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INTERCOOLING DESICCANT-ASSISTED SYSTEM FOR COMPRESSION

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Abstract. Systems which makes use of desiccant technologies have been widely studied because of its economic and environmental advantages. A well-known feature of this type of cycle is that it's able to modify significantly the environmental conditions, performing an inlet air treatment by making use of pre-desiccant. Intercooling between compression stages is a common practice in gas power cycle design. The main objective is to reduce the required compression work between the ambient and turbine inlet pressures, thus unburdening the gas turbine and increasing the amount of available power. The intercooling effect is generally achieved by placing a heat exchanger between the compression stages, which rejects heat to the atmosphere and brings the air temperature back to the ambient level. The proposed system harvests this heat rejection to drive a desiccant cooling system. In turn, the cooling effect provided by the desiccant system is used to cool the air between the compression stages below the ambient level, allowing for a further reduction of the required compression work, when compared to standard intercooled cycles.

Keywords: evaporative cooling, desiccant dehumidification, air-conditioning, thermal design

1. INTRODUCTION

Intercooling is a common power augmentation technique in gas power systems. For significant pressure ratios, the use of multi-stage compression with intercooling allows for a substantial compression work reduction, when compared to single-stage compression Godish (2001). In addition to the improved power output, intercooling also increases the engine operation liability and reduces CO and NO_x emissions Straube (2009). Intercooling also plays a major role in diverse energy intensive compression process, such as in CO₂ capture system ASHRAE (1981). The system proposed by the present paper concerns the cooling of air at the compressor intake in a gas power system. This common practice aims at the increase of air density, which allows for an increase air mass flow rate throughout the system.

Mercer (2002) reported a relative increase of 10 to 15% in power output using evaporative coolers, and as much as 25% for mechanical refrigeration. Although evaporative cooling is the most prevalent technique for intake air cooling, it is not a universal solution. Amell and Cadavid (2002) showed that for climates with high average relative humidity, the vapor-compression refrigeration system offers a much more effective response than evaporative cooling. Alhazmy and Najjar (2004) reported that evaporative coolers can improve the cycle efficiency by as much as 3%, with favorable climate conditions (hot and dry). Vapor-compression systems, conversely, were reported to allow for an increased turbine output from 10 to 18%, depending on the atmospherically conditions, however claiming for compression work, which is detrimental for the cycle efficiency. The use of a desiccant-assisted evaporative cooler was previously suggested by Zadpoor and Golshan (2006), however with an analysis limited to manufacturer catalog data for the desiccant wheel output. The results showed that a desiccant-assisted system allow for a 3% increase in power output, when compared to a system without pre-cooling.

The use of a desiccant system to pre-cool intake air is readdressed in the present work, however considering a mathematical model to simulate the cooling and compression systems. As desiccant systems have been shown to enhance other processes, the purpose of this paper is to investigate the benefits of operating compression systems coupled with desiccants. The results and discussions presents two different types of curves with the proposal to compare traditional compression systems with the modified cycles.

2. PROBLEM DESCRIPTION AND FORMULATION

2.1 Cycle description

The cycle displayed in figure 1, is a desiccant inter-cooling scheme (DIC), which uses the desiccant cooling effect to provide the inter-cooling between compression stages.

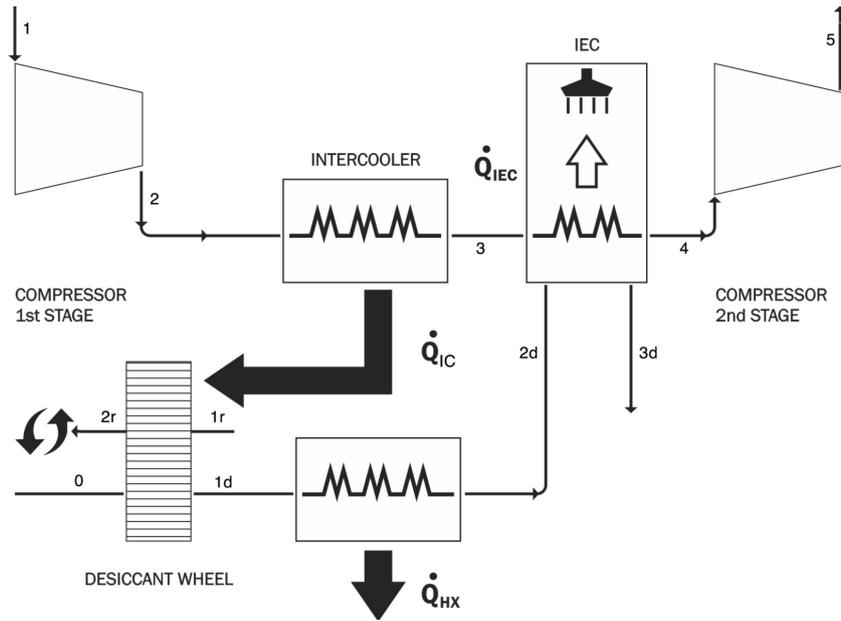


Figure 1. Desiccant inter-cooling scheme.

The numbering using for each state within each scheme is chosen so as to facilitate the modeling of arrangement in a unified fashion. Among the underlying simplifying assumptions that are considered in the forthcoming equations, the most important are given by:

1. Moist air is composed of a mixture of ideal gases (dry-air and water vapor);
2. A constant mass flow rate of dry air is used in the different branches of the proposed arrangement;
3. The concentration of water vapor in the gas mixture is small, such that the dry air concentration remains unchanged;
4. Temperature variations are moderate, such that specific heats can be assumed constant;
5. Only the steady-state operation is considered;
6. The effectiveness of the sensible heat wheel and evaporative coolers are independent of operating conditions;
7. The compression efficiencies are independent of the mass flow rates;
8. No condensation occurs within the compression stages;

2.2 Mathematical formulation

2.2.1 Compression stages

The first (low pressure) compression stage is given by the following equations:

$$T_2 = T_1 \left[\left((p_{Lo}^*)^{\frac{k-1}{k}} - 1 \right) \frac{1}{\eta_{c,Lo}} + 1 \right], \quad (1a)$$

$$Y_2 = Y_1, \quad (1b)$$

whereas the second (high pressure) stage is given by:

$$T_5 = T_4 \left[\left((p_{Hi}^*)^{\frac{k-1}{k}} - 1 \right) \frac{1}{\eta_{c,Hi}} + 1 \right], \quad (2a)$$

$$Y_5 = Y_4. \quad (2b)$$

The compressor efficiency is calculated in terms of the pressure ratio by the relation presented in Alhazmy and Najjar (2004); Korakianitis and Wilson (1992):

$$\eta_c = 0.91 - \frac{r_c - 1}{300} \quad (3)$$

For the desiccant inter-cooling scheme (DIC) $T_1 = T_0$, which is the ambient temperature.

The extensive and intensive (specific) rates of compression work for these stages are calculated as:

$$\dot{W}_{c,Hi} = \dot{m}_C w_{c,Hi}, \quad \text{with} \quad w_{c,Hi} = \tilde{c}_1 (T_5 - T_4) \quad (4a)$$

$$\dot{W}_{c,Lo} = \dot{m}_C w_{c,Lo}, \quad \text{with} \quad w_{c,Lo} = \tilde{c}_4 (T_2 - T_1) \quad (4b)$$

where \tilde{c}_j is a dry-basis specific heat at an arbitrary point j :

$$\tilde{c}_j = c_{p,a} + c_{p,v} Y_j \quad (5)$$

2.2.2 Sensible intercooler (IC)

The equations for the sensible intercooler are given by heat and mass balances across this device and a heat exchanger effectiveness relation:

$$C_C (T_2 - T_3) = \dot{Q}_{IC} = C_R (T_{1r} - T_0) \quad (6a)$$

$$Y_2 = Y_3, \quad (6b)$$

$$Y_{1r} = Y_0, \quad (6c)$$

$$\epsilon_{IC} = \frac{C_C}{C_{\min}} \frac{T_2 - T_3}{T_2 - T_0}, \quad (6d)$$

where $C_{\min} = \min(C_C, C_R)$, and the heat capacity rates of the compression and regeneration streams are given by:

$$C_C = \tilde{c}_1 \dot{m}_C, \quad (7a)$$

$$C_R = \tilde{c}_0 \dot{m}_R, \quad (7b)$$

The heat transfer rate \dot{Q}_{IC} is the energy removed by the sensible intercooler, which is used – for all desiccant-assisted schemes – to provide the heat required to for the regeneration of the desiccant wheel.

2.2.3 Desiccant wheel (DW)

For the desiccant wheel, the process is modeled using heat and mass balances across the two streams

$$\dot{m}_R (\tilde{v}_{1r} - \tilde{v}_{2r}) = \dot{m}_D (\tilde{v}_{1d} - \tilde{v}_0), \quad (8a)$$

$$\dot{m}_R (Y_{1r} - Y_{2r}) = \dot{m}_D (Y_{1d} - Y_0), \quad (8b)$$

and the algebraic correlation of Banks (1985a,b), which is given in terms of two performance parameters (η_1 and η_2) for a desiccant dehumidifier:

$$\eta_1 = \frac{F_1(T_{1d}, Y_{1d}) - F_1(T_0, Y_0)}{F_1(T_{1r}, Y_{1r}) - F_1(T_0, Y_0)}, \quad \eta_2 = \frac{F_2(T_{1d}, Y_{1d}) - F_2(T_0, Y_0)}{F_2(T_{1r}, Y_{1r}) - F_2(T_0, Y_0)}, \quad (8c)$$

where the functions F_1 and F_2 are given by:

$$F_1(T, Y) = -\frac{2865}{T^{1.49}} + 4.344 Y^{0.8624}, \quad F_2(T, Y) = \frac{T^{1.49}}{6360} - 1.127 Y^{0.07969}, \quad (8d)$$

and η_1 and η_2 are previously known. Typical values for these parameters can be found on several sources Sheridan and Mitchell (1985); Van den Bulck *et al.* (1985); Joudi and Dhaidan (2001); Zhang and Niu (2003); Panaras *et al.* (2007).

2.2.4 Sensible heat wheel (HW)

The heat wheel, a typical sensible heat exchanger, used in the desiccant inter-cooling, is accounted for by the following heat and mass balances:

$$C_D (T_{1d} - T_{2d}) = \dot{Q}_{HW} = C_E (T_{1e} - T_0), \quad (9a)$$

$$Y_{1e} = Y_0, \quad Y_{2d} = Y_{1d}, \quad (9b)$$

together with and the effectiveness relation:

$$\epsilon_{HW} = \frac{C_D}{C_{\min}} \frac{T_{1d} - T_{2d}}{T_{1d} - T_0}, \quad (9c)$$

in which $C_{\min} = \min(C_D, C_E)$, and the heat capacity rates of the dehumidification stream and external air stream (for cooling the air after dehumidification) are given by:

$$C_D = \tilde{c}_{1d} \dot{m}_D, \quad (10a)$$

$$C_E = \tilde{c}_0 \dot{m}_E, \quad (10b)$$

2.2.5 Indirect evaporative cooler (IEC) - Two-stage model

This model envisions the IEC as a direct single-stream evaporative cooler followed by a sensible heat exchanger. The first stage is modeled as a direct evaporative cooler with one stream, which is formulated as:

$$\tilde{v}_{2d} = \tilde{v}_{2d'}, \quad (11a)$$

$$\epsilon_{EC} = \frac{T_{2d} - T_{2d'}}{T_{2d} - T_{2d,sat}}, \quad (11b)$$

which are valid for both compression arrangements. Although a common model is valid for the first stage, different equations are required for the second stage, which is modeled as a sensible heat exchanger with effectiveness given by ϵ_{HX} . For the the desiccant inter-cooling scheme (DIC), this stage is given by the following equations:

$$C_C (T_3 - T_4) = \dot{Q}_{IEC} = C_{D'} (T_{3d} - T_{2d'}), \quad (12a)$$

$$\epsilon_{HX} = \frac{C_C}{C_{\min}} \frac{T_3 - T_4}{T_3 - T_{2d'}}, \quad (12b)$$

$$Y_3 = Y_4, \quad Y_{3d} = Y_{2d'}, \quad (12c)$$

where $C_{\min} = \min(C_C, C_{D'})$, and $C_{D'}$ is a modified dehumidification heat capacity rate due to the added moisture in the evaporative cooler:

$$C_{D'} = \tilde{c}_{d2'} \dot{m}_D. \quad (13)$$

2.2.6 Other relations

Some of the previous equations involve enthalpy values. These are written in terms of temperature and humidity ratio by employing the relation

$$\tilde{i} = i_{a,\text{ref}} + c_{p,a} (T - T_{\text{ref}}) + Y \left(i_{v,\text{ref}} + c_{p,v} (T - T_{\text{ref}}) \right), \quad (14)$$

in which $i_{a,\text{ref}} = 0$ kJ/kg, $i_{v,\text{ref}} = 2500.9$ kJ/kg, $c_{p,a} = 1.005$ kJ/kg°C, $c_{p,v} = 1.82$ kJ/kg°C, and the reference temperature is $T_{\text{ref}} = 0^\circ\text{C}$.

3. RESULTS AND DISCUSSION

The equations presented in the previous section are implemented in a simulation program using the *Wolfram Mathematica* system. The coupled system of non-linear equations was solved by an algebraic root finding scheme based on a secant method, carried out by the function FindRoot. As the solution is heavily dependent on the selected initial estimates, a progressive scheme for selecting initial guesses was employed.

The first set of results analyze the desiccant assisted inter-cooling scheme. Figure 2 displays the specific compression work ratio r_{cw} for varying pressure ratios ($p_{Lo}^* = p_{Hi}^*$) for different cycle components effectiveness and two different operating conditions ($T_{en} = 15^\circ\text{C}$ and $T_{en} = 35^\circ\text{C}$, $\phi_{en} = 50\%$). The curves presented in figure 2 shown that the work's compression decreases for the system with the inter-cooling applied between the compressors. In general, the system presents a reduction of the compression work of about 5%. There is a remarkable change in the curves related to inlet temperature for efficiencies of 85% and 100%. The compression ratio reduces asymptotically with the increase of ratio compression. Equal heat capacity flow rates were used for all four streams ($\dot{m}_C = \dot{m}_D = \dot{m}_R = \dot{m}_E$).

In order to illustrate the effects of the desiccant inter-cooling system from another perspective, entropy-temperature diagrams are plotted for components effectiveness of 85% and environment conditions of $T_{en} = 25^\circ\text{C}$ and $\Phi_{en} = 50\%$. Figure 3 shows that as the inter-cooling increases the efficiency of compression works tends to equalize. This is due to the importance that inter-cooling have in the system.

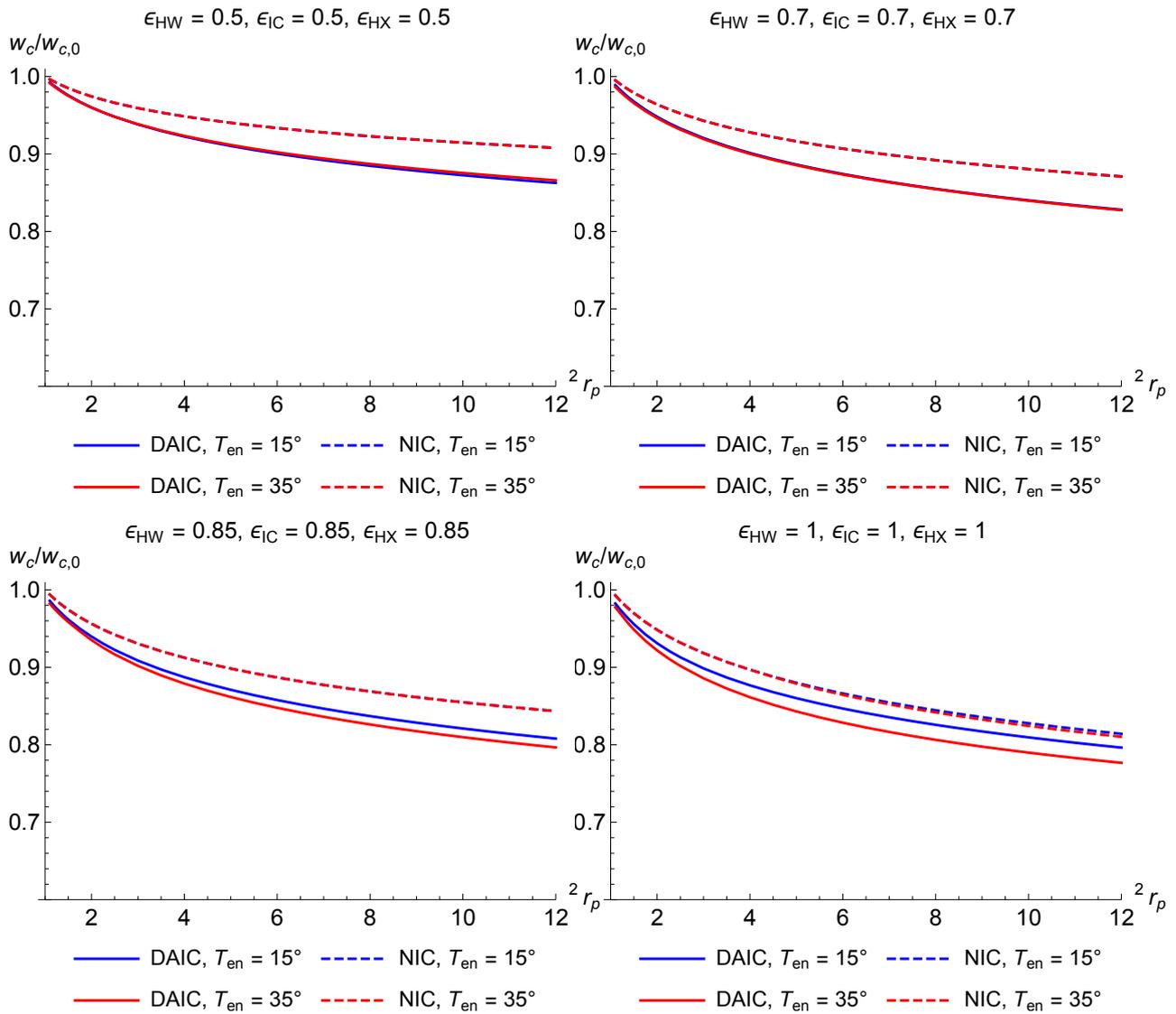


Figure 2. Ratio of compression work r_{cw} for different cycle configurations and two different operating conditions.

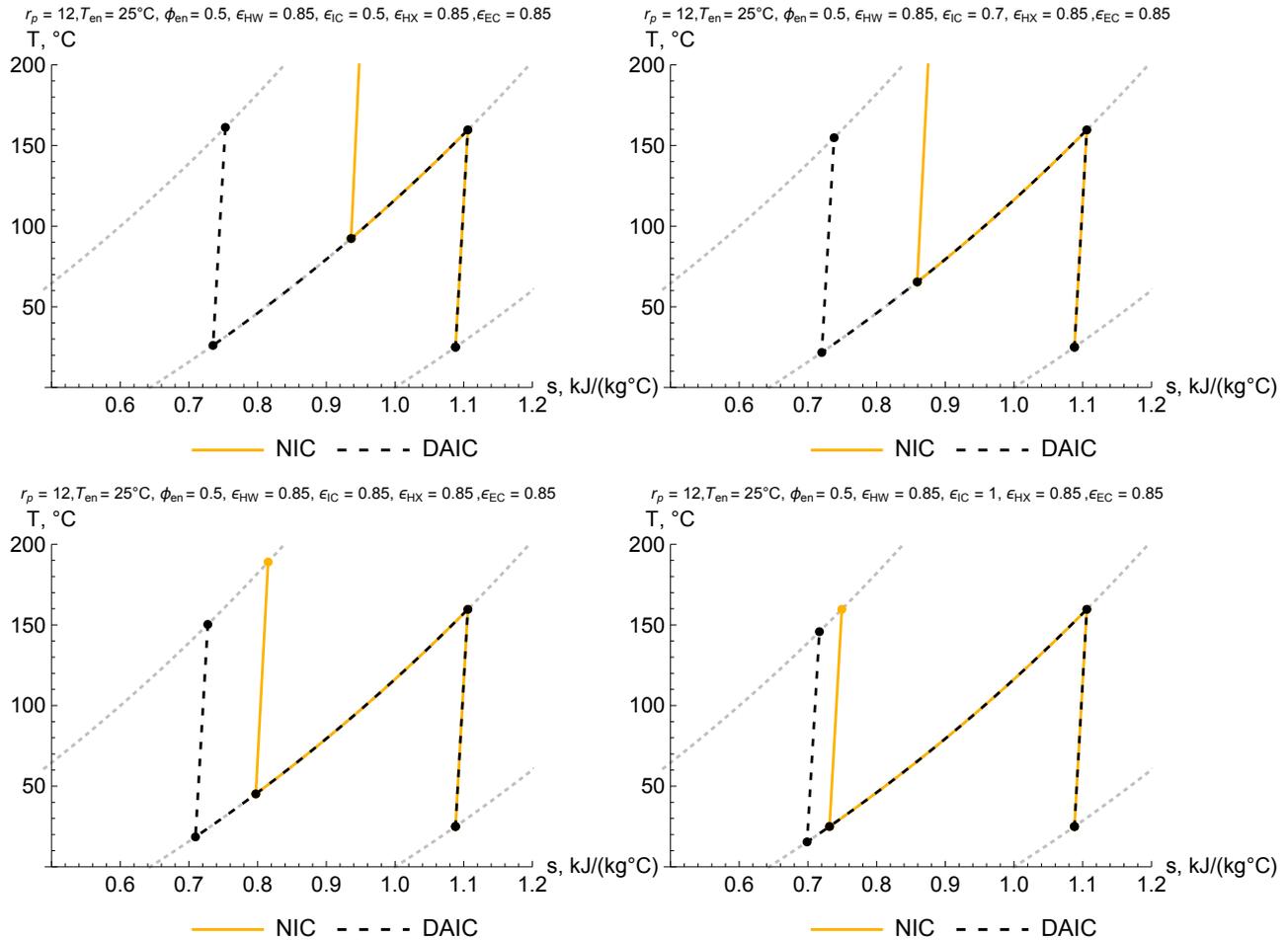


Figure 3. T-s curves that presents the variation's effect of inter-cooler's efficiency ϵ_{IC} .

4. CONCLUSIONS

This work presented a proposal for compression systems with desiccants coupled. $T-s$ curves were generated for this system and showed an overall reduction of the compression work for the second compressor. This may represent a considerable energy gain, depending on such equipment. The efficiency of inter-cooling shown that has a greater impact in the system. A second type of curve was generated for inter-cooling system. These curves showed a reduction in specific work of about 5% for the proposed system when compared to the traditional system.

5. ACKNOWLEDGEMENTS

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