Numerical Study of Water Production from Compressible Moist-Air Flow

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ABSTRACT

In this research a numerical study of water production from compressible moist-air flow by condensing of the vapor component of the atmospheric air through a converging-diverging nozzle is performed. The atmospheric air can be sucked by a vacuum compressor. The geographical conditions represent a hot and humid region, for example Bandar Abbas, Iran, with coordinates, 27° 11’ N and 56° 16’ E and summer climate conditions of about 40°C and relative humidity above 80%. Parametric studies are performed for the atmospheric-air temperature between, 40°C to 50°C, and relative humidity between 49.6% to 100%. For these ranges of operating conditions and a nozzle with the area ratio of 1.17, the liquid mass flow rates falls in the range 0.272 to 0.376 kg/s. The results show that, the energy consumed by the compressor for production 1 kg of water will be 1.279 kWh. The price of 1 kWh is 372 Rials, therefore the price for the production of 1 kg liquid water will be 475.8 Rials, therefore, the scheme is economically suitable.

Keywords: Water Production; Condensation of Moist-Air; Equilibrium thermodynamic; Roe’s scheme.

1. INTRODUCTION

The traditional sources of water in many regions of the world are surface water, ground water and rain water. Atmospheric air usually includes water vapor. One of the recent methods of water production is by condensation of the steam portion of the atmospheric air. Some studies are performed for water production from atmospheric air. Atmospheric water vapor processing (AWVP) is a new technology in which, water can be extracted from moist atmospheric air by phase change from vapor to liquid (Wahlgren, (2001)). Three methods have been described in the AWVP types, (a) heat pumps are used to cool surfaces so water vapor can be condensed and collected, (b) concentration of the water vapor by desiccants where water molecules are absorbed in a liquid or adsorbed on a solid, and (c) inducing convection currents in a tall tower structure, expanding the air, which transforms some of the energy in the air into work, thus cooling the air below its dew-point and causing some of the water vapor to condense into liquid water. Our numerical study in this paper is similar to method (c).

A scheme for large-scale dew collection as a source of fresh water production is studied by Rajvanshi (1981). In this research the cold seawater (5°C) is pumped from about 500 m depth and 5 km from the shore and passes through an onshore heat exchanger field to condense the water vapor of the atmospheric air. The results show that the scheme is not economically suitable due to high pumping power. Gandhidasan and Abualhamayel (2005) developed a mathematical model, based on the energy balance equations to find the condensation rate from the atmospheric air. Habeebullah (2009) developed a new model in which moist-air was cooled over evaporator coils of refrigeration machines for water extraction in hot humid regions.

In humid regions, atmospheric air (moist-air) usually includes a considerable amount of water vapor. Under the expansion processes, say through nozzles, the steam portion of the moist-air can condense and a second phase (liquid phase) forms. Computation of compressible moist-air flows through nozzles and other geometries using equilibrium thermodynamic model, have been studied by Hamidi and Kermani (2013a, 2013b), who compared the content of condensate generation in moist-air case with that of pure steam. It has been observed that the content of wetness at nozzles exit in the case of moist-air is more than that of pure steam under similar operating conditions. The reason is due to the internal flow of heat from steamportion toward air that accelerates the steam condensation rate. In this paper a numerical study of water production from atmospheric air in geographically humid regions is performed.
1.1 Problem Definition

The atmospheric moist-air is accelerated through a converging-diverging nozzle, and is discharged to a liquid/gas separating chamber, in which liquid water is collected from the bottom of the chamber, and a compressor and motor-pump assembly is installed at the top of the chamber to produce the required vacuum pressure of about 30 kPa (abs). Figure 1 shows the schematic of the apparatus. The results show that, the energy consumed by the compressor for production 1 kg of water will be 1.279 kWh. The price of 1 kWh is 372 Rials, therefore the price for the production of 1 kg liquid water will be 475.8 Rials, therefore, the scheme is economically suitable. The present computation is under the isentropic operation of the nozzle, since all of the aerodynamic and thermodynamic losses (see Kermani and Gerber, 2003), have not been included in the present study.

The present study is at its preliminary stages that provide a basis for the design and make of such a system to produce water from atmospheric-air. Some of the issues that will be addressed in future studies include the separation mechanisms of the liquid water from the gas phase and the shape of the downstream chamber. Computational domain used in the present study is a converging-diverging nozzle shown in Fig. 1.

Fig. 1. Schematic of water production facility.

In the present study, parametric studies are performed for the atmospheric-air conditions, 40°C ≤ T ≤ 50°C and relative humidity, 49.6% ≤ Θ ≤ 100%.

For these range of operating conditions and nozzle D (see Fig. 2), chosen from the Moore et al. (1973) paper having the area ratio of A_{out}/A_{throat} = 1.17, the liquid mass flow rates falls in the range 0.272 to 0.376 kg/s.

1.2 Detail of the Present Numerical Algorithm

The continuity, momentum and energy equations have been written in a fully conservative form for quasi one-dimensional flow through a converging-diverging nozzle. The gas portions of the two-phase mixture (water vapor + dry air) are assumed to obey the ideal gas equation of state. The flow is assumed to obey equilibrium thermodynamic model. The governing PDEs for both pure steam and moist-air are numerically solved. The detail of the numerical solution for the condensation of pure steam is given by Kermani et al. (2006), so they are not repeated here. The third-order upwind biased scheme of Roe (1981) has been used for spatial discretization, while for temporal integration the Lax-Wendroff second order scheme is implemented. The spurious numerical oscillations in the present high resolution computations are damped using the van Albanda flux limiter (1982). The expansion shocks have also been avoided using the entropy correction formula given by Kermani and Plett (2001).

![Fig. 2. Geometry of two different nozzles (A) and (D) taken from Moore et al. (1973) paper.](image)

2. Governing Equations

The governing equations for quasi one-dimensional, unsteady, inviscid and compressible flows are composed of the conservation laws of continuity, momentum and energy, and are shown in full conservative form. In the absence of body forces one can write (Hoffmann and Chiang, 1993):

\[ \frac{\partial Q}{\partial t} + \frac{\partial F}{\partial x} + H = 0, \]  

\[ Q = A \begin{bmatrix} \rho \\ \rho u \\ \rho v \\ \rho u^2 \\ \rho v^2 \\ \rho u v \\ \rho h_u \\ \rho h_v \\ \rho h \\ \rho h_n \\ \rho h_t \end{bmatrix}, 
\]  

\[ F = A \begin{bmatrix} \rho u \\ \rho u^2 + P \\ \rho u v \\ \rho u h_u \\ \rho u h_t \end{bmatrix}, 
\]  

\[ H = \begin{bmatrix} 0 \\ -P \frac{dA}{dx} \\ 0 \end{bmatrix}. \]  

Here, Q, F and H are respectively, the conservative vector, the flux vector and the source term. A is the cross-sectional area of the nozzle, pis the mixture density (= \( P_{mix} = \rho_a + \rho_v \)), \( \rho_a \) and \( \rho_v \) are the density of the steam (vapor + liquid) and air respectively and \( \rho \) is the velocity. In this study the slip velocity between the gas and the liquid phases is ignored. This is a reasonable assumption that will be explained in detail in results. In Eqn. (2), \( P \) is the mixture pressure (= \( P_{mix} = P_a + P_v \)), where \( P_a \) and \( P_v \) are the vapor and air pressures respectively. For the present low pressure computation, the ideal gas equation of state is of sufficient accuracy, hence:

\[ P_a = \rho_a R_a T, \quad R_a = 287 \text{ J/kg.K}, \]

\[ P_v = \rho_v R_v T, \quad R_v = 461.4 \text{ J/kg.K}. \]

Here, \( \rho_a \) and \( \rho_v \) are the density of the air and vapor respectively. In wet regions, if the volume of liquid phase is ignored (this is a correct assumption whose accuracy will be discussed later in detail in results), one can write, \( \rho_s = \rho_a/\chi \), where \( \chi \) is the
quality of the steam. It is noted that in this study the steam is referred to the mixture of liquid water and vapor. \( e_t \) and \( h_t \) are respectively the total internal energy and total enthalpy of the mixture (steam + air): 

\[
e_t = \frac{m_a}{m_{mix}} e_a + \frac{m_x}{m_{mix}} e_x + \frac{u^2}{2} m_{mix},
\]

(4) 

\[
h_t = e_t + P/p.
\]

(5) 

The humidity ratio of the moist-air (\( \omega \)) is defined as: 

\[
\omega = \frac{m_x}{m_a},
\]

(6) 

where \( m_a \) is the mass flow rate of the vapor (the dry portion of the steam) and \( m_x \) is the mass flow rate of the dry air. Up to the condensation onset, the humidity ratio along the nozzle remains constant which is equal to humidity ratio of the reservoir: 

\[
\omega_{res} = \frac{m_x}{m_a} = \frac{m_x}{m_a} = \text{constant}.
\]

(7) 

Up to the condensation onset, \( m_x = m_{a} \), beyond the condensation onset \( m_x < m_a \), since the steam is condensed and second phase (liquid) is formed, therefore the mass flow rate of vapor ( \( m_x \)) is reduced. The enthalpy of evaporation \( (h_{fg}) \) of the steam portion is obtained from: 

\[
h_{fg} = e_{fg} + R_a T,
\]

(8) 

where \( e_{fg} \) is the latent internal energy. A second-order polynomial can accurately represent the relationship between \( e_{fg} \) and \( T \), and the coefficients are provided by Kermani et al. (2006). The internal energy of the vapor and air, respectively \( e_a \) and \( e_v \), are determined by assuming a constant value for the specific heats: 

\[
e_a = c_{va} T, \quad c_{va} = R_a (\gamma_a - 1), \quad \gamma_a = 1.4.
\]

(9) 

Equation (9) can be used to obtain the saturated liquid internal energy: 

\[
e_f = e_a - e_{fg}, \quad e_g = c_{vg} T_{sat}.
\]

(11) 

In dry regions the internal energy of the steam is obtained from, \( e_x = e_c = e_{vg} T \), while in wet regions: 

\[
e_x = e_f + \chi e_{fg}.
\]

(12) 

Substituting Eqs. (7) and (12) in Eqs. (4) and (5), the total internal energy and total enthalpy of the mixture (moist-air) in wet regions are obtained as follows: 

\[
e_t = \frac{1}{1 + \omega_{res}} e_a + \frac{\omega_{res}}{1 + \omega_{res}} (e_f + \chi e_{fg}),
\]

\[
e_x = \frac{u^2}{2}, \quad e_a = c_{va} T,
\]

(13) 

\[
h_t = \frac{1}{1 + \omega_{res}} e_a + \frac{\omega_{res}}{1 + \omega_{res}} (e_f + \chi e_{fg})
\]

In this study, the entropy of the flow is computed in a similar manner that applied for internal energy and enthalpy. The entropy of air is obtained using the temperature and partial pressure of air: 

\[
\phi_a = c_{p,a} \ln T - R_a \ln P_a,
\]

(15) 

\[
\phi_{a,a} = \gamma_a R_a (\gamma_a - 1).
\]

Similarly, the entropy of the steam portion in dry regions is obtained using the temperature and partial pressure of vapor: 

\[
\phi_x = \phi_{a,a} + \chi \phi_{fg},
\]

(16) 

while in wet regions: 

\[
\phi_x = \phi_f + \phi_{fg}.
\]

(17) 

Finally the total entropy of the mixture (moist-air) is obtained as follows: 

\[
\phi_t = \frac{m_a}{m_{mix}} \phi_a + \frac{m_x}{m_{mix}} \phi_x = \frac{1}{1 + \omega_{res}} \phi_a + \frac{\omega_{res}}{1 + \omega_{res}} \phi_x.
\]

(18) 

3. NUMERICAL SOLUTION

3.1 Time and Space Discretization

For the time discretization, an explicit two-step predictor-corrector scheme has been used to march the solution from time level \( n + 1 \) to \( n + 1 \) (Tannehill et al. (1997)): 

\[
Q_i^{n+1} = Q_i^n - 0.5 \Delta t \left( \frac{F_{ei}^n}{h^2} - \frac{F_{ei}^{n+1}}{h^2} \right) - 0.5 \Delta t \frac{2}{h^2},
\]

(19) 

where, \( i \) corresponds to any arbitrary internal nodal, and \( F_{ei}^n \) and \( F_{ei}^{n+1} \) are the numerical fluxes evaluated at the east (E) and west (W) faces of the control volume (Tannehill et al. (1997)): 

\[
F_{ei} = \frac{1}{2} \left( F_{E,i} + F_{E,i+1} \right) - \frac{\lambda}{2} \sum_{k=1}^{3} \left[ k_f \phi_{L,k} \phi_{E,k} \right] F_{E,i}.
\]

(20) 

\[
F_{W,i} = F_{E,i-1}.
\]

(21) 

where \( \lambda \) is the eigenvalue of the Jacobian matrix, \( T \) represents the corresponding eigenvector, \( \phi \) is the wave amplitude vector and \( A \) is the cross-sectional area of the nozzle. In Eqn. (20), \( k \) corresponds to each wave propagating in the x–t domain. The predictor step is followed by the corrector step, which gives a central difference around \( n + 1/2 \):
\[ Q^{n+1} = Q^n - \frac{\Delta t}{\Delta x} \left( F_{E,i}^{n+1} - F_{W,i}^{n+1} \right) \]

For the spatial discretization, a third order upwind-biased method based on the Roe average and the conservative vector \( Q \), the primitive variables \( P, \rho, T, u, \ldots \) etc. We use the primitive variables \( P, \rho, T, u, \ldots \) in dry regions, but it is noted that pressure and temperature are not independent in wet regions, so the relationship \( \rho \) is extrapolated to the cell faces instead of pressure in wet regions. The local pressure of steam in the wet regions is determined from the saturated pressure at the local mixture temperature (Khakbaz Baboli and Kermani, 2008). The following third-order upwind-biased scheme is used here:

\[ q_{E,i}^n = q_i + \frac{1}{4} [(1 - k_0)\Delta w q_i + (1 + k_0)\Delta e q_i] \]

\[ q_{W,i}^n = q_{i+1} - \frac{1}{4} [(1 - k_0)\Delta w q_i + (1 + k_0)\Delta e q_i] \]

The density and temperature in wet regions are determined from:

\[ \rho_E = p_E^b \rho_E^b \]

where the density and enthalpy of the moist-air in wet regions are determined from:

\[ \rho_E^b = (\rho_a)_{E}^b + (\rho_v)_{E}^b \]

The density of air(\( \rho_a \)) is determined from \( \rho_a = P_a/(\gamma_a T) \), while the density of the vapor in the wet regions is calculated from \( \rho_v = P_{sat}/(\gamma_v T) \), that \( P_{sat} \) is obtained vs. \( T \). (see Khakbaz Baboli and Kermani, 2008).

### 3.3 Moisture Evaluation

The conservative vector \( Q \), at each time level provides values of \( Q_1, Q_2, Q_3 \) = \( \rho A, \mu A, \rho e A \) where \( \rho \) is the density of the moist-air (air + steam). Removing the density of the air portion from the mixture, the density of the steam can be determined. Then, can be calculated from \( u = Q_2/Q_3 \), and \( e_{mix} = e - u^2/2 \). Now by removing the internal energy of the air portion from that of the mixture, the internal energy of the steam can be obtained. Finally, the thermodynamic state and moisture content (if present) can be determined using a trial and error process from two independent properties of steam, namely internal energy and density, \( e_\alpha \) and \( \rho_\alpha \), respectively.

### 3.4 Boundary Conditions

In this study, two nozzle geometries (nozzles A and D) are taken from the Moore et al. (1973) paper. The geometries of these nozzles are shown in Fig. 2. The inflow is assumed to be subsonic, where the stagnation pressure \( P_{in} \), stagnation temperature \( T_{in} \), and the humidity ratio of the reservoir \( \alpha_{res} \) are specified. At the inlet of the nozzle the flow properties have been determined from the following conditions: mixture pressure \( P_{mix} \) and the steam quality \( \gamma \) are extrapolated from the interior domain to the inlet face, and the entropic condition from the upstream stagnation conditions to the inlet face is enforced. At the exit, the flow is supersonic in which all of the primitive variables are extrapolated from the interior domain.
4. Validation and Results

4.1 Validation

For validation purposes two nozzle geometries are taken from the Moore et al. (1973) paper, namely, nozzles A and D, the geometry of which are shown in Fig. 2. Nozzle A of these series has the highest expansion rate while nozzle D possesses the lowest. Due to the availability of experimental data from the Moore et al. (1973) paper for pure steam, the accuracy assessment of the present computations is performed vs. these experimental data. Comparisons with the Moore et al. data are performed by two different solvers: (i) a pure steam solver and (ii) a moist-air solver in which the humidity ratio of the reservoir is set to a very large number, i.e. $\omega_{res} \to \infty$. The case (ii) corresponds to a pure steam case too and is expected to echo the same results as case (i). In the case (ii) $\omega_{res}$ is set to 1000. Figure 3 (a) shows the results of numerical solutions for pressure distribution along the MooreA and MooreD nozzles for both of the cases (i) and (ii) mentioned above. As shown in this figure, a good agreement between the results is obtained, where the maximum errors (deviation of the computed results from the experimental data) for MooreA and MooreD nozzles have been obtained as 12% and 10%, respectively. Also the wetness fraction profiles along the these nozzles for cases (i) and (ii) have also been compared with those of Kermani et al. (2006), and identical results were achieved, that is shown in Fig. 3 (b).

In the case of low humidity ratio, $\omega \to 0$ as the moist-air tends to dry air, the shock tube data is used to validate our numerical solution (not shown here). The results show that (see Hamidi and Kermani 2013a, 2013b), an excellent agreement between the numerical results and exact solution is obtained. Therefore, the numerical method which is proposed in this paper is capable to simulate compressible flow problems with satisfactory precision.

4.2 Results

Sample of the computed results are shown in this section. We limit the results to MooreD nozzle, but similar results for other nozzles can also be obtained. Figure 4, (a) and (b) show, respectively, the profiles of wetness fractions, and liquid mass flow rates along the MooreD nozzle for both pure steam and moist-air flows. In the case of moist-air the nozzle inflow stagnation conditions were taken as $T_0,\omega=323.15$ K (50$^\circ$C) and $P_0,\omega=1$ atm, where the computations were performed at two $\omega_{res}=0.086$ and 0.04, while pure steam computations were performed at $T_0,\omega=323.15$ K, $P_0,\omega=0.122$ and 0.0604 atm. It is noted that to be able to correctly compare the moist-air and pure steam results, the stagnation pressure of steam at the nozzle inlet in the case of pure steam is adjusted to the stagnation partial pressure of steam at the nozzle inlet. As shown in Fig. 4, the wetness fraction and liquid mass flow rate of moist-air case are much higher than those of pure steam. This is due to the fact that the specific heat of steam is greater than that of air hence there will be a local reversible removal of heat from steam toward the air. As a result, in the moist-air case the wetness fraction and liquid mass flow rate at the nozzle exit are much higher. For example, as shown in Fig. 4 (a) in the case of moist-air flow with, $\omega=100\%$, wetness fraction 35.96% at the nozzle exit is achieved. Similarly, in the case of moist-air with $\omega=100\%$ the condensate production rate, $m_\text{cond}$ at the nozzle exit is 0.376 kg/s, as shown in Fig. 4 (b). While in the case of pure steam at similar conditions ($T_0,\omega=323.15$ K, $P_0,\omega=0.122$ atm) wetness fraction 4.66% and $m_\text{cond}$ = 0.06 kg/s were obtained at the nozzle exit. That is, 6.27 times more condensate is produced in the case of moist-air.

Using this terminology, for the case of moist-air with $\omega_{res}=0.086$, $P_{0_v}$ is determined as 0.122 atm, and similarly for $\omega_{res}=0.04$, $P_{0_v}=0.0604$ atm.
gas species along the nozzle at the conditions, gas and liquid phases will still remain valid. Figure negligible. Hence the no-slip condition between the fraction of the liquid water to that of the mixture remain consistent with the assumptions that the volume fraction of the liquid water to that of the mixture is negligible, and the no-slip condition between the gas and liquid phase.

Moore, nozzle with supersonic outflow

$$T_{0, \text{in}} = 323.15 \text{[K]}, \rho_{0, \text{in}} = 0.986 & 0.04 (\theta = 100 \% & 49.6\%)$$

Also from the computed values of Fig. 5, this is a self-consistency check of the solution. Also from the computed values of Fig. 5, $\frac{m_{\text{mix}}}{m_{l, \text{out}}} = 0.612 \text{kg/s}$ is obtained. Hence $\frac{m_{l}}{m_{\text{mix}}} = 0.366/13.2 = 0.0277$, which is pretty consistent with the assumptions that the volume fraction of the liquid water to that of the mixture is negligible, and the no-slip condition between the gas and liquid phase.

Fig. 5. Mass flow rate profiles of air portion $(m_{l, \text{air}})$, steam portion $(m_{l, \text{steam}} = m_{l, l} + m_{l, s})$ and moist-air mixture $(m_{\text{mix}} = m_{l, l} + m_{l, s})$ along the MooreD nozzle for supersonic outflow.

Figure 6 shows the $T$-$s$ diagram for the expansion of moist-air and its components along the nozzle. As shown in this figure the moist-air (air + steam) undergoes an isentropic expansion $(s_{\text{mix}} = \text{constant})$, while the components (air and steam) experience non-isentropic processes due to the reversible heat exchange between the phases. The direction of heat is from the steam with higher specific heat toward air with lower specific heat. As a result of the internal heat removal from the steam, the entropy of steam$(s_{l, s})$ decreases and the entropy of the air $(s_{l, a})$ increases along the nozzle. Quantitative values of temperature-entropy are also included in this figure for comparison purposes.

Fig. 6. $T$-$s$ diagram for the isentropic expansion of the moist-air mixture along the MooreD nozzle, the mixture undergoes an isentropic expansion while due to internally reversible heat exchange between the air and steam the species experience non-isentropic processes.

Figure 7 (a) and (b), respectively, represent a...
parametric study to illustrate the influence of inflow humidity ratio (relative humidity) on wetness fraction and water production rate along the nozzle. As shown in Fig. 7 (a) by increasing the humidity ratio the wetness fraction at the nozzle exit decreases, since by increasing the humidity ratio the mass fraction of air reduces. As a result the capacity of air as the recipient source of heat from the steam portion reduces too. On the influence on the rate of water production, by increasing the humidity ratio the water production rate increases, since the mass fraction of steam portion of the moist-air mixture increases.

(a)

MooreD nozzle with supersonic outflow

(b)

MooreD nozzle with supersonic outflow

Table 1 summarizes the results of the parametric studies performed in this research. It shows the mass flow rate of water produced at the nozzle exit which shown in Fig. 1. For the range of atmospheric-air conditions $40 \leq T \leq 50$$\deg C$ and relative humidity $49.6\% \leq \Phi \leq 100\%$, and the MooreD nozzle with area ratio of $A_{\text{out}}/A_{\text{throat}} = 1.17$, the mass flow rate of water falls in the range 0.272 to 0.376 kg/s. The highest rate of 0.376 kg/s corresponds to the saturated-air state ($\Phi = 100\%$).

4.3 Sample Calculation and Feasibility Study for Desalination Application

As an application for the present study, desalination of liquid water from moist air in highly humid regions is considered. As a sample calculation the content of 1 kg produced liquid water and consumed energy is determined. Applying the first law of thermodynamics around the compressor (see Fig. 1), we can write:

$$Q_{\text{in}} - W_{\text{c,v}} = \sum m_i (h_i + \frac{V_i^2}{2} + gZ_i) - \sum m_i (h_i + \frac{V_i^2}{2} + gZ_i)$$

We assume that changes in kinetic and potential energy are negligible and the compressor to be adiabatic:

$$W_{\text{c,v}} = m_{\text{in}} C_{\text{p,mix}} (T_{\text{in}} - T_2)/\eta_{\text{sc}}$$

Where $\eta_{\text{sc}}$ is the compressor isentropic efficiency, taken as 0.85. This calculation is performed for nozzle D with 10 cm$^2$ throat cross section and inlet atmospheric condition: $T_{\text{in}} = 323.15 \ [K], P_{\text{in}} = 1 \ [\text{atm}]$ and $\Phi = 93.5\%$. The noted atmospheric condition corresponds to Case # 2 in Table 1. It is noted that the mass flow rate of the components (air, vapor and liquid) are given in Fig. 5. The specific heats ($C_{\text{p,mix}}, C_{\text{v,mix}}$) and the specific heat ratio ($\gamma_{\text{mix}}$) of the dry air and vapor (step 2 to 3 in Fig. 1), are determined by:

| Table 1 Summary of the results: the mass flow rate of water at the MooreD nozzle exit at ambient pressure 1 atm and prescribed temperature and humidity ratio |
|---|---|---|
| Case # | $T_{\text{in}}$ | $\omega_{\text{rel}}$ ($\%$) | $m_{\text{w}}$ (kg/s) |
| 1 | 323.15 K | 0.086 (100) | 0.376 |
| 2 | 323.15 K | 0.080 (93.5) | 0.366 |
| 3 | 323.15 K | 0.070 (83.0) | 0.347 |
| 4 | 323.15 K | 0.060 (72.2) | 0.326 |
| 5 | 323.15 K | 0.050 (61.0) | 0.302 |
| 6 | 323.15 K | 0.040 (49.6) | 0.272 |
| 7 | 318.15 K | 0.065 (100) | 0.360 |
| 8 | 318.15 K | 0.060 (92.9) | 0.348 |
| 9 | 318.15 K | 0.055 (85.8) | 0.336 |
| 10 | 318.15 K | 0.050 (78.6) | 0.323 |
| 11 | 318.15 K | 0.045 (71.3) | 0.307 |
| 12 | 318.15 K | 0.040 (63.8) | 0.291 |
| 13 | 313.15 K | 0.065 (100) | 0.340 |
| 14 | 313.15 K | 0.060 (92.6) | 0.328 |
| 15 | 313.15 K | 0.055 (82.9) | 0.310 |
\[
C_{p,\text{mix}} = \sum_{i=1}^{2} C_i C_{p,i} = \frac{0.612}{12.22 + 0.612} \times 1.872 + \frac{12.22 + 0.612}{12.22} \times 0.75 = 1.0454 \text{kJ/kg.K}
\]
\[
y_{\text{mix}} = \frac{C_{p,\text{mix}}}{C_{p,\text{dry}} + y_{\text{water}}} = 1.394,
\]

Where \( C_i \) is the mass fraction of the vapor or air. At the compressor inlet (nozzle exit), \( P_2 = 33.82 \text{kPa} \), \( T_2 = 293.9 \) (see Fig. 6) and at the compressor outlet, \( P_3 = 101.325 \text{kPa} \). For the ideal, isentropic process from 2 \( \rightarrow \) 3 in Fig. 1, we can write:
\[
\frac{T_{3,3}}{T_2} = \left( \frac{P_3}{P_2} \right)_{\text{mix}}^{\gamma - 1} \Rightarrow \frac{T_{3,3}}{T_2} = \left( \frac{101.325}{33.82} \right)^{\frac{1.394-1}{1.394}} = 0.612
\]
The mass flow rate of the liquid water is:
\[
\dot{m}_{\text{water}} = 12.832 \text{ (kg/s)} \times \frac{10 \text{ cm}^3}{635 \text{ cm}^3} = 0.2021 \text{ kg/s}
\]
Therefore the power consumed by the compressor will be:
\[
|W_{2-3}| = 0.2021 \times \frac{10454}{(400.7 - 293.9)} \times 0.85 = 26.55 \text{ kW}.
\]
The mass flow rate of the liquid water is:
\[
\dot{m}_{\text{water}} = 0.366 \text{ (kg/s)} \times \frac{10 \text{ cm}^3}{635 \text{ cm}^3} = 0.005764 \text{ kg/s}
\]
Therefore the time for production of 1 kg water is:
\[
t = \frac{1}{0.005764} = 173.5 \text{ sec.}
\]
The energy consumed by the compressor for production 1 kg of water will be:
\[
W_{\text{comp}} = 26.55 \times \frac{173.5}{3600} = 1.279 \text{ kWh}
\]
The price of 1 kWh is 372 Rials, therefore the price for the production of 1 kg liquid water (almost 1 Lit.) will be:
\[
\text{Price} = 1.279 \text{ kWh} \times 372 \text{ Rials} = 475.8 \text{ Rials}.
\]

These results show that the scheme is economically suitable.

5. CONCLUSION

Condensation phenomena for flows of pure-steam and moist-air through converging-diverging nozzles are numerically studied. The main focus was on the fluid mechanics of the flow and the rates of the liquid condensate (water) produced. The task is performed using a high resolution flux difference splitting scheme of Roe (1981) with a spatially third order and temporally second order accurate algorithm. The flow is assumed to obey equilibrium thermodynamic model. In order to assess the accuracy of the present computations the results are compared with the experimental data of Moore et al. (1973) for pure steam case (\( \omega \rightarrow \infty \)). Comparisons show good agreement between the results. The following conclusions can be drawn:

1. For the isentropic process of moist-air studied here, it is observed that each of the species, steam and air experience non-entropic processes due to the heat flow from a species with higher specific heat value (steam) toward the other species with lower specific heat value (air here).

2. As a result of item 1 above, in the case of moist-air, the content of wetness at the nozzle exit becomes much higher than that of pure steam case.

3. The idea mentioned in items 1 and 2 above can be generalized as follows. When the mixture of steam and an additive gas of lower specific heat values (like air, carbon dioxide, oxygen or an appropriate combination of them) are used to produce liquid water condensate, the content of liquid water at the nozzle exit can be much higher than the case of pure steam. In the present study, the additive gas is dry air with mixture being atmospheric-air in hot and humid climates.

4. The results show that, the energy consumed by the compressor for production 1 kg of water will be 1.279 kWh. The price of 1 kWh is 372 Rials, therefore the price for the production of 1 kg liquid water will be 475.8 Rials. These results show that the scheme is economically suitable.

5. Future Work: As described in this study, in the case of moist-air, a significantly higher condensate is generated, but, the separation mechanisms of the liquid water from the gas phase and the shape of the downstream chamber should be studied.

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