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Full Length Research Paper

Performance of a water ammonia absorption system operating at three pressure levels

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The present study deals with a compression-absorption machine. The proposed hybrid cooling system uses water-ammonia as a working fluid and operates at three pressure levels. The absorber is at an intermediate pressure (Pint) taken between the evaporator pressure (PEV) and the condenser pressure (PCD), unlike the single stage machine, which works between two pressure levels. The proposed new system is studied and compared to the conventional machine. In order to evaluate the performance of the invoked system, a procedure based on the MAPLE software is set up to compute accurately the thermodynamic properties of the working fluid. The analyses of the numerical results highlight that the performance of the novel proposed configuration is better than that relative to the conventional cycle. The study reveals the great impact of the intermediate pressure on the performance improvement and on reducing the generator temperature allowing the system to work at low enthalpy sources. In fact, for an evaporator temperature and a condenser temperature fixed respectively at -10 and 40°C, the proposed hybrid refrigeration cycle operates at a generator temperature TGE = 75°C and the installation's COP is about 0.56. While for the same conditions, the single stage machine COP cannot exceed 0.51 with a generator temperature of about 100°C.

Key words: Absorption, ammonia, COP, hybrid, refrigeration.

INTRODUCTION

Refrigeration is one of the leading consumers of electric power in the world. It is encountered in everyday life and it is greatly used in human activity. Its applications are numerous in many fields and sectors, such as agriculture, industry and commercial sector (Thioye, 1997). For these reasons, the valorisation of energy from low enthalpy resources is today of a great interest. These calories can be retrieved through tritherme cycles. This work is part of the systems development. We focus particularly on the single and double stage tritherme absorption machine, operating with water-ammonia. The main objective is to adapt them to the low temperature heat sources (Riffat et al., 2004). In this investigation we attach a particular interest to the optimization of the system's performance and the minimization of energy consumption of refrigeration cycles (Göktun, 2000; Laouir et al., 2002). In fact, a single-stage machine, composed of an evaporator, an absorber, a pump, a

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Abbreviations: COP, Coefficient of performance; **F,** circulation ratio; **h,** specific enthalpy (J/kg); $\dot{m}$, mass flow rate (kg/s); **P,** pressure (bar, Pa); $\dot{Q}$, heat-transfer rate (W); **T,** temperature (K, °C); **X,** mass fraction.

Subscripts: AB, Absorber; **CD,** condenser; **COM,** compressor; **EV,** evaporator; **GE,** generator; **η,** efficiency (%).

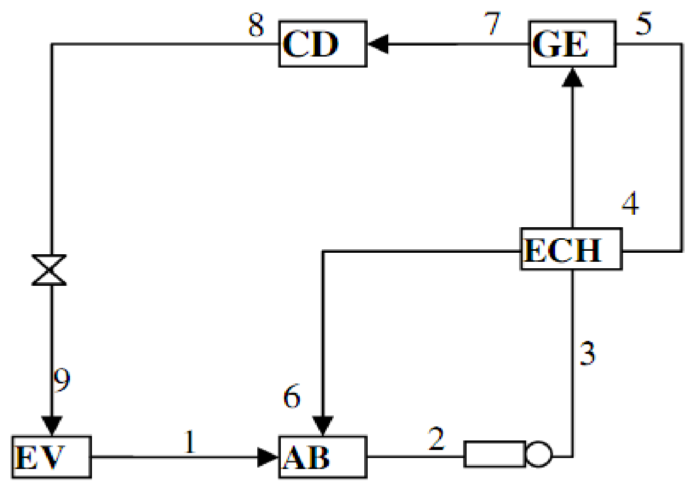

Figure 1. Conventional cycle.

generator, a condenser, two generators and an inter-exchange solution, has been frequently studied (Mumah et al., 1994; Alvares et al., 1987). We note that this standard absorption system using NH_3-H_2O cannot be operating at a generator temperature below 100°C, for a condensing and evaporating temperature respectively of 40 and -10°C. In addition, the maximum value of the single-stage absorption COP is approximately equal to 0.51 (Syed et al., 1999). Besides, it has been proved that if the number of stages increases, the coefficient of performance decreases (Hulten et al., 2002). In the present work, we propose a new configuration of a hybrid water ammonia absorption operating at three pressure levels. The novel configuration object of this study presents the double advantage of operating at low enthalpy sources and presents a relatively improved COP.

Modelisation

Several authors have studied different models of absorption machines using H_2O/NH_3 and $LiBr/H_2O$ as working fluids and geothermal energy and solar energy as energy sources (Mumah et al., 1994; Alvares et al., 1987). The absorption machine consists of six main elements evaporator, absorber, pump, generator, condenser and expansion valve. Some systems are composed of simple stage machine (Robert et al., 1997) and others are formed by a succession of stages with various component associations and sometimes inserting other new components (Arh et al., 1990; Sahina et al., 2002). To model an absorption cooling cycle, it is sufficient to model each component of the refrigeration cycle (Fukuta et al., 2002; Sun et al., 1996). Subsequently, we present the simple effect absorption machine and we develop a novel absorption hybrid configuration. We will explain and quantify its adaptability to low-enthalpy sources.

The components of the conventional system are the evaporator, the absorber, the pump, the generator, the condenser and two valves: one is used for the poor solution return to the absorber and the other for the refrigerant relaxation in the evaporator. It is well known that a single stage cooling system works between two pressure levels: evaporator pressure (PEV) and condenser pressure (PCD) (Kang et al., 2000). We have shown in a previous work that the conventional absorption machine, using NH_3-H_2O as a working fluid, cannot operate with a generator temperature below 100°C with evaporator and condenser temperature fixed respectively to -10 and 40°C (Kairouani et al., 2005). Indeed, its coefficient of performance (COP) cannot exceed 0.51. We keep in mind that the theoretical COP is calculated by considering the following assumptions:

(i) The temperature at each main component is uniform.
(ii) The refrigerant vapour is in thermodynamic equilibrium with the rich solution at the absorber entrance and the poor solution leaving the generator.
(iii) The fluid at the generator outlet can be considered in a saturated state with a steam quality less than unity (Romero et al., 2005).

In this study, the conventional absorption cycle parts are shown in Figure 1 and the corresponding cycle on the Oldham diagram is represented in Figure 2.

Subsequently, we propose the novel refrigeration system object of this work. The enhanced configuration is

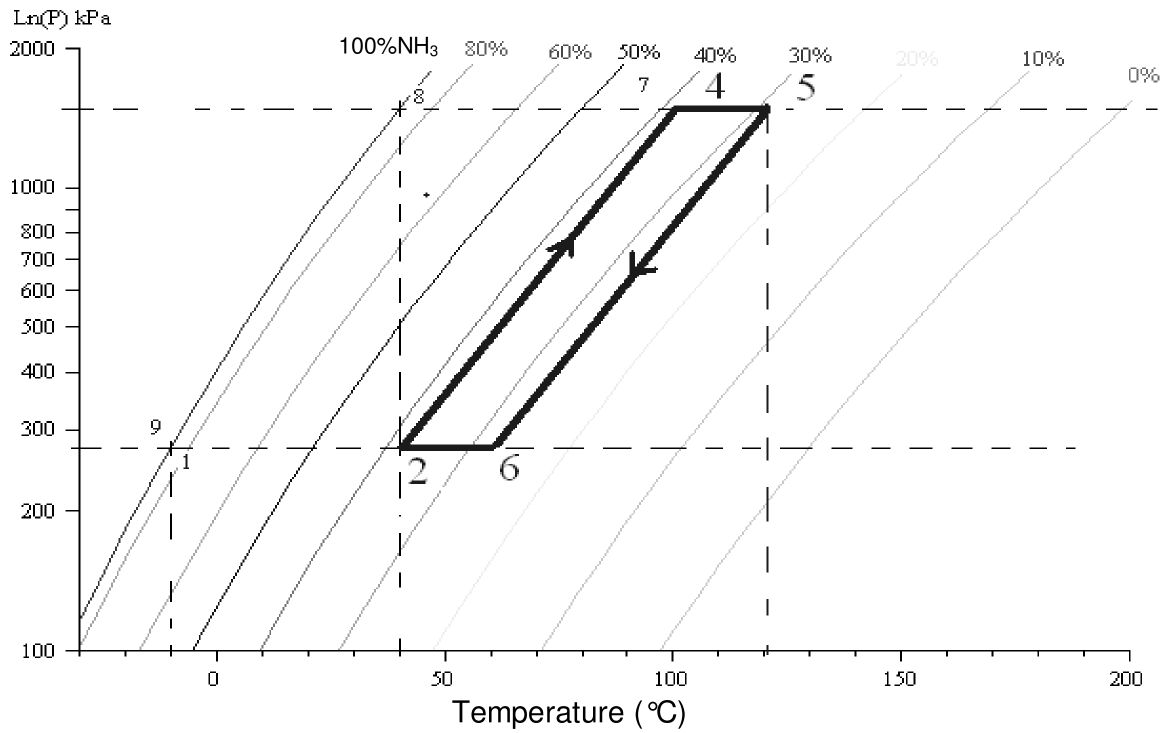

Figure 2. Diagram of Oldham for conventional machine.

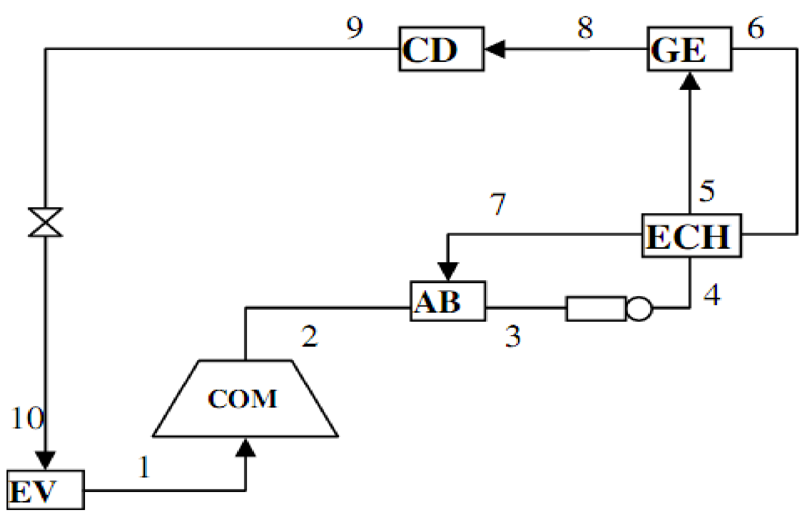

Figure 3. Proposed hybrid configuration.

operating at three pressure levels. This machine is composed by an evaporator, an absorber, a generator, a condenser and a compressor inserted between the evaporator and the absorber. The cycle parts are represented in Figure 3 and the corresponding cycle on the Oldham diagram is represented in Figure 4.

In the following, we develop the energy and mass balance. The specific solution circulation factor represents the mass of solution per kg of refrigerant vapour evaporated in the generator.

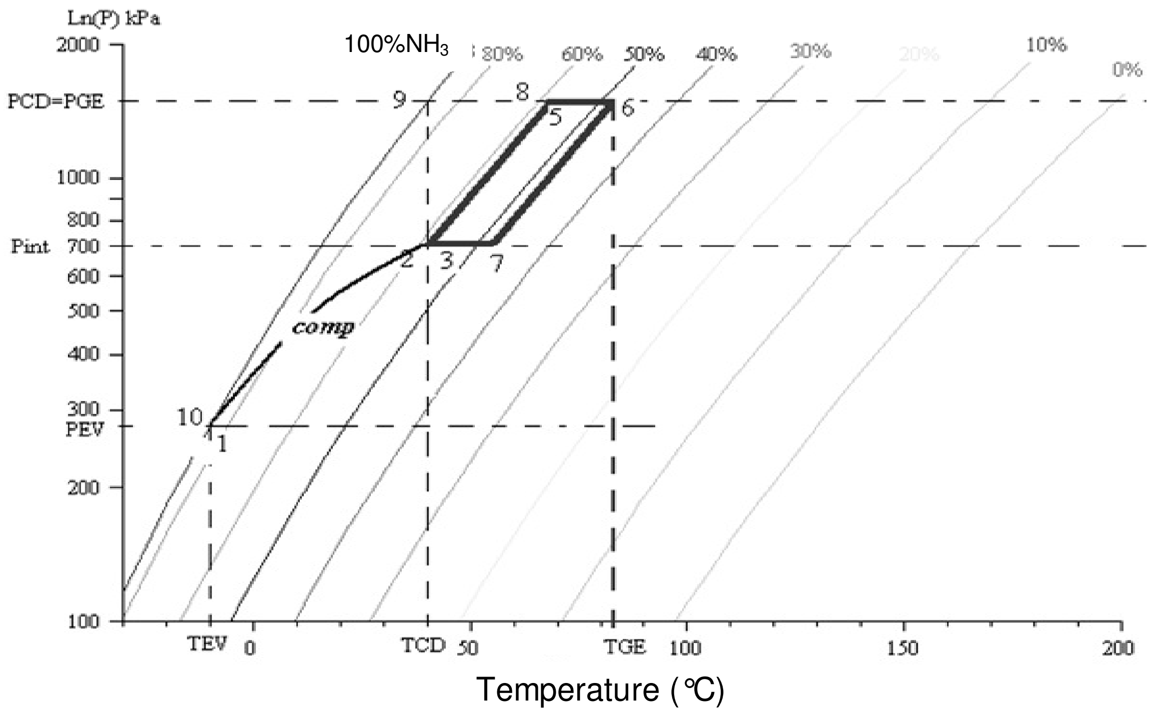

Figure 4. Diagram of Oldham for proposed configuration.

Table 1. Energy balances in each component of the hybrid absorption system

Condenser (8-9)	Evaporator (10-1)	Generator (5-6;8)	Absorber
$\dot{m}_9=\dot{m}_8=\dot{m}_{NH3}$	$\dot{m}_{10}=\dot{m}_1=\dot{m}_{NH3}$	$\dot{m}_5=\dot{m}_6+\dot{m}_8$	$\dot{m}_3=\dot{m}_2+\dot{m}_7$
$T_9=T_{CD}$	$T_1=T_{EV}$	$T_8=T(x_5,P_{CD})$. $T_6=T_{GE}$	$T_3=T_{CD}$
$h_9=h(T_{CD})$	$h_{10}=h_9$. $h_1=h(T_{EV})$	$h_6=h(x_6,P_{CD})$. $h_5=h(x_5,P_{CD})$	
$P_9=P(T_{CD})=P_{CD}$	$P_{10}=P_1=P(T_{EV})=P_{EV}$	$P_6=P_5=P_8=P_{CD}$	
	$\dot{Q}_{EV}=\dot{m}_{NH3}(h_1-h_{10})$	$x_6=x(T_6,P_{CD})$, $x_5=x_6+\Delta x$	$x_3=x_7+\Delta x$
$\dot{Q}_{CD}=\dot{m}_{NH3}\cdot(h_8-h_9)$		$\dot{Q}_{GE}=\dot{m}_{NH3}(h_8+(f-1)\cdot h_6-f\cdot h_5)$	$\dot{Q}_{AB}=\dot{m}_{NH3}(h_2+(f-1)\cdot h_7-f\cdot h_3)$

$$f=\frac{1-x_5}{x_4-x_5}$$

It carries the energy balances in each component of the installation, exchanging heat or work with the external environment. Neglecting the rectifier, Table 1 was obtained.

In order to determine the compressor power, we consider the following hypothesis:

(i) At the evaporator outlet (EV), vapour ammonia is assimilated as an ideal gas, so we have:

$$T_1\times P_1^{(k-1)/k}=T_2\times P_2^{(k-1)/k}$$

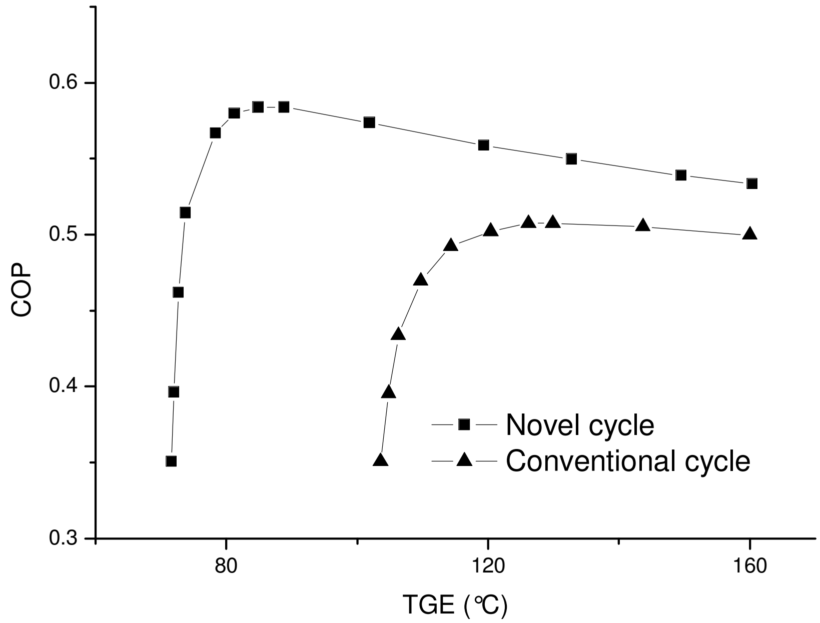

Figure 5. COP evolution versus TGE for the conventional cycle and the novel configuration.

(ii) In an ideal case, adiabatic process, it consumes power given by:

$$\dot{Q}_{is} = cp_{NH3} \times (T_2 - T_1)$$

Considering the isentropic efficiency, the real power is defined as follows:

$$\dot{Q}_{réel} = \frac{\dot{Q}_{is}}{\eta_{is}}$$

where: $\dot{Q}_{réel} = h_2 - h_1$

So we can deduce from the two previous expressions the value of the steam enthalpy at the compressor outlet:

$$h_2 = h_1 + \frac{\dot{Q}_{is}}{\eta_{is}}$$

The term performance is determined as follows (Laouir et al., 2002):

$\eta_{is} = 0.874 - 0.0135 \cdot \tau$ and $\tau = \frac{P_2}{P_1}$

We deduce the COP from the generator conservation:

$$COP = \frac{\dot{Q}_{Ev}}{\dot{Q}_{GE}} = \frac{h_1 - h_{10}}{h_8 + (f-1).h_6 - f.h_5 + h_2 - h_1}$$

RESULTS AND DISCUSSION

Several studies have been devoted to determine the COP and limitations of the absorption system operating conditions (Laouir et al., 2002; Fernandez et al., 2001).

In order to evaluate the refrigeration absorption system performance, we have developed a numerical program. The calculating procedures of the fluid thermodynamic properties and the performance coefficient were made using MAPLE computer tools. The numerical simulation, developed in the present investigation carries out a comparative study of the two systems performances.

Figure 5 shows the COP's evolution versus the generator temperature, for an evaporator temperature and a condenser temperature fixed respectively to -10 and 40°C. For the novel configuration, the absorption pressure is fixed to 650 kPa.

The analysis of Figure 5 shows that for the new proposed absorption machine, the coefficient of performance is higher than that relative to the classic system. It's about 0.58. While for the conventional configuration, the COP cannot exceed 0.51.

On the other hand, the operating temperature is relatively

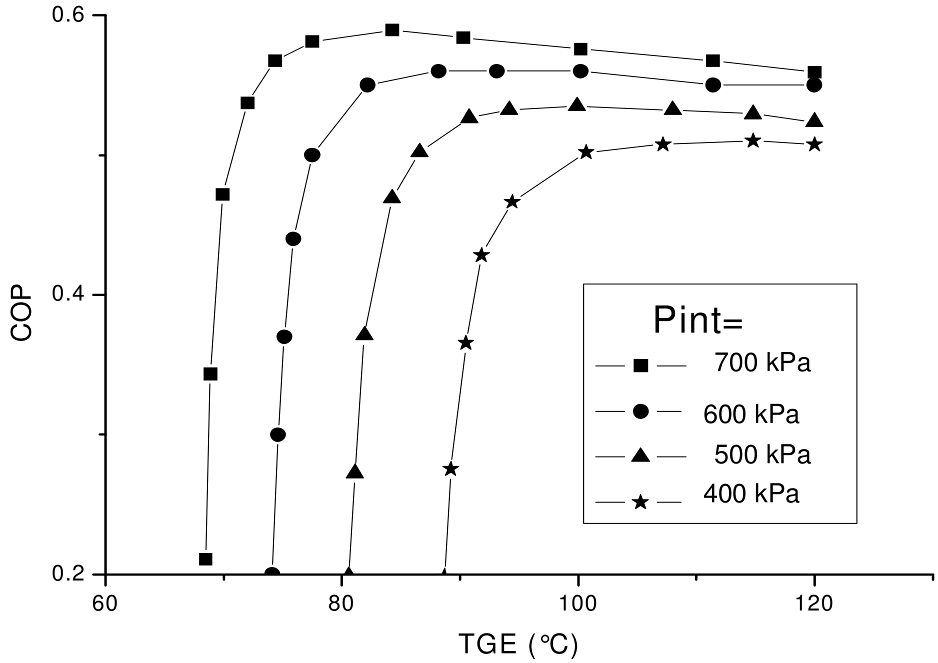

Figure 6. COP Evolution versus TGE with TEV = -10°C and TCD = 40°C for different intermediate pressure.

low, around 70°C compared to that relative to the simple effect cycle which is about 105°C. In addition, even for high temperature generator (TGE>105°C) the COP continues to be higher than that relative to the conventional cycle.

Besides we note that for the installation object of this work, a new parameter reveals interesting to study the performance of the cycle. It's the pressure of the absorber that is an intermediate pressure (Pint) as shown in Figure 6.

For a condensation temperature of 40°C and an evaporation temperature of -10°C, Figure 6 presents a family of curves showing the COP evolution versus the generator temperature for different intermediate pressure. We note that the new machine works on the whole domain of temperature and its coefficient of performance increases if the intermediate pressure increases. Besides, the operating temperature (TGE) decreases if the pressure of the absorber increases.

The enhanced configuration, object of this investigation allows consequently a gain of energy and gives the possibility to operate at low enthalpy sources.

On the other hand, we note that the COP depends mainly on the evaporating temperature (necessary for the production of the desired cold), the condensation temperature (function of cooling temperature of the absorber and condenser components) and finally generator temperature.

Figure 7 shows the COP evolution versus TGE for different TCD, with TEV = -10°C and Pint = 650kPa.

Figure 8 shows the COP evolution versus the intermediate pressure (pressure of the absorber) for TGE = 80°C, an evaporator temperature and a condenser temperature fixed relatively to -10 and 30°C. We recall that under these conditions, the COP of the conventional system does not exceed 0.4.

Conclusion

In this investigation, we proposed a novel configuration of an absorption cooling machine using water-ammonia as a working fluid and operating at three pressure levels. The proposed hybrid cycle, where the absorber is at an intermediate pressure taken between the evaporator and the condenser pressure is analysed and compared to the conventional cycle. The numerical simulation highlights the improved performance of the novel system. It showed the double advantage of the hybrid absorption configuration. In fact the hybrid configuration presents the opportunity to operate at low enthalpy source with a better COP. For an evaporator temperature and a condenser temperature fixed respectively to -10 and 40°C, the compression-absorption machine operates at a generator temperature, TGE = 75°C with an installation's COP about 0.56 while for the same conditions, the one

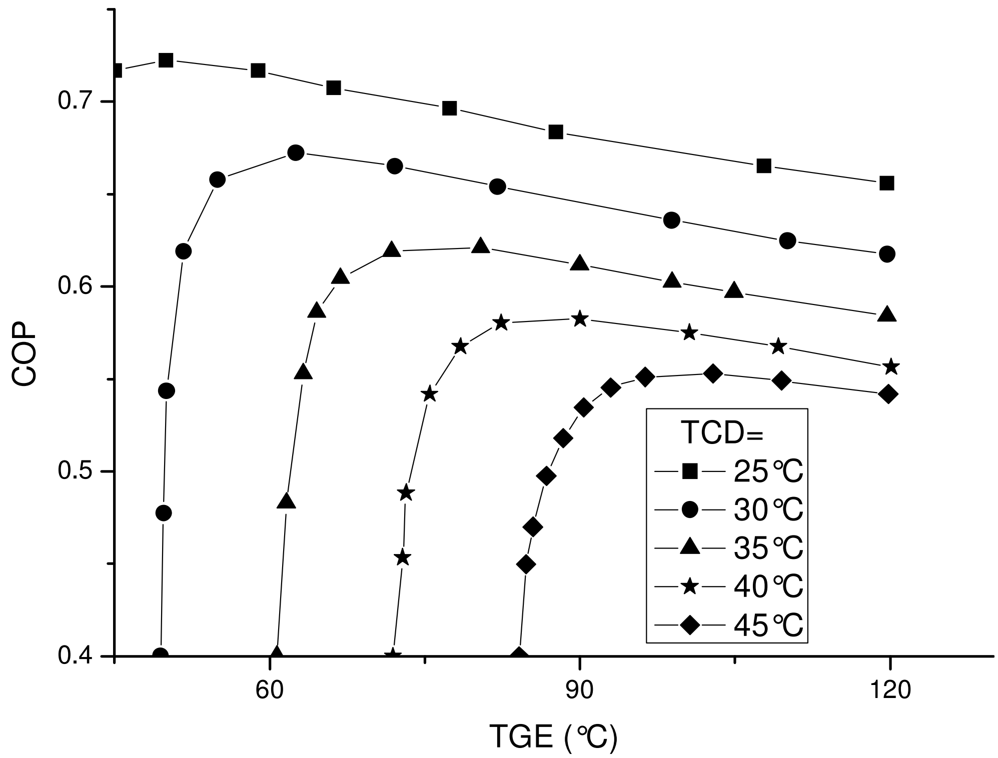

Figure 7. COP evolution versus TGE with TEV = -10℃ and Pint = 650 kPa for different TCD.

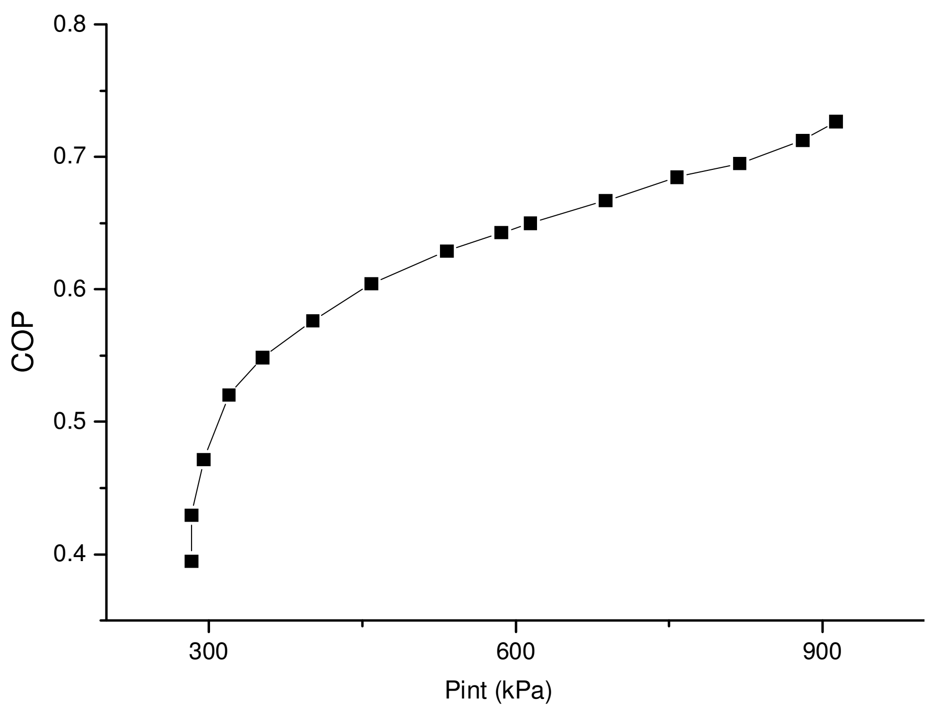

Figure 8. COP evolution versus pint for TGE = 80℃, TCD = 30℃ and TEV = -10℃.

stage machine, functioning at two pressure levels has a COP which cannot exceed 0.51 with a generator temperature of about 100°C. The numerical simulation results carried out in this investigation show also that the intermediate pressure has not only influence on the COP enhancement of the refrigeration cycle but it also has a great impact on reducing the generator temperature. Consequently, it contributes to the adaptability of refrigeration systems to low temperature sources allowing the use of waste heat in many industrial fields.

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