THEORETICAL ANALYSIS OF A NOZZLE CONDENSATION- DRYING CYCLE

تحليل نظري لدورة تجفيف- تكثيف بالبوق

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الخلاصة

الطاقة المطلوبة في عمليات التجفيف الصناعية بالمجففات التقليدية تتم بدون القدرة على استعادة الحرارة الكامنة لبخار الماء. وفي محاولة لاستعادة الحرارة الكامنة لبخار الماء تم اقتراح دورة تجفيف مغلقة تستخدم بوق لاستعادة هذه الطاقة من الهواء الرطب. هذه الدورة تتكون من : المجفف, البوق، الناشر, مجفف ماء, المبادل الحراريّ وضاغط.

يظهر تحليل هذه الدورة أنه في أحسن الأحوال أنها تتطلب طاقة حرارية أقل بالمقارنة بدورة تجفيف مفتوحة مثالية أو دورة تجفيف بالمضخة الحرارية. في الحالة المثالية أظهرت النتائج أن تكلفة الطاقة لهذه الدورة ثلث تكلفة الطاقة تقريبًا لدورة التجفيف المفتوحة، كما أن الدورة يمكن أن تعمل عند ظروف مختلفة تسمح بعمل تصميم واحد يُستَغلّ لاستخدامات التجفيف المختلفة . ومن الميزات الأخرى لهذه الدورة هي استخدامها في تجفيف سوائل خطيرة من منتج

Abstract

The energy required for industrial drying processes consumed by a conventional dryers are essentially thrown away without the ability to recover the latent heat of water vapor in the air, significant reductions in dryer energy usage cannot be made. In an attempt to recover the latent heat of the water vapor, a closed drying cycle is proposed, which uses a nozzle as the means to recover this energy from moist-saturated air. This nozzle condensation drying cycle (NCDC) is composed of the following components: dryer, nozzle, diffuser, air-water separator, heat exchanger and compressor. Analysis of this drying cycle shows that ideally this cycle requires lower energy compared with an ideal open cycle heated dryer or a heat pump dehumidification cycle. However, when equipment isentropic efficiencies are included, the energy costs are approximately a third of the cost of the ideal open cycle heated dryer. The cycle can be operated at varying conditions, allowing for a single design to be utilized for various applications. Other advantages are in applications that require drying of hazardous liquids from a product.

Keywords: Drying; Recovery; Heat of vaporization; Nozzle; Condensation.

Nomenclature

a_{2ig}	Sonic velocity at state 2 assuming an ideal gas, (m/s)
h_i	Specific enthalpy at state i, *(kJ/kg)*
h_C	Specific enthalpy of condensate, *(kJ/kg)*
h_{ai}	Specific enthalpy of air at state i, *(kJ/kg)*
h_{f/T_1}	Specific enthalpy of saturated liquid evaluated at T_1, *(kJ/kg)*
h_{vi}	Specific enthalpy of water vapor at state i, *(kJ/kg)*
$\dot{m}_{air}$	Mass flow rates of air, *(kg/s)*
$\dot{m}_{cond}$	Mass flow rates of condensate, *(kg/s)*
P_i	Pressure at state i, *(kPa)*
$\dot{Q}$	Rate of heat transfer, *(kJ/s)*
q_{air}	Specific heat transfer per unit mass of dry air, *(kJ/kg)*

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q_{cond} Specific heat transfer from the heat pump cycle's condenser per unit mass of dry air, *(kJ/kg)*

q_{evap} Specific heat transfer from the heat pump cycle's evaporator per unit mass of dry air, *(kJ/kg)*

q_{heat} Specific heat input required for an open heated dryer per unit mass of condensate, *(kJ/kg$_{cond}$)*

R specific universal gas constant, , *(kJ/kg.K)*

s_i Specific entropy at state i, *(kJ/kg.K)*

s_c Specific entropy of condensate, *(kJ/kg.K)*

T_i Temperature at state i, *(K)*

ΔT_{pinch} Temperature differential at evaporator and condenser pinch points, °C

u_{f/T_1} Specific internal energy of saturated liquid evaluated at T_1, *(kJ/kg)*

V_2 Velocity at state 2, (m/s)

w_{air} Specific work required by the compressor per unit mass of dry air, *(kJ/kg$_{air}$)*

w_{cond} Specific work required by the compressor per unit mass of condensate, *(kJ/kg$_{cond}$)*

k Specific heat ratio

η_{comp} isentropic efficiency of the compressor

η_{diff} isentropic efficiency of the diffuser

η_{noz} isentropic efficiency of the nozzle

ω_i Specific humidity at state i, *(kg/kg$_{air}$)*

1.Introduction

Of the total energy required for many manufacturing processes, a significant portion is used for industrial drying processes, estimated at 12 % [1]. Often these industrial dryers operate without any provisions for heat recovery [2]. This means that almost all the energy consumed by the dryer is essentially thrown away. Without the ability to recover the latent heat of the water vapor in the air, significant reductions in dryer energy usage cannot be made.

Additionally, many applications require that the drying process is performed over a range of temperatures and for various amounts of water removal. Moraitis and Akritidis [2] cite an example in cereal facilities where the temperature must be varied between 120 and 45°C, and the amount of water removed varies between 0.025 and 0.005 kg water/kg dry air for different types of grains. For applications where operating parameters are varied, any method to recover the latent heat of the water vapor must be capable of adapting to these varying operating conditions.

Present research will examine the operation of a closed drying cycle that can recover the latent heat of the water vapor. Therefore, this cycle is operated without any means of external heat addition. The only energy addition for this new drying cycle is electrical energy supplied to a compressor. Additionally, this cycle can operate over a wide range of operating conditions without modification.

2. Existing technologies

The most widely recognized method of drying is by a heated open cycle. For this cycle, ambient air is heated by passing the air over electric coils or through a flame. The hot air is passed over a wet product, here the energy, which was added to the air during heating, is transferred to the water in the wet product. This energy transfer provides the water with the necessary heat of vaporization to evaporate the water. This evaporated water (water vapor) mixes with the air and is carried out of the drying area. This psychometric mixture (moist air) can be exhausted to the environment. By this means, water is removed from a wet product and expelled to the surroundings. A disadvantage to this open cycle is that the added heat is exhausted to the environment and not recovered, essentially, it is thrown away. Additionally, the performance of the cycle is dependent upon the atmospheric conditions and will change with changes in humidity and / or temperature.

An example of a closed cycle that is used for drying is a heat pump dehumidification cycle. In this cycle, the moist air exiting the dryer is passed through the evaporator of a heat pump cycle. In the evaporator, heat is transferred from the air to the heat pump cycle's refrigerant, cooling the moist air. Cooling the moist air causes the water vapor to condense. The air exits the evaporator at a lower temperature and a lower humidity. This air is often passed through the condenser where heat is transferred from the refrigerant to the air. The air exits the condenser at a high temperature and humidity, where it can no enter the dryer. This closed cycle does not require an external heat source; however, energy is required to run the heat pump's compressor. Disadvantages of this cycle are the capital costs associated with a heat pump cycle (i.e., condenser, evaporator and compressor) and the inconvenience of having to use refrigerants.

Nozzle closed drying cycle the expansion of moist air in a nozzle provides the water-condensation means for a novel drying cycle. The nozzle closed drying cycle (NCDC) shown in Fig. 1 is composed of the following six components: dryer, nozzle, diffuser, air-water separator, heat exchanger and compressor. Tracing the flow of the moist air through the components provides an understanding of steady state operation of the cycle.

Hot of moist air with a low of relative humidity enters the dryer. Water within the dryer (provided the wet product) evaporates. This evaporation cools the moist air. The saturated air enters the nozzle where it is expanded to a high velocity and a lower temperature and pressure. As a result of the lower temperature and pressure, condensation from the moist air occurs, assuming thermal equilibrium. The two-phase mixture, moist air and condensate, enters the diffuser where the mixture is slowed to stagnation conditions, assuming frozen flow. Discussion of these thermal equilibrium and frozen assumptions are provided in the next section. The mixture enters the separator where the condensate is rejected. At that point in the cycle the moist air exists at a lower pressure and a higher temperature than the state at which the air enters the dryer. Therefore, Heat is rejected by means of heat exchanger to lower the temperature of the flow. Finally the air enters a compressor to increase the pressure of the air to the state at which it enters the dryer.

The need to have a heat exchanger (external heat rejection) in a drying cycle may seem counterproductive. However, if the entire cycle is looked at from a conservation of energy perspective, the need to have this heat rejection is shown. Fig. 1 shows that energy is being input into the cycle by the compressor and by the enthalpy of the water in the wet product entering the dryer. Without the heat exchanger, the only energy that would be exiting the cycle would be the enthalpy of the condensate. When the cycle operates at steady state, the sum of the energy entering equals the sum of the energy exiting. In general, the enthalpy of the condensate exiting the cycle does not equal the sum of the compressor work and the enthalpy of the water entering the dryer. Therefore, the heat exchanger must be included in the cycle for it to operate at steady state.

The nozzle in the cycle provides a means of heat recovery, which allows the cycle to operate with no external heat addition. In the dryer, the relatively dry air provides the heat of vaporization to the water in the wet product, causing evaporation and a reduction in the air temperature. In the nozzle, water condenses from the moist air allowing the air to recover its lost heat of vaporization. However, condensation within the nozzle induces a stagnation pressure loss, requiring a compressor for the cycle [3].

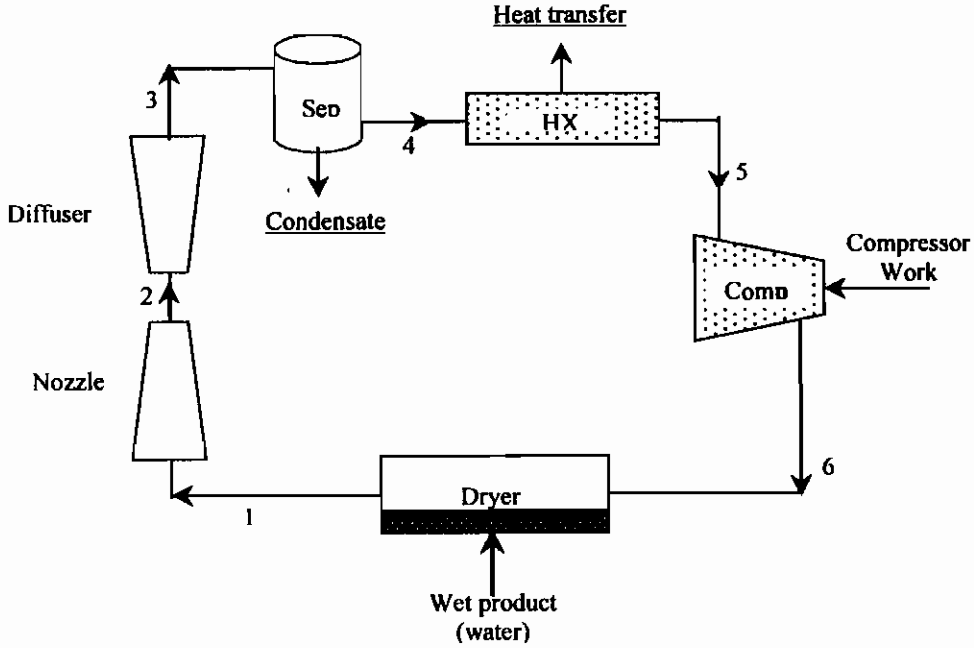

Fig. 1. Schematic of nozzle condensation- drying cycle

3. Analysis

3.1. Governing equations

The dryer is modeled as an adiabatic saturation process. This process is assumed to occur at constant pressure with saturated exit conditions [4]. At steady state, the water entering the control volume is assumed to be a saturated liquid at the same temperature as the moist air leaving the dryer. For these assumptions the energy equation on a per unit mass of dry air basis is:

$$h_1 - h_6 = (\omega_1 - \omega_6) h_{f/T_1} \qquad (1)$$

$$h_i = h_{ai} + \omega_i h_{vi} . \qquad (2)$$

For this analysis, the dryer will be treated as a continuous process with water (wet product) being continuously supplied. However, the dryer can be modeled as a batch process such as a clothes dryer.

The mixture at the exit of the nozzle is assumed to be composed of saturated moist air and condensate in thermal equilibrium. According to Wegener [5], thermal equilibrium can be assumed for condensation within a nozzle when aerosols are present in the flow, such that heterogeneous nucleation of the condensate can occur. It is assumed that the industrial or residential applications, for which this cycle is used, will have sufficient particles, i.e. dust, lint, etc. that the mixture will be in thermal equilibrium. The energy equation for the process from state 1 to state 2, assuming no slip between the condensate and the air is:

$$h_1 = h_2 + (\omega_1 - \omega_2) h_C + (1 + \omega_1) \frac{V^2_2}{2} . \qquad (3)$$

In classical compressible duct flow of an ideal gas without condensation, the flow is assumed to be isentropic. An equivalent isentropic analysis for this flow is made by assuming no entropy generation, yielding the following entropy conservation equation:

$$S_1 = S_2 + (\omega_1 - \omega_2) S_C . \tag{4}$$

Eqs. (3) and (4) along with the saturated assumption, define an isentropic state 2. For the remainder of this paper the subscript "s" will denote isentropic state. A nonisentropic saturated state 2 at the same pressure, as state 2s is determined from the nozzle efficiency, defined by:

$$\eta_{noz} = \frac{V^2{}_2}{V^2{}_{2s}} = \frac{h_1 - h_2 - (\omega_1 - \omega_2) h_c}{h_1 - h_{2s} - (\omega_1 - \omega_{2s}) h_{2s}} . \tag{5}$$

The analysis of the diffuser is similar to the analysis of the nozzle, except that the flow is assumed to be thermodynamically frozen and the moist air exiting the diffuser is not saturated. Assuming frozen flow in the diffuser treats the condensate as if it is isolated from the moist air. Therefore, the specific humidity of the moist air is determined to be constant through the diffuser from a conservation of mass analysis. The condensate is assumed to remain at constant temperature with no evaporation through the diffuser. The reason that the flow acts as if it is frozen, which is because of the evaporation of the droplets, is limited by the kinetics of the evaporation process. The time that a water droplet take to travel through the diffuser is assumed to be small compared with the time that it would take to evaporate. This argument is supported by the following observations. Wegener reports that through an entire wind tunnel circuit, ice clusters and/or water droplets can exist [5]. Additionally, Boerrigter notes that when condensation occurs in the H3 wind tunnel at the von karmen Institute water droplets accumulate in the settling chamber [6]. Based upon these observations, the assumption that the flow is frozen through the diffusion process appears to be a good approximation. Non-frozen flow will be looked at in a later section.

The conservation of entropy, diffuser efficiency, and conservation of energy equations for the diffusion process from state 2 to state 3s and state 3, are as follow:

$$S_{3s} = S_2 \tag{6}$$

$$\eta_{diff} = \frac{h_{3s} - h_2}{h_3 - h_2} \tag{7}$$

$$h_2 + (1 + \omega_1) \frac{V_2^{\,2}}{2} = h_3 \tag{8}$$

In the air-water separation process all of the condensate is assumed to be removed. According to Talavera [7], gas / liquid separators can have gas pressure drops as low as one half of an inch of water. However, the type of separator used and therefore, the pressure drop, will vary according the application. Therefore, for simplicity, it is assumed that the separator removes all of the condensate and that the pressure, temperature, and specific humidity of the moist air remains constant from state 3 to state 4.

The final two components of the cycle, the heat exchanger and compressor, are modeled by the standard thermodynamic assumption for each component. The pressure and specific humidity across the heat exchanger are assumed to be constant, the relative humidity of the moist air entering the heat exchanger is low enough (less than 20% for most cases) so that condensation in the heat exchanger will not occur. The specific humidity across the compressor is constant and the exit state from the compressor is related to the isentropic exit state by an isentropic efficiency, η_{comp}. The specific heat transfer from the heat exchanger and the specific work required by the compressor, on a per unit mass of dry air basis, are given as follow:

$$q_{air} = \frac{\dot{Q}}{\dot{m}_{air}} = h_4 - h_5 \tag{9}$$

$$w_{air} = \frac{\dot{W}}{\dot{m}_{air}} = h_6 - h_5 \tag{10}$$

The mass flow rate of water removed (condensate) per unit mass of dry air is given by:

$$\frac{\dot{m}_{cond}}{\dot{m}_{air}} = \varpi_1 - \omega_2 \qquad (11)$$

Dividing Eq. (10) by Eq. (11) yields a convenient form of the work equation, the work required per unit mass of water removed,

$$w'_{cond} = \frac{\dot{W}}{\dot{m}_{cond}} = \frac{w_{air}}{\dfrac{\dot{m}_{cond}}{\dot{m}_{air}}} \qquad (12)$$

3.2. Solution technique

The proceeding equations, as a result of the necessary of including air and steam/water properties, cannot be solved explicitly. The use of the computer program Engineering Equation Solver (EES)[10] provides the means of numerically solving the system of equations. EES has air and steam table functions that allow the user to inelude the thermodynamic properties with the equations. EES then iterates until all equations and properties are satisfied.

The system of equations with the specified assumptions has 3 degrees of freedom, allowing three parameters to be chosen. Three logical design parameters are P_6, T_6 and V_2.. P_6, the pressure at the entrance of the dryer and in the dryer, is prescribed by the design of the dryer. T_6, is the temperature to which the product will be exposed. V_2, the velocity of the mixture exiting the nozzle, is determined by the design of the nozzle.

4. Results and discussion

4.1. Base case

For a base case, P_6 is taken as101.3 kpa and T_6 is taken as 70^{0}C. The value of P_6 was chosen because of the fact that for many applications it may be desirable to have atmospheric pressure in the dryer. The velocity at the nozzle exit (V_2) is prescribed by the design of the nozzle. An illustrative velocity determined by assuming an ideal gas at state 2:

$$a_{2ig} = \sqrt{k\,RT_2}\,.$$

For a pure ideal gas, setting V_2 equal to a_{2ig} represents a Mach number of unity. However, the flow at state 2 is a mixture of condensate and moist air, which means that the flow is not a pure ideal gas. Therefore, the Mach number at state 2 may not be unity, but setting V_2 equal to a_{2ig} *provides an* illustrative veloeity for the base case. This veloeity is varied parametrically in the next section.

The results for this base case are shown in Table 1 for various values of η_{noz}, η_{diff} and $\eta_{comp.}$. The isentropic efficiencies are varied between unity and typical values for each particular component. Specifically, the nozzle effieiency is varied from 1.0 to 0.9, while the diffuser efficiency and the compressor efficiency are varied from 1.0 to 0.8 and 1.0 to 0.6, respectively [8].

The first row of Table 1 shows the results for the ideal cycles, i.e. all isentropic efficiencies equal to unity, and the Table 1.1 show the results of temperature, pressure and specific humidity for different points of cycle at all isentropie efficiencies equal to unity. The Table 1 shows that the cycle's ideal energy requirement is 186 kJ/kg$_{cond.}$ For comparison, the energy requirements for an open cycle heated dryer are shown in Table 2 for various ambient temperatures and relative humidities. However, the energy input to the NCDC is from a compressor, which is electrically powered. The 186 kJ/kg$_{cond}$ does not take into account the efficiency of the power plant that produces the electricity. The energy source for the heated dryer can be a directly first heated source, which has small energy losses. Comparing the average annual cost of electricity to the cost of natural gas, as reported by the department of Energy's Information Administration, shows that the cost of electricity is approximately five times greater than the cost of natural gas for commercial customers [9].

The remaining rows of Table 1 show the effect of isentropic efficiencies on the cycle. These results show that the performance of the NCDC departs greatly from the ideal as isentropic efficiencies are decreased from unity. The nozzle and diffuser efficiencies have the greatest effect on the performance, while the compressor efficiency has a relatively small effect on the cycle. The case for which

$\eta_{noz} = 0.95, \eta_{diff} = 0.9 \, and \, \eta_{comp} = 0.8$ may be a good approximation of the performance of an actual cycle. For this case, the work required is 993KJ/kg_{cond}. Note that the values in Table 2 are not taken into account any burner efficiencies or energy losses. Therefore, comparing the non-ideal NCDC to the ideal open cycle heated dryer is a conservative comparison.

Another comparison of interest is to compare the NCDC with a heat pump dehumidification cycle. Table 3 shows the amount of work and heat transfer required by a heat pump cycle to remove the same amount of condensate as the NCDC's base case. The condenser and evaporator heat exchanger performance was simply modeled using the nearest temperature approach for the two fluids, or pinch point. A comparison of Table 1 with Table 3 shows that the ideal heat pump cycle, $\eta_{comp} = 1.0 \; and \, \Delta T_{pinch} = 0\,^{\circ}C$ requires almost three times the energy requirement and more than 20 times the total heat transfer requirement than the ideal NCDC. Since both these cycles would be electrically powered, a direct comparison of energy requirement is correct.

The latter part of table 3 shows the effect of non-ideal compressor and non-ideal heat exchangers (i.e. finite ΔTs at pinch points). The last two rows of the table give a good idea as to the requirements for actual cycles. The energy requirements for these two cases are of the same order as the NCDC's requirement with

$\eta_{noz} = 0.95, \eta_{diff} = 0.9 \; and \; \eta_{comp} = 0.8$.

However, these values are not taken into account the work required to overcome the air side pressure drop through the heat exchangers. Therefore, this comparison is also a conservative comparison. Additionally, the heat transfer requirement for the heat pump cycle is more than six times greater than the NCDC's requirement.

Table 1
Results of the analysis of the NCDC at the base case
(P_6 =101.3kPa, $V_2 = a_{2ig}$=334m/s) for various isentropic efficiencies

η_{noz} %	η_{diff} %	η_{comp} %	q_{heat} (kJ/kg_{air})	$\frac{\dot{m}_{cond}}{\dot{m}_{air}} \left(\frac{kg_{cond}}{kg_{air}}\right)$	q_{heat} (kJ/kg_{cond})
100.00	100.00	100.00	4.01020	0.01510	186.321
95.00	100.00	100.00	7.49860	0.01500	422.028
90.00	100.00	100.00	11.40000	0.01480	687.235
100.00	90.00	100.00	9.81200	0.01510	565.000
100.00	80.00	100.00	15.51000	0.01510	945.600
100.00	100.00	80.00	4.71130	0.01510	232.470
100.00	100.00	60.00	5.90210	0.01510	308.103
95.00	90.00	80.00	16.10000	0.01500	993.000
90.00	80.00	60.00	35.60000	0.01480	2,315.000

Table 1-1

Point	Temperature K	Pressure kPa	*Specific humidity* kg/kg_{air}
1	306.5	101.3	0.03378
2	277	41.32	0.03378
3	328.5	69.5	0.0189
4	328.5	69.5	0.0189
5	308	69.5	0.0189
6	343	101.3	0.0189

Table 2

Energy required for an open cycle heated dryer to heat air to 70^0C and the amount of condensate removed for various ambient temperatures and relative humidities

dry bulb temperature for ambient air °C	Relative humidity for ambient air (%)	q_{heat} (kJ/kg_{air})	$\frac{\dot{m}_{cond}}{\dot{m}_{air}}\left(\frac{kg_{cond}}{kg_{air}}\right)$	q_{heat} (kJ/kg_{cond})
10	30.00	60.60000	0.017821	3,400.482576
10	40.00	60.70000	0.017658	3,437.535394
10	50.00	60.70000	0.017493	3,469.959412
10	60.00	60.90000	0.017426	3,494.777918
10	70.00	61.00000	0.017257	3,534.797473
15	30.00	55.70000	0.017631	3,159.208212
15	40.00	55.80000	0.017467	3,194.595522
15	50.00	55.90000	0.017200	3,250.000000
15	60.00	56.00000	0.017029	3,288.507839
15	70.00	56.10000	0.016754	3,348.454100
20	30.00	50.70000	0.017445	2,906.276870
20	40.00	50.80000	0.017080	2,974.238875
20	50.00	51.00000	0.016808	3,034.269395
20	60.00	51.10000	0.016530	3,091.349062
20	70.00	51.40000	0.016200	3,172.839506
30	30.00	40.80000	0.016650	2,450.450450
30	40.00	40.90000	0.016200	2,524.691358
30	50.00	41.40000	0.015700	2,636.942675
30	60.00	41.50000	0.015300	2,712.418300
30	70.00	41.70000	0.014800	2,817.567567
35	30.00	35.90000	0.016200	2,216.049382
35	40.00	36.20000	0.015600	2,320.512820
35	50.00	36.20000	0.015000	2,413.333333
35	60.00	36.60000	0.014500	2,524.137931
35	70.00	37.00000	0.013900	2,661.870503

Table 3
Work and heat transfer requirements for a heat pump dehumidifier cycle, using R22, to remove the same amount of condensate as the drying cycle's base case

η_{comp}	$\Delta T_{pinch}\,(^0C)$	$w_{cond}\,(\frac{kJ}{kg_{cond}})$	$q_{evap}\,(\frac{kJ}{kg_{air}})$	$q_{cond}\,(\frac{kJ}{kg_{air}})$
1.0	0.0	531	45.3	52.8
1.0	2.5	610	45.3	53.9
1.0	5.0	693	45.3	55.1
1.0	7.5	781	45.3	56.4
0.8	0.0	631	45.3	54.3
0.8	2.5	724	45.3	55.6
0.8	5.0	820	45.3	56.9
0.8	7.5	922	45.3	58.4

Fig.2 and 3 summarizes the specific work required per unit mass of condensate and the costs of operating the cycles, including the cost of open heated dryer cycle heated wit electric resistance h and fuel. Using the assumption that the electrical energy cost for the NCDC is five times that of the gas fuel for the heated dryer, the cost of the energy for the NCDC is of the same order as the energy costs for a heated dryer. Therefore, ideally, the cost of using the NCDC is approximately a third of the cost of the ideal open cycle heated dryer.

4.2. Effect of dryer inlet temperature

Figs. 4 show the effect of dryer inlet temperature (T_6) on the specific work required per unit mass of condensate (w_{cond})for NCDC cyclee. This figure shows that w_{cond} decreases with increasing temperature. Fig. 5 and 6 shows that the specific work required per unit mass of condensate (w_{air}) and the mass flow rate of the condensate increase with increasing temperature. Since both w_{air} and condensate flow rate increase, a trade-off occurs. Increasing the temperature requires more work (w_{air}) , but more water is rejected. As shown in Eq. (12), w_{cond} is directly proportional to w_{air} and inversely proportional to the flow rate of condensate. Therefore, the decrease of w_{cond} with increasing temperature indicates that the condensate flow rate increase faster than w_{air} . however, at higher temperatures, w_{cond} decreases slowly, while w_{air} and the condensate flow rate increase rapidly. This trend shows that the rates at which w_{air} and condensate flow increase must be approximately equal at higher temperatures. The discontinuities in the curves at approximately 50^{0}C occur as a result of the condensate freezing in the nozzle.

4.3. Effect of drying process pressure

The effect of the drying process pressure (P_6) variations on w_{cond}, w_{air} and condensate flow rate are shown. Fig. 7 shows that w_{cond} increase almost linearly with increasing pressure. Fig 7 shows that the condensate flow rate decreases linearly with increasing pressure, while w_{air} is relatively constant (decreasing slightly). Again a trade-off occurs between w_{air} and the condensate flow rate. Increasing the pressure requires less work, however, less condensate forms. The increase in w_{cond} in Fig. 6 indicates that the condensate flow rate decreases faster (with increasing pressure) than w_{air}.

4.4. Effect of the nozzle exit velocity

Figs. 10 to 12 give the effect of variations of the nozzle exit velocity (V2) on w_{cond}, w_{air} and the condensate flow rate. The figures show that w_{cond}, w_{air} and the condensate flow rate increase with increasing velocity. Increasing the velocity requires more work (w_{air}), but more condensate is rejected. Since w_{cond} increases with increasing velocity, w_{air} must increase faster than the condensate flow rate. to this assumption, the cycle was analyzed, assuming that some percentage of the condensate re-evaporates in the diffuser. The relatively small percentages of re-evaporation, the increase in the work required is also relatively small.

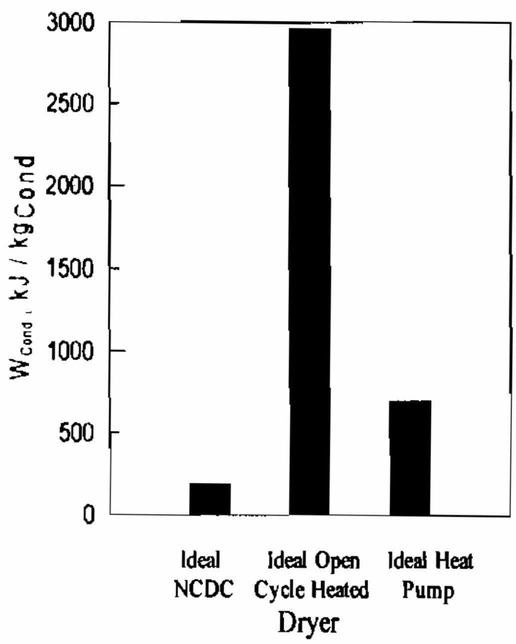

Fig. (2) comparison of specific work required per unit mass of condensate of the NCDC, open heated dryer and heat pump dehumidification cycle.

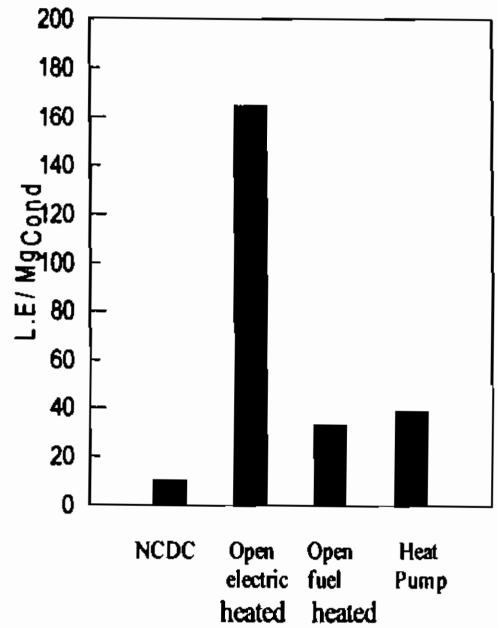

Fig. (3) comparison of operating costs of the NCDC, open heated dryer and heat pump dehumidification cycle.

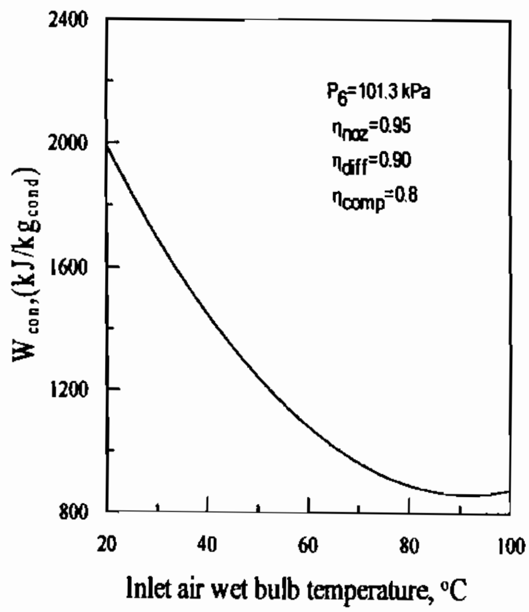

Fig. (4) Effect of dry inlet temperature on specific work required per unit mass of condensate.

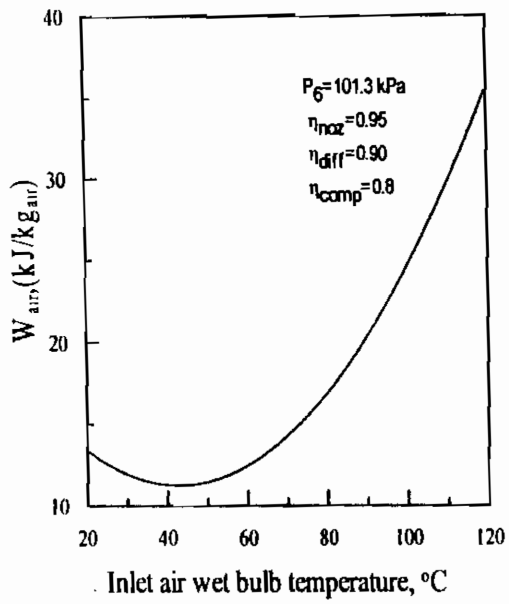

Fig. (5) Effect of dry inlet temperature on specific work required per unit mass of dry air.

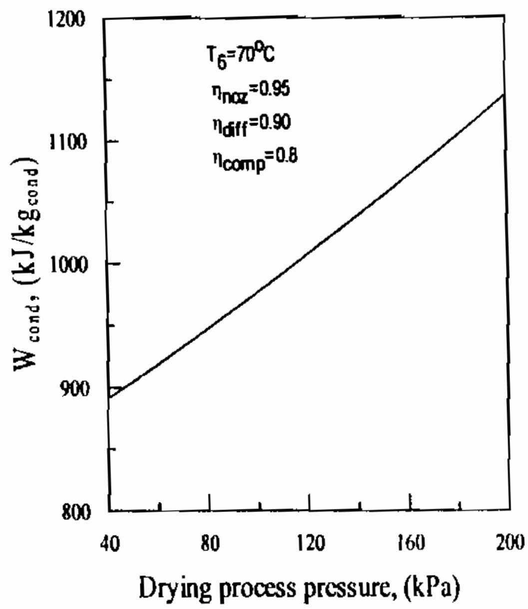

Fig. (7) Effect of drying process pressure on specific work required per unit mass of condensate.

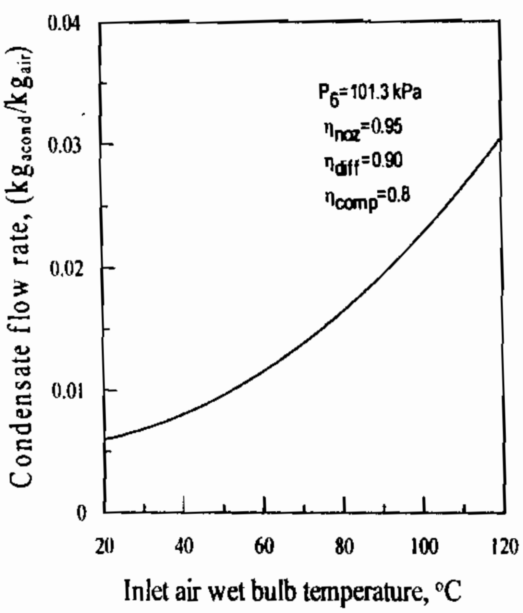

Fig. (6) Effect of dry inlet temperature on condensate flow rate.

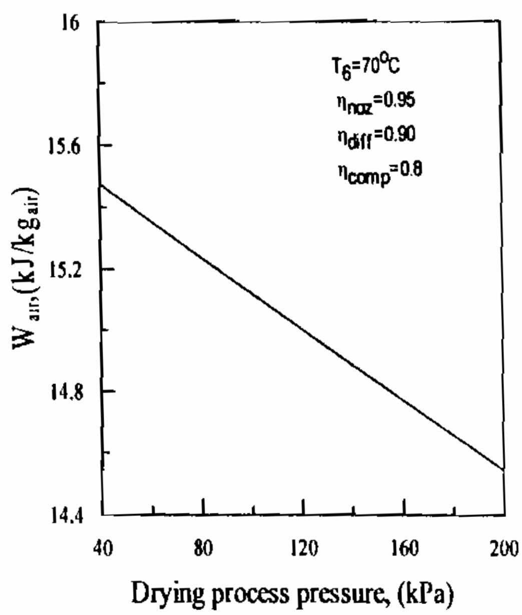

Fig. (8) Effect of drying process pressure on specific work required per unit mass of dry air.

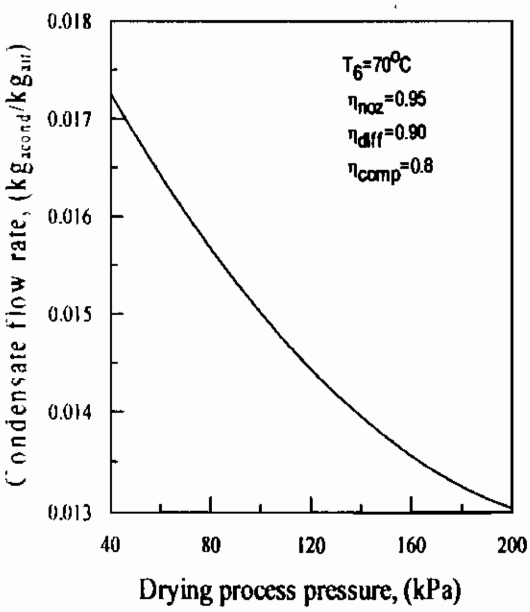

Fig. (9) Effect of drying process pressure on condensate flow rate.

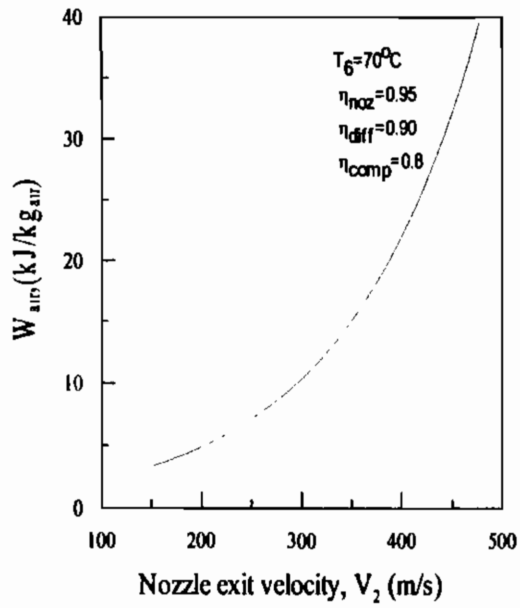

Fig. (11) Effect of nozzle exit velocity on specific work required per unit mass of dry air.

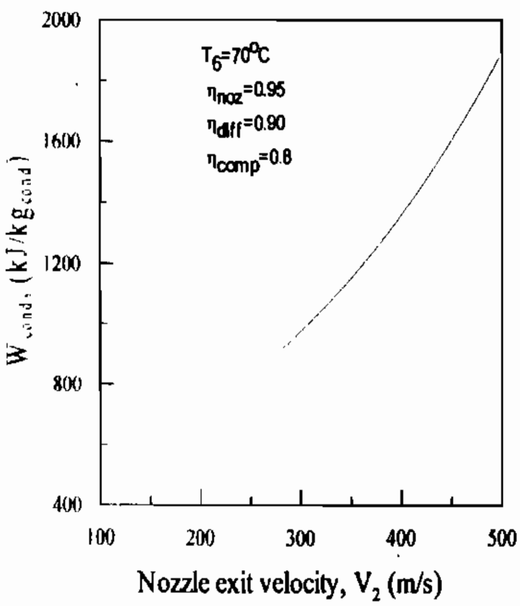

Fig. (10) Effect of nozzle exit velocity on specific work required per unit mass of condensate.

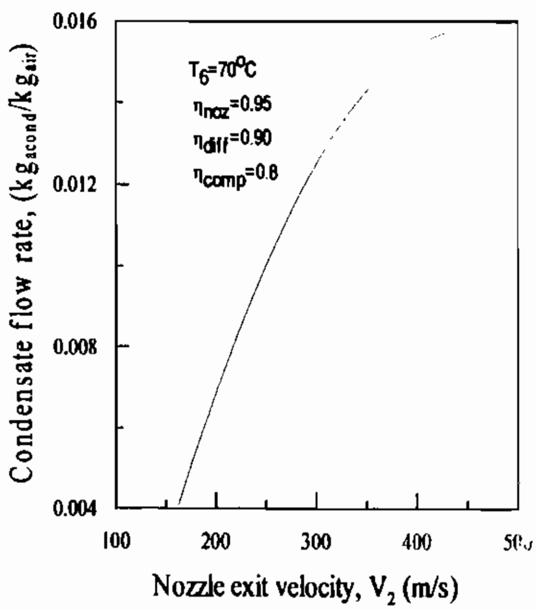

Fig. (12) Effect of nozzle exit velocity on condensate flow rate.

5. Conclusions

The following conclusions can be drawn:

1. The cycle can operate without any external heat addition. The energy used to provide the heat of vaporization to water for the purpose of removing the water from a wet product can be recovered by the use of a nozzle and diffuser.
2. The ideal cycle has much lower energy requirements and operating costs than other ideal cycles. Ideally, the cost of using the NCDC is approximately a third of the cost of the ideal open cycle heated dryer. Compared with a closed heat pump dehumidification cycle. In addition, the NCDC cycle can be operated at varying conditions, allowing for a single design to be utilized for various applications.
3. A final potential application for this technology is for processes that require the removal of hazardous liquids from a product. With an open cycle, the hazardous vapor would have to be removed from the air by the use of scrubbers, etc., using more energy. Since the NCDC is a closed cycle, the liquid can be removed from the product without releasing it to the environment. The liquid is evaporated in the dryer, condensed in the nozzle, and then removed, safely, in the separator.

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