THEORETICAL STUDY OF THE COMBINED M-CYCLE/EJECTOR AIR-CONDITIONING SYSTEM

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The Maisotsenko cycle (M-cycle) for air-conditioning technologies offers opportunities for energy conservation and reduction of greenhouse gas emissions. It also improves the quality of the cooled air without additional inputs for the return air system.

The high efficiency of the M-cycle is observed at low relative humidity of the outside air, which restricts the M-cycle detached application. Obviously, an application of the M-cycle paired with conventional electrically-driven cooling systems will worsen energy characteristics due to the compressor work required. If the ejector refrigerating system (ERS) is combined with the M-cycle, the overall system performance increases as described in this paper. The combination of the M-cycle with ejector-based cooling systems explores application spheres of the M-cycle associated air conditioners.

KEY WORDS: dew point, evaporative cooling, Maisotsenko cycle, effectiveness, binary fluid, ejector, air cooling

INTRODUCTION

Conventional air-conditioning systems (ACS) have one major demerit (related to a high-return airflow) that worsens the quality of the inhalant air and favors the abundance of bacteria and viruses. Air-treatment measures appear to be quite expensive and increase power consumption. The M-cycle, advertised in 1990s, can serve as a valuable asset to change the concept of air conditioning and create a
new generation of air conditioners with improved performance and energy conservation rate (Zhan et al., 2011). However, in the majority of climatic zones, the single M-cycle air conditioner is not always efficient for reaching the necessary parameters on the respiration air, and an additional cooling system must be paired. At the present time, conventional vapor-compression refrigeration systems are applied as an attached cooler, which results in additional power consumption. As a substitute for a vapor-compression system, ejector refrigerating systems (ERS) are suggested that will serve to dehumidify air after the M-cycle and cool the treated air with no extra load on the power grids.

**BASELINE FOR ERS APPLICATION IN AIR-CONDITIONING MODE**

The application of steam–water ERS is limited due to its low energy performance. Developments of solar thermal systems encouraged an interest in solar cooling, in which ERS became one of the few applied technologies. Various ERS operating with low-boiling refrigerants and binary fluids have been examined. It was found that application of binary fluid ERS (BERS) could significantly increase the COP of air-conditioning systems (Buyadgie et al., 2011, 2012 b).

Figure 1 shows COP curves of the steam–water, single-fluid, and binary fluid ERS versus condensation temperature at the set evaporation temperature $t_{eva} = 3{^\circ}C$ and generation temperature $t_{gen} = 100{^\circ}C$. 

### NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>COP</td>
<td>coefficient of performance</td>
</tr>
<tr>
<td>$d$</td>
<td>absolute humidity (%)</td>
</tr>
<tr>
<td>$i$</td>
<td>enthalpy, kJ/kg</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature, K</td>
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<tr>
<td>$w$</td>
<td>power, kW</td>
</tr>
<tr>
<td>$\eta$</td>
<td>energy conversion efficiency</td>
</tr>
<tr>
<td>$q$</td>
<td>specific heat of evaporation, kJ/kg</td>
</tr>
<tr>
<td>$Q$</td>
<td>cooling capacity, kW</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>relative humidity, %</td>
</tr>
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<table>
<thead>
<tr>
<th>Subscripts</th>
<th>Superscripts</th>
</tr>
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<tr>
<td>ACS</td>
<td>ERS</td>
</tr>
<tr>
<td>cooling</td>
<td>comp</td>
</tr>
<tr>
<td>cooling capacity of the ACS</td>
<td>compressor</td>
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The condensation temperature varied in a range from 34°C to 45°C. The graph shows that the highest COP values belong to BERS over the entire condensation temperatures range; single-fluid ERS shows intermediate results, and steam–water ERS shows the lowest COP values. Binary fluid ERS operating with R-11/R-600 reaches the highest COP.

The COP of the ERS is defined in Eq. 1:

\[
COP = U \frac{q_o}{q_w},
\]  

where \(q_o\), specific heat of evaporation of refrigerant fluid (kJ/kg); \(q_w\), specific heat of evaporation of working fluid (kJ/kg).

The entrainment ratio was calculated using a method described by Buyaday et al. (2012a). The enthalpy of vapor and liquid in evaporator and vapor generator were defined from REFPROP (Lemmon et al., 2010). The fluids studied were selected according to the Montreal and Kyoto protocols. The critical temperature of the fluids, their toxicity and explosive properties were carefully considered prior to selection. The binary fluid R-11/R-600 represents a pattern of high thermodynamic properties of the cycle with possible replacement by R365mfc/R600 as quite tolerable to ODP and GWP requirements.

M-CYCLE EFFICIENCY IN A WIDE RANGE OF OUTSIDE RELATIVE HUMIDITY

At the first stage, limitations of relative humidity for the direct M-cycle were determined considering that the onset temperature of the outside air is 43°C and the target temperature of the inside air is 22°C (Gillan, 2008; Maisotenko, 2002, 2003; Wani et al., 2012). These results were described by Maisotenko et al. (2011).
As Fig. 2 shows, the performance range of the M-cycle for cooling the inside air down to 22°C lies between 12.5% and 30% of the outside relative humidity. For this purpose, the mass flow rate of the cooled air makes fifty-fifty with the inlet air at relative humidity $\varphi = 12.5\%$ and makes $6.25/93.75$ at $\varphi = 30\%$. Thereby, the number of cycles increases from 1 to 4 and final humidity of the cooled air reaches 100% from 41% in the first case (as 22°C is almost on the dew point level). In the last case, the fan power will be increased eightfold and bring the total power consumption of the M-cycle to the level of the conventional vapor-compression cooling systems. By pairing the additional cooler with the M-cycle it is possible to cool the inside air down to 22°C at the outside air temperature of 43°C and outside relative humidity exceeding 30%. Even at the lower outside relative humidity (16–18%), the additional cooler is still required because the relative humidity of the air, cooled in the M-cycle to the temperature of 22°C, exceeds 50–60%. When outside air is lower than 12.5%, the M-cycle can cool the inlet air down to the temperature of 13°C and desirable relative humidity.

As a result, while designing an air conditioner for a particular climate, the maximal cooling capacity of the additional cooler should be considered.

**FIG. 2:** Schematic of the evaporative cooling process. 1–2 evaporative cooling process of the inlet air at $i_{\text{const}}$; 1–3 cooled air cooling process at $d_{\text{const}}$; 4 cooled air dew point; 5–6 cooled air step cooling process in M-cycle to the dew point 6
COMBINATION OF M-CYCLE WITH VAPOUR-COMPRESSION AND EJECTOR AIR-CONDITIONING SYSTEMS

A schematic of the M-cycle paired with vapor-compression refrigerating system (VCRS) is presented in (Kozubal and Slayzak, 2009). All parameters to be used for the ejector refrigerating system are taken from the above paper. Figure 3 shows the M-cycle with BERS as an additional cooler.

The BERS operation conditions are set as follows: generation temperature $t_{\text{gen}} = 100$, evaporation temperature $t_{\text{eva}} = 3^\circ C$, and condensation temperature varies from $30^\circ C$ to $48^\circ C$. These parameters correspond well with the working parameters of the air coming out from the Coolerado heat mass exchanger (HMX) (Fig 3).

The outside fresh air is split into two streams inside the HMX channels. Humid air is cooled at the expense of moisture evaporation that cools treated air in a dry channel. As a result, before entering a room, the cooled air with a temperature of $22^\circ C$ and existing humidity is delivered into air cooler 8 for subcooling down to $13^\circ C$ and equilibrium humidity. The exhaust air from the working channel with an initial temperature of the $26^\circ C$ is supplied first to the refrigerant condenser 5 and then to the fractionating condenser 4. Condensation temperature in the fractional condenser is 8–10°C higher than in the refrigerant condenser, thus the heat exchange process is more reversible comparing to condensation at the constant temperature.

Heat-transfer fluid is heated in the solar collector and releases its heat to the vapor generator. The ejector is applied as a jet compressor in the system.

In the proposed combined system, condensation of the working fluid occurs in the fractionating condenser simultaneously with the refrigerant vapor separation.

**FIG. 3:** Diagram of the combined ACS with M-cycle and BERS. 1 solar collector, 2 vapor generator, 3 ejector, 4 fractionating condenser, 5 refrigerant condenser, 6 thermal pump, 7 throttling valve, 8 air-cooler
The refrigerant vapor condenses at the lowest condensation temperature by the processed cold air delivered to the refrigerant condenser directly from the Coolerado’s HMX. Figure 4 shows an i–d diagram of the M-cycle paired with an additional cooler.

The COP comparison of three variants of the air-conditioning systems was carried out using Eqs. 2–5:

\[
COP_{ACS}^{VCRS} = \frac{Q_{\text{cooling}}}{w_{\text{comp}} + w_{ca}}, \tag{2}
\]

\[
COP_{ACS}^{ERS} = \frac{Q_{\text{cooling}}}{c_{\text{fan}} + \frac{Q_{\text{cooling}}}{COP_{ERS}} (1 - \frac{T_{at}}{T_{gen}}) \eta_{pc}} \tag{3}
\]

\[
COP_{ACS}^{M-Cycle,VCRS} = \frac{Q_{\text{cooling}}}{w_{\text{comp}} + w_{ca} + w_{\text{fan}}} \tag{4}
\]

\[
COP_{ACS}^{M-Cycle,ERS} = \frac{Q_{\text{cooling}}}{w_{ca} + w_{\text{fan}} + \frac{Q_{\text{cooling}}}{COP_{ERS}} (1 - \frac{T_{at}}{T_{gen}}) \eta_{pc}} \tag{5}
\]
where $COP_{VCRS}^A$, COP of the VCRS; $COP_{ERS}^A$, COP of the ERS; $COP_{M-Cycle,VCRS}^A$, COP of the combined M-cycle and VCRS; $COP_{M-Cycle,ERS}^A$, COP of the combined M-cycle and ERS; $Q_{cooling}$, main system’s cooling capacity (kW); $w_{comp}$, compressor power for the additional cooling (kW); $w_{comp}^{ca}$, compressor power for the full cooling capacity; $w_{fan}$, fan power for the cooled air (kW); $w_{fan}^{ia}$, fan power for the inlet air; $Q_{cooling}^{ca}$, additional cooler cooling capacity; $T_{air}$, outside air temperature; $T_{gen}$, generation temperature; $\eta_{pc}$, exergy COP of the power cycle.

Energy efficiency was calculated for 12.5% and 100% of relative humidity and ambient temperature of 43°C and 30°C. The COP was calculated for different types of the air-conditioning systems: conventional vapor-compression air conditioner, BERS air conditioner, combined M-cycle with vapor-compression air conditioner, and M-cycle with BERS air conditioner. Also energy standing costs per 1 kg/s of processed air from ambient temperature and humidity to 13°C and relative humidity of 50% were calculated (Figs. 5, 6, 7).

The presented diagrams show that the efficiency of each system decreases when the outside relative humidity increases. This is due to increase of the required cooling capacity with simultaneous decrease of the evaporative cooling effect. At $\varphi = 60\%$ the evaporative cooling effect becomes almost negligible.

When the outside relative humidity is lower than 13%, the single M-cycle or air humidification at constant enthalpy can cool the inlet air to the comfort conditions without any additional coolers. On the other hand, when the relative humidity exceeds 80%, the efficiency of the combined cycles almost matches that of a single refrigeration cycle, making unreasonable the application of the combined cycles.

![FIG. 5: COP comparison of the ACS at various outside relative humidity ($t_{air} = 43°C$)](image-url)
The same tendency is observed when the ambient temperature is 30°C. In this case, an application of the M-cycle combined with BERS makes sense if the outside relative humidity is below 60%.

The specific energy costs of each variant of ACS increase when the outside relative humidity grows. However, the combined cycles with BERS will always be the

**FIG. 6:** COP comparison of the ACS at various outside relative humidity ($t_{at} = 30^\circ C$)

**FIG. 7:** The diagram of energy costs per unit for various ACS vs. outside relative humidity at ambient temperature 43°C
least expensive. This favors the combined M-cycle and BERS system, whose scope of application covers almost the entire range of the ambient relative humidity. At high values of outside relative humidity, the main function of the M-cycle is cooling the inlet air for BERS condensation heat release, which increases the COP of BERS and the whole system itself. When the ambient temperature decreases, the impact of the M-cycle as a part of ACS goes down.

Figure 8 illustrates that the relative reduced COP of combined M-cycle/BERS system is 40–70% higher than for the M-cycle/VCRS combined system at the all range of the outside relative humidity.

CONCLUSIONS

This research confirmed the practicability of the combined BERS and M-cycle system for air conditioning in almost the entire outside air relative humidity range with initial temperature over 40°C. Furthermore, M-cycle-based air-conditioning systems must have an additional cooler even in dry climate countries because its sustainability decreases when $\varphi > 20$–30%. The COP calculation in a wide range of outside air relative humidity and temperature showed that with the outside air temperature decreasing the use of combined cycles is inadvisable.

REFERENCES


