Maisotsenko cycle applications for multistage compressors cooling

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Abstract. The present study provides the overview of Maisotsenko Cycle (M-Cycle) applications for gas cooling in compressor systems. Various schemes of gas cooling systems are considered regarding to their thermal efficiency and cooling capacity. Preliminary calculation of M-cycle HMX has been conducted. It is found that M-cycle HMX scheme allows to brake the limit of the ambient wet bulb temperature for evaporative cooling. It has demonstrated that a compact integrated heat and moisture exchange process can cool product fluid to the level below the ambient wet bulb temperature, even to the level of dew point temperature of the incoming air with substantially lower water and energy consumption requirements.

1. Introduction
Gas compression and transportation is one of the most energy consuming technologies in the world. Compressors are a vital link in the conversion of raw materials into refined products. Compressors also handle economical use and transformation of energy from one form into another. They are used for the extraction of metals and minerals in mining operations, for the conservation of energy in natural gas reinjection plants, for secondary recovery processes in oil fields, for the utilization of new energy sources such as shale oil and tar sands, for furnishing utility or reaction air, for oxygen and reaction gases in almost any process, for process chemical and petrochemical plants, and for the separation and liquefaction of gases in air separation plants and in LPG and LNG plants. The economy and feasibility of all these applications depend on the reliability of compressors and the capability of the compressors selected to handle a given gas at the desired capacity [1]. Since, value of specific work for all compressors is determined by several factors as: specific heat ratio $k$, specific gas constant $R$, pressure ratio $\pi$ and inlet temperature $T_{in}$. As a rule, all the enumerated factors except the inlet temperature are defined by the initial conditions and/or performance specification for the designed compressor. That is why inlet temperature reduction remains the most feasible way to reduce compressors’ specific work.

There are a lot of approaches to reduce $T_{in}$ value (depending on the place of cooling):...
• Pre-cooling (cooling at suction pipe).
• Cooling in compression process (cooling of cylinder jacket, envelope of compression channels).
• Interstage cooling (cooling after each compressor stage or section).
• Aftercooling (cooling at discharge pipe).
However, these approaches could be implemented by different means:
• Vapor compression refrigeration (VCR)
• Adsorption refrigeration (AR)
• Injection cooler
• Air/water-cooled heat exchanger
• Indirect evaporative cooling (IEC) heat and mass exchanger (HMX)
• Maisotsenko cycle (M-cycle) heat and mass exchanger (HMX)

Considering first two methods of refrigeration, one should say that VCR and AR systems could provide the highest level of cooling due to using a low-boiling refrigerant. Though, vapor compressor refrigerator has lower efficiency than absorption refrigerator, but has smaller size and simple construction. However, VCR and AR have common disadvantages, which restrict or turn off their benefits: 1) consume too much energy; 2) refrigerant leakage risk; 3) high price of the refrigeration units, their installation and maintenance.

Injection cooler is applicable restrictively as to excessive moistening of the gas. Moreover, it could lead to compressors parts corrosion in case of off-designed cooling regime.

Air/Water-cooled heat exchanger has big specific quantity of metal and installation costs. It is necessary to provide a large amount of water and its cooling in case of closed circuit of water supplying system. Air-cooled heat exchanger has poor low cooling capacity due to low specific heat of cooling air. Therefore these type of heat exchangers requires large area of cooling surfaces.

IEC has acceptable cooling capacity, but wet bulb temperature is the limit of gas cooling for one-stage design apparatus.

The most promising technology of gas cooling is M-cycle process. This process allows to reach dew-point temperature in one HMX stage.

2. Fundamentals of M-cycle HMX
The M-Cycle is a thermodynamic process which captures energy from the air by utilizing the psychrometric renewable energy available from the latent heat of water evaporating into the air. It combines thermodynamic processes of heat transfer and evaporative cooling to facilitate product temperature to reach the dew-point temperature of the ambient air [2]. The basic principle and features of the M-Cycle can be explained from Figure 1 (a) and Figure 1 (b) representing the partial extraction of air and product air cooling M-Cycle, respectively [3]. The psychrometric representation of both variations of M-cycle is shown in Figure 1 (c). It consists of two kinds of primary channels named as wet and dry channels.

![Figure 1. Schematic diagram of Maisotsenko Cycle for: (a) old M-Cycle, (b) modified M-Cycle, (c) psychrometric representation, and (d) sequential temperature decrement in wet-channel.](image-url)
Product Air Cooling arrangement keeps working and product dry channels separated. Partial Extraction of Air arrangement fractions a portion of the dry channel as usable product. Both arrangements are psychrometrically identical while arrangement in Figure 1 (b) with product channel has essential advantages as compared to arrangement in Figure 1 (a). Thus, any gas or liquid matter (for example hot waste product) can be used as a product flow in arrangement in Figure 1 (b) while arrangement in Figure 1 (a) is limited to the same gas for both channels. Moreover using the product channel allows reducing pressure drop in the system. Another advantage of arrangement in Figure 1 (b) compared with Figure 1a would be a possibility to use the product channel as a condenser for gas dehumidification [4].

For example, ambient air (1) is flowed into the dry-channel where it is sensibly cooled at constant humidity to cycle point (2) by transferring the heat to the wet-channel. The operational principle of M-Cycle is based on diverting the cooled air (2) to the wet-channel in order to use as working air. It results in subsequently decrement of effective dry-bulb (1 - 2a; 2b; 2c; 2) and wet-bulb (1w - 2a,w; 2b,w; 2c,w; 2dp) temperatures of the working air in the wet-channel as shown in Figure 1d. Sequential decrement of dry-bulb temperature in the wet-channel brings the effective wet-bulb temperature to be ideally equal to the dew-point temperature. Hence for an ideal heat transfer surface, the product air can be sensible cooled to the dew-point temperature of the ambient air. Moreover, saturated hot air (3) is rejected from the wet-channel equivalent to the evaporated water and recovered heat.

Thereby, M-Cycle HMX is suitable for all kind of air cooling (pre-cooling, inter-cooling and aftercooling). The basic scheme of transported gas cooling is represented in Figure 2. Gas is cooled in shell and tube heat exchanger. Water absorbs sensible heat which is transferred from the gas to the water through the tube walls. Hot water then goes to cooling tower, where its temperature decreases by direct evaporating process.

**Figure 2.** Conventional scheme of a compressed gas cooling.

Proposed cooling scheme of gas cooling via M-cycle is represented in Figure 3. The M-cycle utilizes the air potential energy available from latent heat of water evaporating into the air. The main advantage of proposed M-cycle cooling is possibility to get rid off cooling tower and closed cycle water supplying system. It is connected with the fact that M-cycle HMX required significantly less amount of water for cooling process than conventional cooling scheme shown in Figure 2.

Conceptual design of M-cycle HMX is represented in Figure 4. The predicted heat and mass transfer of the IEC could only be achieved on the condition of obtaining complete wetting of the wet channels surfaces. Poor water distribution on the surfaces would reduce the heat/mass transfer rate
between the primary air and the water film, and between the water film and secondary air, which would eventually undermine the effectiveness of the indirect evaporative cooler [5]. The most preferable way of water distribution system is applying superhydrophilic capillary porous medium with high moisture wicking and evaporation ability. A wide range of materials have been commonly used as the heat/mass exchanging medium. However, it is almost impossible to choose one all-mode material for various M-cycle heat exchanger applications. That is why research activities aimed to improve ability of wet channel surface to retain a water film become very important.

Posessing already installed cooling tower and sufficient amount of circulating water at the place where compressor works leads to considerable increase of M-Cycle HMXs operating effectiveness. This is connected with the phase change process in the wet channel that intensify heat transfer from hot compressed gas to working medium. For preliminary M-cycle HMX calculation it is proposed the following algorithm. It allows to estimate material and heat values in relevant (coresponding) points of M-cycle.

Figure 3. Scheme of compressed gas cooling via M-cycle.

There is pressure drop in product channel. Therefore:
\[ P_1 = P_2 + \Delta P_p \]  
(1)

where
\( P_1 \) and \( P_2 \) – absolute inlet and outlet pressure in product channel;
\( \Delta P_p \) – pressure losses in product channel.

Pressure of water in wet working channels should be equal to ambient air(gas) pressure at inlet working dry channel. Therefore:
\[ P_3 = P_3 \]  
(2)

where \( P_3 \) is water pressure and \( P_3 \) is absolute inlet pressure in dry working channel.

Absolute outlet pressure in working wet channel will be lower then inlet pressure due to pressure drop:
\[ P_4 = P_3 - \Delta P_w \]  
(3)

where
\( P_4 \) – absolute outlet pressure in working wet channel;
\( \Delta P_w \) – pressure losses in working channel.
The energy balance then would be:

\[ h_1 \cdot G_1 + h_3 \cdot G_3 + h_5 \cdot G_5 = h_2 \cdot G_2 + h_4 \cdot G_4 \]  

(4)

where

- \( h_1, \ G_1 \) and \( h_2, \ G_2 \) – gas enthalpy and massflow rate at the product channel inlet and outlet respectively;
- \( h_3, \ G_3 \) and \( h_4, \ G_4 \) – gas enthalpy and mass flow rate at the working channel inlet and outlet respectively;
- \( h_5, \ G_5 \) – water enthalpy and mass flow rate in M-cycle.

Material balance in apparatus can be written as:

\[ G_1 + G_3 + G_5 = G_2 + G_4 \]  

(5)

Material balance in product channel:

\[ G_1 = G_2 \]  

(6)

Material balance in working channel:

\[ G_2 = G_3 + G_5 \]  

(7)

Gas temperature in product channel outlet will be slightly larger then dew point of ambient working air:

\[ t_2 = t_{dp} + \Delta t_{dp} \]  

(8)

where

- \( t_2 \) – outlet product gas temperature;
- \( \Delta t_{dp} \) – temperature drop regarding the heat exchange imperfection (dew-point temperature unattainability);
- \( t_{dp} \) – dew-point temperature of ambient working air which is defined for pressure equal to partial pressure of water vapour \( P_{H_3} \). This pressure is defined by:

\[ P_{H_3} = P_3 \cdot r_{H_2O-3} \]  

(9)

where \( r_{H_2O-3} \) – mole fraction of water vapour at working channel inlet.

Outlet temperature in working channel \( t_4 \) is defined depending on the user-specified heat transfer at the ends of HMX:

\[ t_{dp} < t_4 = \max(t_1, \ t_3) - \Delta t_{in-out} < \max(t_1, \ t_3) \]  

(10)

where \( \Delta t_{in-out} \) – the temperature difference at the ends of the HMX, set by the user as a parameter.

Mole fraction of water vapour at working channel outlet:

\[ r_{H_2O-4} = \frac{P_H(t_4)}{P_4} \cdot \frac{\varphi_4}{100\%} \]  

(11)

where

- \( P_H(t_4) \) – water vapour partial pressure at working channel outlet;
- \( \varphi_4 \) – relative humidity of air at working channel outlet (set by user as a parameter in the range of 90%-100%).
Water mass flow rate $G_5$ is proportional to difference between absolute humidity ratio at working inlet (dry) and outlet (wet) channels. Therefore it can be expressed as:

$$G_5 = G_{w4} + G_{w3}$$  \hspace{1cm} (12)

where $G_{w4} + G_{w3}$ – working mass flow rate increment caused by water evaporation.

$$G_{w3} = G_3 \cdot m_{H_2O,3} = G_3 \cdot \frac{r_{H_2O,3} \cdot \mu_{H_2O}}{\mu_{mixture,3}}$$ \hspace{1cm} (13)

$$G_{w4} = G_4 \cdot m_{H_2O,4} = G_4 \cdot \frac{r_{H_2O,4} \cdot \mu_{H_2O}}{\mu_{mixture,4}}$$ \hspace{1cm} (14)

$$G_5 = (G_3 + G_4) \cdot \frac{r_{H_2O,4} \cdot \mu_{H_2O}}{\mu_{mixture,4}} - G_3 \cdot \frac{r_{H_2O,3} \cdot \mu_{H_2O}}{\mu_{mixture,3}}$$ \hspace{1cm} (15)

$$\frac{G_4}{G_3} = \frac{\frac{r_{H_2O,4} \cdot \mu_{H_2O}}{\mu_{mixture,4}} - \frac{r_{H_2O,3} \cdot \mu_{H_2O}}{\mu_{mixture,3}}}{1 - \frac{r_{H_2O,3} \cdot \mu_{H_2O}}{\mu_{mixture,3}}} = X$$ \hspace{1cm} (16)

Equation (16) establishes relation between $G_5$ and $G_4$. Required amount of working air could be defined as:

$$G_3 = \frac{G_4}{h_f - h_i} \cdot \left( h_i - h_x \right)$$ \hspace{1cm} (17)

3. M-cycle enhanced cooling tower concept

The total effectivenes of cooling system could be increased significantly by using advanced fill for cooling tower based on M-cycle [6]. Figure 4 shows comparison between conventional and advanced fills for cooling tower.

![Figure 4. Cooling tower: conventional design and advanced design.](image-url)

The lowest attainable water temperature using conventional fill is ambient air wet bulb temperature. Through an integrated design whereby the ambient air flow is in effect pre-cooled by...
itself, the “dew point” (M-cycle) approach reaches cooled water temperature level nearing the ambient
dew point temperature. This “dew point” approach, using advanced fill, provides higher performance
via the psychrometric principle - when a parcel of air is sensibly cooled, the saturated water vapor
pressure decreases, therefore, reducing the wet bulb temperature and increasing the evaporative
cooling potential down to the ambient air dew point temperature.

In additional, solid or liquid desiccants can be used in dry channels to dehumidify the air to
increase its latent heat potential before it is sent into the wet channels. This process will lower the dew
point attainable in the second stage of cooling. Also, continuous cooling of the desiccant by air
increases its absorption capacity and rate, thus drying the air faster. In humid environments, the
incoming ambient air should be heated so as to increase its saturation vapor pressure and latent heat
capacity [7].

A truly integrated precooling was theorized for air cooling [8] and then developed for water
cooling [9] by Maisotsenko et al., achieving dew point cooling without an initial external indirect-
direct evaporative precooling stage and with an integrated heat and mass exchanger (fill) optimized for
sensible precooling and pressure drop with a hybrid cross/counter flow arrangement.

Dew point cooling tower concept which is proposed by Maisotsenko [9] has numerous benefits,
comparing to traditional cooling techniques:
- Reduced capital and installation costs due to uselessness of additional external heat exchangers
  and/or cooling equipment (e.g. chillers).
- Extended performance in certain climate zones.
- Decreasing of cooling tower size and weight.
- Decreased evaporation losses, requiring less makeup water.
- Decreased air side pressure drop, due to absence of an external heat transfer equipment.

Critical to achieving these claims is the advanced fill, which integrates the air precooling and direct
evaporative cooling tower segments into a single heat and mass exchanger, bringing innovative
advancements in evaporative cooling to cooling towers. Integrating these stages into one system
realizes a positive feedback mechanism that cannot be realized with separate (external) air precooling.

It is feasible to achieve the benefits of integrated precooling-cooling tower operation in a truly
integrated and compact heat and mass exchanger, capable of effective heat and mass transfer. Unlike
direct evaporative or staged precooling technologies employing an initial direct evaporative precooling
step (e.g. [10, 11] concept), heat transfer within this advanced dew point cooling tower is driven by the
dew point temperature gradient, not wet bulb temperature. As such, only the absolute outside ambient
humidity is important, not the temperature of incoming air or water.

GTI and Idalex Technologies, Inc. are currently developing and optimizing the advanced fill for the
dew point cooling tower. The concept is shown in Figure 5, whereby the diagrams show the
integration of vertical dry precooling and wet direct evaporative channels in a unique flow
arrangement. This integrated method of achieving dew point cooling with a single heat and mass
exchanger, is referred to as the thermodynamic M-Cycle [12, 13]. In comparison to indirect-direct
technologies, researchers have demonstrated water and process air (fan power) reductions of 45% and
74% respectively are feasible with this method [14]. In this case, it is not absolutely necessary to
precool the incoming air outside of the fill, or use any multistage cooling [10, 11] in order to get the
dew point temperature level achieved with supplementary mechanical refrigeration. Instead, the air
flow can be specially arranged in the fill to be precooled by evaporating water, which in principle is
what drives this novel cycle.
In a regenerative evaporative cooler, dry and wet channels are placed close together in contact with each other. A fan draws outside ambient air into the dry channels. A second fan draws a part of the primary air coming out from the wet channels into the dry channels. The directions of air flow are opposite to each other in dry and wet channels. Fins are provided in order to enhance heat transfer between air and water [15]. The fluid is evaporatively cooled by outside air. The cooled fluid is then divided into two parts. The first part is circulated into the section to be cooled using a heat exchanger. The second part is again sensibly cooled using outside air to a lower wet bulb temperature without changing its humidity. The second stream is then mixed with the first air stream and recirculated into the section to be cooled. In this process, the fluid can be cooled to even lower than the wet bulb temperature and ideally to the atmospheric dew point temperature [16-19].

4. Conclusions
Compressor energy consumption can be reduced dramatically using M-cycle HMX for cooling. This HMX is not directly comparable to any existing heat-rejection/recovery cycle but it allows to cool any fluid, below the wet bulb temperature and approach the dew point temperature of the working air, without a compressor or refrigerant. Unlike conventional cooling towers, the COP of the Maisotsenko cooling tower increases with the increase in ambient air temperature which distinguishes its applicability in hot climate. So, the advanced heat rejection characteristics of M-cycle HMX can revolutionize the industrial gas cooling and heat rejection processes. Therefore, a lot of research work still needs to be done in order to find the optimum performances of M-Cycle for each application. Perspective application of the M-cycle is heat utilization during the dehydration of materials in multi-stage dryers [20] as well as granulation plants [21-24].

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