STUDY THE EFFECT OF WORKING MASS FLOW RATE ON THE DISTILLATOR PERFORMANCE IN DOUBLE – CYCLE DISTILLATION HEAT PUMP SYSTEM

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Abstract:

The performance of a distillator in a double – cycle distillation heat pump system, utilizing water as a working fluid in finned tube heat exchangers for evaporator and condenser, has been investigated. A mathematical model was set for the whole system and a simulating program was developed to simulate the double – cycle processes during the steady – state operation. This subroutine program to calculate the water and steam properties in the range of $(65 - 115)^{o}C$ and $(0.25 - 1.6906)$bar.

The experimental results showed good agreement with the results obtained from the computer program. The concentration on the distillator performance during steady – state operation of the system is occurred, and it was found that the distillator performance ($COP_{dist.}$) increased when the working mass flow rate increased and also it was increased when the working fluid pressure increased and had a maximum value (83.3%) at working mass flow rate of 5.5 kg/hr, working fluid pressure of 0.8 bar and process fluid pressure of 1.01825 bar.

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دراسة تاثير معدل التدفق على اداء التقطير في المضخات

مهدي هاتف كاظم

الكلية التقنية نجف

الخلاصة:

تم دراسة أداء مقطر في مضخة حرارية ثنائية الدورة باستخدام الماء كمائع شغل في مبادلات حرارية ذات أنابيب مزعنفة لكل من المبخر و المكثف كمنظومة تقطير. تم وضع الموديل الرياضي لكلا الدورتين، ثم تم بناء برنامج حاسوبي لتمثيل عمليات المنظومة خلال عملها في الحالة المستقرة. هذا البرنامج الفرعي يعمل ضمن مدى درجة حرارة $(115 - 65)$م^{o} و ضغط $(1.6906 - 0.25)$ جو. النتائج العملية أظهرت توافقا جيدا مع النتائج المستحصلة من البرنامج الحاسوبي. تم التركيز على أداء المقطر خلال عمل المنظومة في الحالة

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المستقرة ووجد إن معامل أداء المقطر يزداد عندما يزداد معدل كتلة جريان مائع الشغل و كذلك يزداد عندما

يزداد ضغط مائع الشغل و يصل إلى أعلى قيمة له(83.3%) عند كتلة جريان مائع الشغل (5.5)كغم/ساعة،

ضغط مائع الشغل (0.8) جو و ضغط المائع الثانوي (1.01825) جو.

<u>Nomenclature</u>

Symbol	Description	Units
A_b	Bare tube outside surface area	m^2
A_f	Fin surface area	m^2
d	Pipe diameter	m
D	Depth	m
H	Enthalpy	J/kg
F	Convective boiling factor	–
h	Heat transfer coefficient	$W/m^2.k$
L_i	Distance along the evaporator from its entrance	m
Pr	Prandtle number	–
Re	Reynolds number	–
S	Suppression factor	W
t	Thickness	m
Wi	Width	m
W	work	W
X	quality	–
$X_{i,j}$	Martinelli parameter where i , j refer to the nature of flow of the liquid and vapor phases	
Greek symbols		
α_{pf}	Ratio of the total heat transfer area to the total volume of the heat exchanger	
α_v	Void fraction $= A_v / A_{tot}$	
β	Ratio of fin surface area to the total outside	
σ	Surface tension	N/m
$\Phi_{I,J}^2$	Ratio of the two – phase pressure gradient to the pressure gradient for single – phase liquid (or vapor) for that phase flowing alone in the pipe	

subscript		
a	acceleration	
av	Average	
b	Boiling	
con.	Condensate	
dist.	Distillator	
f	Fin	
ft	Flat tube	
HE	Heat exchanger	
i	Element step	
L	Liquid – phase	
L	Low	
1p	Single – phase	
2p	Two – phase	
pf	Process fluid	
sat.	Saturation	
v	Vapor – phase	
w	Wall	
wf	Working fluid	

Introduction:

In recent years, population growth and its effect on industrial and agricultural requirements, has made the enhancement of fresh water resources a vital endeavor, as the demand for steady, economical supply of water is constantly increasing all around the world. Many solutions to the water problem have been proposed, such as, control of water consumption, improved distribution, storage and desalination (AL – Atrooshi 1999)

Desalination

Desalination describes a range of processes which are used to reduce the amount of dissolved solids in water. As a means of producing potable water, desalination is usually an expensive option. It is often associated with electricity generation plants, from which both

electricity and waste heat are available (Smith and Shaw 2002)

Desalination by Distillation

Distillation is the oldest and most commonly used method of desalination. In distillation process the saline water is heated to produce water vapor, which is then condensed to produce fresh water. The various distillation processes used to produce potable water, including multistage – flash (MSF), multiple effect (MED), vapor compression (VC), and waste – heat evaporators, all generally operate on the principle of reducing the vapor pressure of water within the unit to permit boiling to occur at lower temperatures, without use of additional heat.

Distillation is effective as a means for desalination because most of the chemical substances found in saline water are nonvolatile at boiling temperature, and hence remain in the unevaporated blow down liquids.

The Double (Twin) Cycle Distillation Heat Pump System

The double – cycle distillation system is a new method for producing potable water. One such system was presented at **the committee of technology transfer conference** (Committee of Technology 2001), by the name of **Zyclodest plant** at 2001. **Zyclodest plant** is a special mechanical vapor compression method. In it the working fluid exists in a separate loop for the heat recovery. Thus, the product vapor, secondary fluid, is only contact with heat – exchangers and does not pass through the compressor and its rotating elements. The schematic diagram of this heat pump can be envisioned as shown in the Fig. (1), and the double – cycle represented by the Pressure – Enthalpy diagram, as in Fig. (2).

According to Fig.s (1) and (2) the coefficient of performance of distillator (COP$_{dist.}$) as follows:

$$COP_{dist} = \frac{Q_L}{W_1 + W_2} = \frac{H_6 - H_7}{(H_4 - H_3) + (H_8 - H_7)} = \frac{H_3 - H_2}{(H_4 - H_3) + (H_8 - H_7)} \qquad ....(1)$$

The double – cycle distillation heat pump plant consists of two main cycles; the water compression – expansion cycle, which called also the compression – expansion vapor (CEV) cycle, and the process cycle, see Fig. (3). Each of these cycles is described as follows:

Compression – Expansion Vapor Cycle

The compression – expansion vapor cycle consists of compressor, expansion device, and two heat – exchangers with interconnecting piping. One of the heat – exchangers submerged in a pool boiling liquid and represents a condenser coil, which is mounted at an angle of inclination of about 15 degree with respect to the horizontal so as to allow for condensate drainage. The other heat – exchanger is exposed to a collective vapor and represent an evaporator coil. The evaporator and the condenser are mounted at the same angle to simplify the construction.

Process Cycle

This cycle consists of three main parts with interconnecting piping, namely: evaporation chamber, intermediate (IM) zone and condensation chamber. A brief description of this cycle is as follows:

In the evaporation chamber (boiler), the evaporator coil submerged in a pool of water and the heat will transfer through the evaporator walls to the water in the pool. Due to this heat transferred, water will boil and the amount of evaporative mass depends on the amount of heat transfer.

The vapor will rise due to the difference in density to pass through the intermediate   zone through corrugated channel. This part is used to avoid mist flow, and then the vapor enters the condensation chamber as a saturated vapor.

In the condensation chamber, the condenser coil is exposed to the saturated vapor. Due to the heat transferred to the working fluid inside the condenser through the walls of the condenser, the vapor will condensate on the outside walls of the condenser. The condensate will exit from the condensation chamber and return to the evaporation chamber by the distillate water pump, by gravity, or the pressure difference.

The heat exchangers used for evaporator and condenser of the double – loop distillation heat pump system are finned tube heat exchangers, as shown in Fig. (4).

Theory;

A mathematical model will be set for the two loops and a computational program will be developed to simulate the two loops during steady – state operation for knowing the effect of the working mass flow rate on the distillatior performance.

The analysis for the working and process fluid, the evaporator and the compressor during the steady – state operation of the system may describe as follows:

Physical Properties

Working and Process Fluid

The working and process fluids in both the CEV and process loops respectively are that of pure water. The properties of pure water as required for simulation can be divided into:

a. Thermodynamic Properties: which include (υ, i, s, Cp and Cv), obtained from (UK Steam Tables in SI Units 1970) .

b. Transport Properties: which include (ρ, K, μ and σ), obtained from (AL – Atrooshi 1999)

All the above thermodynamic and transport properties are calculated for subcooled and saturated liquid and saturated and superheated vapor using standard table correlations, in the temperature range of (65 – 115) ^{o}C.

For saline water, the thermodynamic and transport properties are direct functions of

temperature and degree of salinity and in some cases they are established based on comparison with the same properties for pure water. The essential empirical equations for these properties as a function of temperature and degree of salinity were obtained from (AL – Atrooshi 1999)

Walls and Insulation

The properties of the system materials and insulation are assumed constant. Also the thermal conductivity for insulation (glass wool), pipes (copper and wrought iron), Fins and tubes of the heat exchangers (aluminum), and system body (galvanized plats) are obtained from (ASHRAE handbook 1997) and (Holman 2008).

Heat Transfer Analysis

Modeling of Evaporator

Referring to Fig. (5), the heat transfer rate from the process fluid (vapor) to the working fluid (two – phase flow) in an element (i) in the evaporator is:

Q_i = convective heat transfer   = conduction heat transfer   =   convective heat transfer

from process fluid to the   through the wall of   from the wall of element

the wall of element (i)   element (i)   (i) to the working fluid

………(2)

$$= \frac{T_{pf} - T_{w,o}}{R_{pf}} \ = \ \frac{T_{w,o} - T_{w,in}}{R_w} \ = \ \frac{T_{w,in} - T_{wf}}{R_{wf}} \qquad\qquad ………(3)$$

$$= \frac{T_{pf} - T_{wf}}{R_{pf} + R_w + R_{wf}} \qquad\qquad ………(4)$$

And,

$$(UA)_o \ = \ \frac{1}{R_{wf} + R_w + R_{pf}} \qquad\qquad ………(5)$$

Outside Convective Thermal Resistance

Vapor (process fluid) – side convective thermal resistance is given as (Holman 2008).

$$R_{pf} \ = \ \frac{1}{A_{i,o} \ h_o \ \eta_{tot.}} \qquad\qquad ………(6)$$

Outside surface area of the heat exchanger is calculated for each element length as follows:

$$A_{i,o} \; = \; \alpha_{pf} \left(L_i \; Wi_{HE} \; D_{HE} \right) \qquad\qquad …….. (7)$$

Outside Heat Transfer Coefficient

The outside condensation heat transfer coefficient is calculated by the following correlation (AL – Atrooshi 1999)

For $Re_L < 1800$

$$h_o \; = \; 0.725 \left[\frac{\rho_L \left(\rho_L - \rho_v \right) g \, K_L^3 \, H_{fg}}{\mu_L \, d_h \left(T_v - T_w \right)} \right]^{\frac{1}{4}} \qquad\qquad …….. (8)$$

For $Re_L > 1800$

$$h_o \; = \; 0.725 \, B \left[\frac{\rho_L \left(\rho_L - \rho_v \right) g \, K_L^3 H_{fg}}{\mu_L \, d_h \left(T_v - T_w \right)} \right]^{\frac{1}{4}} \qquad\qquad …….. (9)$$

Where,

$$B \; = \; 1.23795 + 0.0353808 \left(nr \right) - 0.00157035 \left(nr \right)^2 \qquad\qquad …….. (10)$$

nr : is the number of tube rows of the heat exchanger.

The condensate in an actual condensation process is cooled farther to some average temperature between saturation temperature ($T_{sat.}$) and wall temperature (T_w), releasing more heat in the process. Therefore, the actual heat transfer will be large, and the (h_{fg}) replacing by the modified latent heat of condensation (h_{fg}^*), which defined as follows(Yunus. and Michael 2007)

For saturated vapor at the evaporator outside,

$$H_{fg}^* \; = \; H_{fg} + 0.68 Cp_L \left(T_{sat.} - T_w \right) \qquad\qquad …….. (11)$$

For superheated vapor at the evaporator outside,

$$H_{fg}^* \; = \; H_{fg} + 0.68 \, Cp_L \left(T_{sat.} - T_w \right) + Cp_v \left(T_v - T_{sat.} \right) \qquad\qquad …….. (12)$$

Fin Analysis:

The total fin efficiency ($\eta_{tot.}$) is calculation from the following equation (Sarsam 1995)

$$\eta_{tot.} = \frac{\text{total heat transfer from tube and fin surface}}{\text{maximum heat transfer from tube and fin surface}} \qquad \text{……..(13)}$$

$$= \frac{h_{pf}\, A_b\,(T_{pf} - T_{o,w}) + h_{pf}\, A_f\, \eta_f\,(T_{pf} - T_{o,w})}{h_{pf}\,(A_b + A_f)(T_{pf} - T_{o,w})} \qquad \text{……..(14)}$$

$$= \frac{A_b + A_f\, \eta_f}{A_b + A_f} = 1 - \beta_{pf}\,(1 - \eta_f) \qquad \text{……..(15)}$$

For continuous fins, fin efficiency (η_f) is calculated using the following equation :

$$\eta_f = \frac{\tanh\,(m\, H_f)}{m\, H_f} \qquad \text{……..(16)}$$

Where,

$$m = \left[\frac{2\, h_o}{K_f\, t_f}\right]^{\frac{1}{2}} \qquad \text{……..(17)}$$

The fin efficiency (η_f) is multiplied by (2), because there are two fins protrude from the same point at the outer – surface of the flat tube of the evaporator, see **Figs. (4) and (5)**.

Conduction Thermal Resistance

The thermal resistance due to conduction is given as (as Holman 2008) shown in fig(4):

$$R_w = \frac{t_{ft}}{A_{av}\, K_{ft}} \qquad \text{……..(18)}$$

Where,

$$A_{av} = \frac{A_o - A_{in}}{\ln\left(A_o / A_{in}\right)} \qquad \text{……..(19)}$$

$$A_o = 2\,(L_{ft} + Wi_{ft})\,L_i \qquad \text{……..(20)}$$

$$A_{in} = 2\,\big[(L_{ft} - t_{ft}) + (Wi_{ft} - t_{ft})\big]\,L_i \qquad \text{……..(21)}$$

Inside Convective Thermal Resistance

The working fluid – side convective thermal resistance is given as:

$$R_{wf} = \frac{1}{h_{in}\, A_{in}} \qquad \text{……..(22)}$$

Since the working fluid inlet to the evaporator as two – phase flow, therefore, the two – phase heat transfer coefficient is calculated according to the correlation proposed by (Chen 1966) as follows:

$$h_{2p} = h_{con.} + h_b \qquad\qquad\qquad \text{……..(23)}$$

$$h_{con.} = 0.023 \left[\frac{G\,(1-X)\,d}{\mu_L}\right]^{0.8} Pr_L \frac{K_L}{d} F \qquad\qquad \text{……..(24)}$$

$$h_b = 0.00122 \left[\frac{K_L^{0.79}\,Cp_L^{0.45}\,\rho_L^{0.49}}{\sigma^{0.5}\,\mu_L^{0.29}\,H_{fg}^{0.24}\,\rho_v^{0.24}}\right] (T_w - T_{sat.})^{0.24}\,(P_w - P_{sat.})^{0.75}\,S \qquad \text{……..(25)}$$

Where,

$$F = 1 \qquad\qquad\qquad \text{when } \frac{1}{X_{t,t}} < 0.1$$

$$\qquad\qquad\qquad\qquad\qquad\qquad\qquad\qquad\qquad \text{……..(26)}$$

$$F = 2.35 \left(\frac{1}{X_{t,t}} + 0.213\right)^{0.736} \qquad \text{when } \frac{1}{X_{t,t}} > 0.1$$

$$S = \left(1 + 0.12\,Re_{2P}^{1.14}\right)^{-1} \qquad \text{for } Re_{2P} < 32.5$$
$$S = \left(1 + 0.42\,Re_{2P}^{0.78}\right)^{-1} \qquad \text{for } 32.5 < Re_{2P} < 70 \qquad \text{……..(27)}$$
$$S = 1 \qquad\qquad\qquad\qquad \text{for } Re_{2P} > 70$$

$$\frac{1}{X_{t,t}} = \left(\frac{X}{1-X}\right)^{0.9} \left(\frac{\rho_L}{\rho_v}\right)^{0.5} \left(\frac{\mu_v}{\mu_L}\right)^{0.1} \qquad\qquad \text{……..(28)}$$

$$Re_{2P} = \frac{G\,(1-X)\,d}{\mu_L}\,F^{1.25} \cdot 10^{-4} \qquad\qquad \text{……..(29)}$$

When the quality (X) of the two – phase working fluid reaches to unit, then single – phase (vapor) is occurred. Therefore may using the correlation proposed by (Sieder, and Tate 1936) to calculate the heat transfer coefficient (h_{in}), for laminar or turbulent fully developed hydrodynamic flow.

For laminar flow (Re < 2400),

$$h_{1p} = 1.24 \frac{K}{d_h} \left[Re\,Pr\,\frac{d_h}{L_i}\right]^{1/3} \left[\frac{\mu}{\mu_w}\right]^{0.14} \qquad\qquad \text{……..(30)}$$

For turbulent flow (Re > 2400),

$$h_{1p} = 0.027 \frac{K}{d_h} Re^{0.8}\,Pr^{0.4} \left[\frac{\mu}{\mu_w}\right]^{0.14} \qquad\qquad \text{……..(31)}$$

Modeling of Compressor

Since, the working fluid always enters the compression chamber as vapor. Thus, ideal gas laws may be applied on it for relations of pressure – temperature and pressure – volume, (ASHRAE 2000)

The shaft work of the compressor with known mass flow rate may calculated as follows:

$$\text{W} = \frac{\gamma}{\gamma - 1} \, \text{P}_1 \, \dot{v}_1 \left[\left(\frac{\text{P}_2}{\text{P}_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] \qquad\qquad \text{……..(32)}$$

When the compression of the gas occurred with existence of losses (poletrophic compression process), thus, the specific heat ratio (γ) equal to the power (n).

Hydrodynamic Analysis

Working Fluid – Side

For single – phase flow, the frictional pressure drop is calculated from the following relation (Franzini. and Finnemore 1997)

$$\Delta \text{P}_{\text{frc,1p}} = 4 \, f \, \frac{\text{L}_i}{\text{d}_h} \frac{\text{G}^2}{2 \, \rho} \qquad\qquad \text{……..(33)}$$

Where, for single – phase laminar flow

$$f = \frac{16}{\text{Re}} \qquad\qquad \text{……..(34)}$$

And, for single – phase turbulent flow

$$f = \frac{0.079}{\text{Re}^{0.25}} \qquad\qquad \text{……..(35)}$$

For two – phase flow, the correlations of Lockhart and Martinelli are used to relate the two – phase frictional pressure drop to the single – phase liquid frictional pressure drop, evaluated using equations (33) to (35), assuming that the liquid – phase alone is occupied the tubes. In addition, their empirical correlation for the void fraction (αv) in the two – phase flow region is used (Sarsam 1995)

$$\Delta \text{P}_{\text{frc,2p}} = \Phi^2_{i,j} \, \Delta \text{P}_{\text{frc,1p}} \qquad\qquad \text{……..(36)}$$

And,

$$\alpha_v = \left(1 + \text{X}^{0.8}_{i,j} \right)^{-0.378} \qquad\qquad \text{……..(37)}$$

Subscripts i and j represents the nature of the flow for the liquid and vapor phases

respectively. Four such conditions may obtain from (Sarsam 1995)

The appropriate relation for Xi,j is used to calculate the void fraction and Φi,j , which in turn is used to determine the two – phase pressure drop.

The hydrostatic pressure drop for the working fluid is calculated from:

$$\Delta P_h = \rho_m \, g \, L_i \, \sin(\theta)$$ ……..(38)

Where,

ρm : is the mean density of the mixture given by the following equation:

$$\rho_m = \alpha_v \, \rho_v + (1 - \alpha_v)\rho_L$$ ……..(39)

And,

θ : is the angle of inclination of the heat exchanger.

The acceleration pressure drop is calculated from the following correlation (Sarsam 1995)

$$\Delta P_a = \frac{G^2}{\rho_L}\left[\frac{(1-X)^2}{1-\alpha_v} + \frac{X^2}{\alpha_v}\frac{\rho_L}{\rho_v} - 1\right]$$ ……..(40)

Further more, the pressure losses occurring at the entrance and exit of the heat exchanger are evaluated using the relations proposed by,(Kays and London1964)

$$\Delta P_{ent.} = \frac{G^2}{2\rho}\left(1 - \sigma^2\right)$$ ……..(41)

$$\Delta P_{exit} = \frac{G^2}{2\rho}\left(\sigma^2 - 1\right)$$ ……..(42)

Where,

σ : is the ratio of free flow area to frontal area.

Scope Of Experiment

This work performed for investigation of the distillation performance process in the heat pump system with varying the distillation conditions.

Experimental Setup

The double – cycle distillation heat pump system for desalination, utilizing water as the working and process fluid, is designed and constructed in accordance with the schematic diagram in **Fig. (6).**

The two – loops of the heat pump are a sealed system charged with a known mass of working and process fluid and work under different pressures. The two – loops are insulated in

order to minimize heat flow to ambient. A pictorial view of the test rig is shown in **Fig. (7).**

Experimental program was initiated to studying the influence of mass flow rate of the CEV loop on the evaporator (distillator) performance.

The evacuating and charging the system are attained using two different points for CEV loop and four different points for thermosyphon loop, and a fifth point at the bottom of the liquid line is used to specify the level of the feed water in the evaporation chamber.

The heat exchangers used for the evaporator and condenser has an outside surface area of about (0.325 m^2).

The liquid, liquid – vapor and the vapor lines are constructed from 1/2 and 5/8 inch soft copper tubes. The expansion device is a 1/2 inch ball valve and the compressor is a rolling piston (fixed van) type.

The evaporation and condensation chambers are constructed with a shape coincide with the size and configuration of the condenser and the evaporator coils respectively. The intermediate zone is constructed from three inclined channels in the space between the evaporation and the condensation chambers. A (600) watt water heater is installed in the evaporation chamber using 5/8 inch carbon steel nut which welded to (20) mm diameter hole at the outer surface of the chamber below the charging point. This heater is used as start – heating to reach the raw water to the boiling degree at the beginning of the operation.

The liquid (distillated water) and vapor lines are constructed from 1/2 and 3/4 inch galvanized (wrought iron) pipes respectively. 3/4 inch gate valve is used as expansion device.

Distillated water is used as the working fluid in CEV loop since it's a cheap, good, and available working fluid. Its also used as process fluid in thermosyphon loop to facility the distillator performance investigation.

Instrumentation and Calibration

Temperature Measurement

Thermally sensitive resistors (Thermistors) are used to measure the working and process fluid temperatures in all parts of the test rig. The thermistors are connected to a selector switch. The selector switche are connected to a digital multimeter (hp 3435A) with a sensitivity of 0.1 Ω. Calibration of thermistor is carried out with temperature range of (70 – 110) ^{o}C.

Pressure Measurement

A pressure gages, bourdon type, are calibrated and used to measure the pressure of working fluid at the inlet and outlet of the evaporator and compressor.

Water Flow Rate Measurement

An orifice plate is calibrated and used to measure the volumetric flow rate of water inside the compression – expansion vapor loop.

Leak Tests

Before starting up the test runs for the system, the leak tests are carried out for the two – loops of the system.

System Insulation

In order to minimize the heat transfer with the surrounding, a glass wool sheets, with different thickness (1, 1.5, 2) cm, insulation covered by an aluminum foil are used to cover the all parts of the two – loops of the system, as shown in Fig. (7).

Results and Discussion:

The performance of the CEV loop working with different mass flow rate is defined as a coefficient of performance for distillator ($COP_{dist.}$), as follows:

$$COP_{dist.} = \frac{Q_e}{W_{comp.shaft}} \qquad \text{……..(43)}$$

Where,

$$Q_e = \dot{m}\left(H_{out} - H_{in}\right) \qquad \text{……..(44)}$$

$$W_{comp.shaft} = \frac{n}{n-1} P_1 \dot{v}_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad \text{……..(45)}$$

The effect of working mass flow rate distillator performance for different process fluid pressure is shown in Fig.(8) to Fig.(10) at different working fluid pressure. The experimental results are projected on the computed curves. From the figures it is shown that the experimental results are lower than the computed values. This difference may attribute to the experimental errors involved in the measurement of the COP_{dist}. In addition to the experimental errors, the systematic errors such as the presence of non – condensable gases in the loop, this gas will collect on the condensing side while condensation takes place.

For all figures, it is shown that at any process fluid pressure, increasing the working mass flow rate causes an increase in the distillator performance (i.e. COP_{dist}), this can be explained by the fact that when the working mass flow rate is increased, the liquid – vapor column in the evaporator will be increased. So that, the effective evaporator heat transfer surface area is increased as well. Therefore, the heat absorbed by the evaporator is increased and causes the loop performance to increase.

Conclusions:

1. The computed results were in good agreement with the experimental data obtained from the test rig, for the distillator performance.
2. The water is used as working fluid and can provide reasonably accurate results during the experiments. In addition it is cheap and available.
3. For all working fluid pressure, i.e. 0.7, 0.75 and 0.8 bar, any increase in the working mass flow rate will clearly increase the CEV loop performance due to increase the liquid – vapor column in the evaporator.
4. The maximum CEV loop performance was seen to occur at a working mass flow rate of 5.5 kg/hr at working fluid pressure of 0.8 bar.

Acknowlegement:

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