HYBRID SINGLE-STAGE TRIPLE PRESSURE LEVEL ABSORPTION/COMPRESSION CYCLE OPERATED BY LOW GRADE HEAT SOURCES

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Abstract: The performance of the triple-pressure level (TPL) single stage absorption cycle operated with organic refrigerants and absorbents showed many advantages over the common double pressure level (DPL) absorption cycle. In order to enhance these advantages (increased COP) a mechanical compressor and a mixing device were inserted in the super heated refrigerant line between the evaporator and the absorber. The influence of the elevated pressure on the performance of the TPL absorption cycle with the working fluid pentafluoroethane (R125) and N,N'-dimethylethylurea (DMEU) was predicted by a computerized simulation program. The performances of two configurations of the TPL absorption cycle operated with mechanical compressor were compared; a) with common solution heat exchanger (HS) and b) divided solution heat exchanger (pApG). Based on the analysis, in the first configuration a significant reduction of the required generator temperature with increased coefficient of performance (COP), reduction in the circulation ratio (*f*) and reduction of the actual size of the solution heat exchanger was found. In the second configuration a significant increasing of the COP with less reduction of the required generator temperature but with increased actual size of the solution heat exchanger was found. The disadvantage of inserting the compressor is the increased electrical consumption.

Key words

Hybrid absorption system, triple pressure absorption system, absorption/compression hybrid cycle.

1. Introduction

Among various heat sources, the range of low grade temperature sources, preferably up to 130°C, such as solar energy, waste heat etc., is an important and difficult range for utilization and recovery. The utilization of these low grade heat sources for cooling and refrigeration by means of absorption systems with different working fluids usually leads to the necessity of a cooling tower. Various configurations of absorption systems are practically utilized.

The basic DPL absorption cycle includes two sub cycles; the solution and the refrigerant sub cycles. The solution sub cycle includes an absorber (where the cold refrigerant vapor from the evaporator is absorbed at low pressure), a generator (where the hot refrigerant vapor is generated at high pressure), a solution heat exchanger (an economizer where heat is transformed from the hot weak solution to the cold strong solution), a solution pump and a pressure reduction device. The refrigerant sub cycle includes the condenser, evaporator, refrigerant heat exchanger (an economizer where the hot condensate is sub-cooled by the cold refrigerant vapor) and expansion valve. In the basic DPL absorption cycle, the pressure of the absorber and the evaporator is the same and similarly the pressure of the generator and the condenser is the same (pressure drop is neglected).

An advanced single-stage triple pressure level (TPL) absorption cycle that utilizes a low potential heat source for cooling, by integrating a specially designed jet ejector at the absorber inlet, as presented by Levy et al. [1-2] and Jelinek et al. [3] is shown in Fig 1a. The major functions of the jet ejector are the ability to facilitate mixing and pressure recovery i.e. higher absorber pressure relative to the evaporator

pressure. Jelinek et al. [3] presented the improved performances of the TPL cycle with working fluids based on various HCFC and HFC refrigerants and DMEU (N,N'-dimethylethylurea) as the absorbent due to the pressure recovery by the jet ejector.
While the ability to pressure recovery by the jet ejector is limited, a compressor can be added between the absorber and the evaporator instead the jet-ejector mixer in order to increase and control the absorber pressure. This improvement leads to a hybrid single-stage TPL absorption/ compression cycle as shown in Fig 1b.

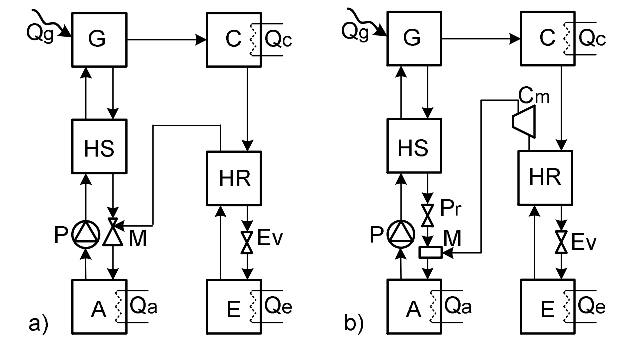

Fig. 1: Single-stage TPL absorption cycle with: a) jet-ejector mixer and b) a compressor and mixing device instead the jet-ejector mixer (cycle HS).

The performances in terms of COP, *f*, Qhs/Qe and kW/T of the TPL absorption cycles (Fig 1) with the working fluid R125 (pentafluoroethane) as the refrigerant and DMEU as the absorbent were studied by Jelinek et al. [4] under the following conditions: generator temperature in the range of 50 to 120°C, evaporator temperatures of -5°C and cooling water temperature of 25°C (condenser temperature 32°C and absorber temperature 28°C) where an isentropic compressor was assumed and the pressure drops along the cycles were neglected. Jelinek et al. [4] showed that as the pressure difference between the absorber and the evaporator increases, the COP increases and the generator temperature at maximum COP decreases. The COP takes into account the increasing in the electrical consumption due to the added compressor [COP=Qe/(Qg+Wp+ Wcomp)].
Further improvement of the cycle in terms of COP can be achieved by dividing the solution heat exchanger into two separate economizer heat exchangers; pre-generator pG and pre-absorber pA as shown in Fig 2.

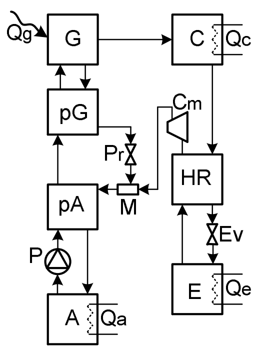

Figure 2: Single-stage TPL absorption/ compression cycle with pre-generator - pG, pre-absorber - pA and a compressor and a mixer instead the jet-ejector (cycle pApG).

2. Present study

Comparison between the performances of the two TPL cycles (HS and pApG shown in Fig 1b and Fig 2, respectively) under the same operating conditions shows that the circulation ratio *f* and the electrical consumption kW/T are the same (due to the same weight fractions and pressures). However, the COP and the Qhs/Qe are showing different behavior.
The calculated COP of TPL absorption cycle with a compressor in the two configurations as a function of the generator temperature at various pressure differences dPcomp is shown in Fig 3. The COP for the cycle pApG (Fig 2) was found to be higher than for the cycle HS (fig 1b) at the same dPcomp but at higher generator temperature. As can be seen, the range of dPcomp in cycle pApG (4-6bar) is narrower than in cycle HS (0-6bar). For dPcomp in the range of 4 to 6bar the maximum COP in cycle HS is in the range of 0.69-0.66 and generator temperature in the range of 60-70^{o}C respectively, while in the cycle pApG the maximum COP is in the range of 0.96-0.79 and generator temperature in the range of 85-100^{o}C, respectively. This means that at same cooling capacity, higher temperature heat source with less heat input is required for the cycle pApG in comparison with cycle HS.
The calculated Qhs/Qe for the two configuration cycles as a function of the generator temperature at various dPcomp is shown in Fig 4. In cycle pApG (Fig 2) where Qhs=QpA+QpG, Qhs/Qe show higher value then in cycle HS (Fig 1b) for the same dPcomp at relevant generator temperature. For dPcomp in the range of 4 to 6bar the value of Qhs/Qe in cycle HS is 0.7-0.4

while in the cycle pApG the value of Qhs/Qe is 0.8-1.2. This means that heat transfer area in the cycle pApG is up to double than in cycle HS for the same operating conditions. This behavior of the cycle pApG is due to the fact that the necessary temperature differences have to be kept in each heat exchanger (pA and pG) along the cycle.

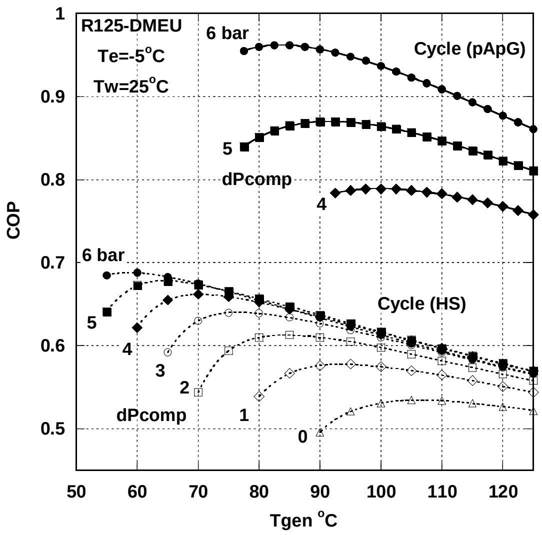

Fig. 3. The calculated COP of hybrid TPL absorption cycle as a function of the generator temperature at various dP with R125-DMEU.

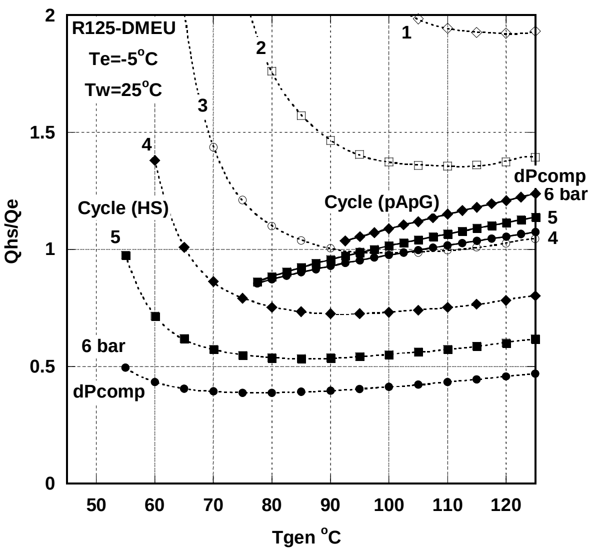

Fig .4. The calculated Qhs/Qe of hybrid TPL absorption cycle as a function of the generator temperature at various dP with R125-DMEU.

The circulation ratio *f* and the electrical consumption kW/T were taken from Jelinek et al. [4] and are shown in Fig. 5 and Fig. 6.

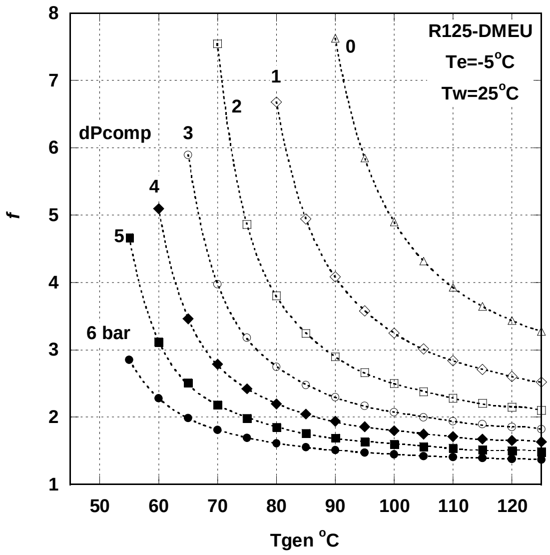

Fig. 5. The calculated circulation ratio f of hybrid TPL absorption cycles as a function of the generator temperature at various dP with R125-DMEU.

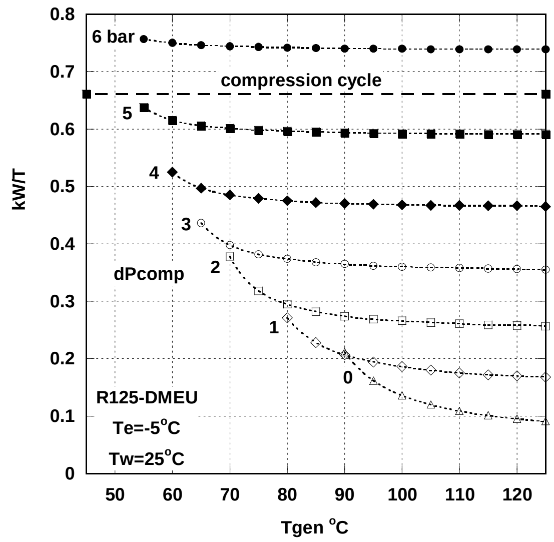

Fig. 6. The calculated kW/T of an isentropic compression cycle and of hybrid TPL absorption cycles as a function of the generator temperature at various dP with R125-DMEU.

The differences in the behavior of the performances of these two configurations in terms of COP versus generator temperature and the effects on the other components are clarified by the following example. For evaporator capacity of 1TR (refrigerant mass flow rate of 109.5 kg/hr) and dPcomp of 4bar, the data from Fig 3 to Fig 6 at maximum COP for the two hybrids TPL cycle configurations (cycle HS and cycle pApG) and for isentropic compressor are summarized and compared in Table 1.

Table 1. The performances of TPL cycles with R125-DMEU at maximum COP.

	Cycle HS	Cycle pApG
Tg °C	70	100
COP	0.662	0.789
f	2.786	1.796
Qhs/Qe	0.863	1.088
kW/T	0.485	0.468
High press. (bar)	16.50	
Interm. press. (bar)	9.71	
Low press. (bar)	5.71	
Qg kcal/hr (kW)	4528 (5.27)	3799 (4.42)
Qhs kcal/hr (kW)	2589 (3.01)	3264 (3.80)
$\dot{m}s$ kg/hr	305	196.6
EC* kW	0.485	0.468
$Qe/\Sigma W$ **	7.2	7.45
Isentropic compressor		
EC kW	0.661 and COP=5.28	
$Qe/\Sigma W$	5.28	

* Electrical consumption
** Solar energy taken as a "free of charge" heat source (1TR = 3000 kcal/hr = 3.488 kW, $\dot{m}r$ = 109.5 kg/hr).

It can be seen from Table 1 that the advantage of cycle pApG over cycle HS is evident in term of higher COP (0.789 and 0.662, respectively), the lower heat source capacity (3799 and 4528 kcal/hr) and by the lower solution mass flow rate (196.6 and 305 kg/hr). The disadvantage of cycle pApG over cycle HS is evident in terms of higher generator temperature (100 and 70°C) and heat transferred in the economizer heat exchangers (3264 and 2589 kcal/hr).

The electrical consumption of the two hybrids TPL cycle configurations (cycle HS and cycle pApG) are almost the same (0.485 and 0.468 kW) while for the isentropic compressor the electrical consumption is 0.661 kW at the same operating conditions. This means electrical saving of about 28%.

If solar energy is used as the heat source and taken as "free of charge" for the absorption/compression cycle, the value of the product of the heat rejected by the evaporator divided by the sum of the electrical input ($Qe/\Sigma W$) is equal to 7.2-7.45 while the same product for the isentropic compression cycle is 5.28 as shown in Table 1.

3. Discussion and conclusion

From the above analysis, these two hybrid TPL cycle configurations (cycle HS and cycle pApG) are showing two different solutions for the same operating conditions depending on the nature of the heat source. If the available heat source temperature is above 110°C, cycle pApG is preferable, but if not, cycle HS has to be used although it's performance is inferior.

If solar energy is used and is taken as "free of charge", the electrical saving of these two hybrids TPL cycle configurations (cycle HS and cycle pApG) in comparison with the isentropic compressor is about 36 to 41%.

References

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Nomenclature

COP - Coefficient of performance [Qe/(Qg+Wp+ Wcomp)]
dPcomp- The pressure difference between the absorber and evaporator [bar]
f - Circulation ratio [mass flow rate of strong solution divided by mass flow rate of refrigerant]
kW/T - Shaft work per ton refrigeration
$\dot{m}r$ - Refrigerant mass flow rate [kg/hr]
$\dot{m}s$ - strong solution flow rate [kg/hr]
Qe - The heat rejected by the evaporator
Qg - The heat supplied to the generator

Qhs - The heat transferred at the solution heat exchanger

Qhs/Qe- The heat transferred at the solution heat exchanger divided by the heat rejected by the evaporator

Te - Evaporator temperature [oC]

Tg - Generator temperature [oC]

Tw - Cooling water temperature [oC]

Wp - Energy supply to the pump [kcal/kg]

Wcomp- Energy supply to the compressor [kcal/kg]