation of 0.7% as shown in figure 15. This is because as condensation temperature increase there is a corresponding decrease in the performance of the cycle and as such an increase in the fluid quality at which there is no cooling effect on the natural gas

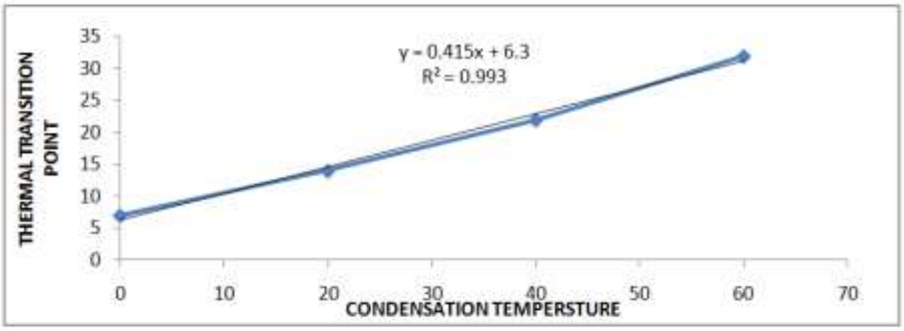

Figure 16;Graph of the thermal transition point (TTP) against condensation temperature at any condenser capacity

IV. CONCLUSIONS

A simulation model analysis has been carried out to investigate the effect of incomplete condensation /condensers fluid quality on the performance of a simple closed vapor compression refrigeration system using ASPEN HYSYS version 7.1 and plant data from an LNG facility in Nigeria the performance parameters investigated are compressor duty, cooling effect and the ratio of cooling effect and compressor duty. The result obtained showed that

1. increase in the level of incomplete condensation or lower condensers fluid quality result in a lower performance of the system, and the degree of effect is dependent on the condensation temperature but independent on the size or heat duty of the condenser.
2. The fluid quality point at which the cycles fail (thermal transition point) is dependent on the condensation temperature and independent on the condenser capacity. The higher the condensation temperature the higher the fluid quality point at which the cycle fail.
3. Three regions and two point were identified they includes cooling effective region (CER), cooling ineffective region (CIR), heating region (HR) thermal transition point (TTP) and critical cooling point (CCP) the size of these regions and positions of these points are dependent on the condensation temperature but independent on the condensers capacity.

REFERENCES

[1] Ahmed J.H. Fawziea M,H, Sura S,Y2015,"Comparative Study of Auto Cascade Refrigeration System Performance Using Alternative Mixed Refrigerant" International journal of Energy Research and Industrial Applications"(IJERIA) ISSN 0974-1518 vol.8 pp(111-126)
[2] AjeetS.S, Devendra .D, Surendra K. A. 2013"Performance Analysis of Surface Condenser under varied operating parameter" .International Journal of Engineering Research and Applications (IJERA) pp 416-421
[3] Arora C.P, 2010,"Refrigeration and Air Conditional" 3rd Edition, Tata McGraw
[4] Hill New Dehli.
[5] Ashish, K. P, Gupta R .C, 2013, "Effect of Subcooling and Superheating on Vapour Compression Refrigeration System Using R22, Alternative Refrigerant" International Journal of Emerging Trend in Engineering and Development. Vol 1 pp521-530
[6] Austin .N, Senthel Kumar P, Kenthavelkumaran N. 2012, "Thermodynamic Optimization of Household Refrigerators Using Propane-Butane as Mixed Refrigerant" .International Journal of Engineering and Research Applications (IJERA)VOL 2 PP 268-271
[7] Aspen HYSYS Training manual version 7.1 2011.
[8] Elgendy E, 2013,"Parametric Study of a Vapour Compression Refrigeration Cycle Using a Two-Phase Constant Area Ejector" International Journal of Mechanical Aerospace, Industrial, Mechatronics and Manufacturing Engineering Vol. 7 pp 565-571.
[9] Garbon, R. Mayhew Y. 2011, "Engineering Thermodynamics Work and Heat Transfer".4thEdition.Dorling Kinderley India pub.Ltd pp416-418
[10] Kharagpur IIT 2008, "Refrigeration And Air Condition.40 Lessons on Refrigeration and Air Condition, useful Training Material For Mechanical Engineering Student/College Or A Reference For Engineer". India.Version 1 ME.
[11] Michael .T.N, Howard S.N, 2000, "Fundamental of Engineering Thermodynamics" 3rd Edition John Wiley & Sons inc.pp512-515
[12] Nigeria LNG Operation Manual unit 1400 Liquefaction (2000)
[13] Rajput R K. 2006, A Text Book of Engineering Thermodynamics. Laxmi Publication LTD pp736-737