THE THEORETICAL THERMAL PERFORMANCE OF AIR-CYCLE HEAT PUMPS WITH WATER INJECTION

FREDERICK A. CRESWICK, P.E.
Member ASHRAE

ABSTRACT

A thermodynamic analysis of the heating performance of air cycle heat pumps with and without liquid water injected at the compressor inlet was conducted. Two basic cycles were considered: a simple cycle and a regenerative cycle. Various levels of compressor and expander isentropic efficiency and heat-exchanger pressure drop were assumed. Calculations predict that water injection can substantially improve the efficiency of an ideal air-cycle heat pump, as much as doubling the coefficient of performance at low pressure ratios. However, with an assumed 5% pressure drop and component efficiencies as high as 0.95, no significant improvement in efficiency can be derived from water injection with either cycle. The injection of water drastically reduces the heating output per unit mass of air circulated.

INTRODUCTION

Air-cycle heat pumps have several potential advantages over vapor-compression systems which make them interesting subjects for investigation. A principal advantage of some air cycles is that their capacity and efficiency are theoretically less sensitive to outdoor air temperature; thus, they could be well suited to application in areas having extremely cold winter climates. Because they do not require a large refrigerant charge, their on-off cycling losses are potentially low. Also, there would be no susceptibility to loss of refrigerant. An open-cycle air-cycle system does not require an outdoor heat exchanger; therefore frosting and defrosting losses can be eliminated.

However, these potential advantages have not compensated in the past for the characteristically low efficiency of the air cycle, which has inhibited its use for most space-conditioning applications, both for cooling and heating. The use of water injection as a means of improving the efficiency of the cycle has been suggested by Edwards(1) and Hise and Wilson.(2) In both of these presentations, an analysis was described in which the ideal air-water cycle had shown a higher efficiency than the ideal R-22 vapor compression cycle.

Air-cycle cooling systems with water injection have been investigated analytically by Scott and Davis(3) and Cooper and Sumner.(4) Component efficiency losses were assumed in their analyses, but not pressure losses. Both concluded that there was no thermodynamic advantage from water injection in a high-pressure (compression first) cycle with realistic assumptions for component efficiencies. However, Cooper and Sumner found that water injection improved the efficiency of a practical low-pressure (expansion first) cycle.

The investigation described in this paper was part of a broad analytical reexamination(5) of various air cycles for possible energy-conservation opportunities. In the case of the water-injection cycles, the objective was to determine whether the potential efficiency advantage of the air-water heating cycle could be realized with practical component efficiencies and pressure drops, and if so, to define the most advantageous thermodynamic cycle.

DESCRIPTION OF CYCLES

Both a simple cycle and a regenerative cycle were examined; these are described in Fig 1. Both are high-pressure cycles; that is, compression precedes heat rejection and expansion. Low-pressure cycles (expansion first) were not examined with water injection because they are open to the indoor space, require an outdoor coil, and require a compressor and expander of larger volumetric capacity. Thus the low-pressure systems seemed less practical.

In the regenerative system, there is heat exchange between the air entering the system before compression and the air leaving the indoor coil before expansion. The principal effect of the regenerator is to make the system less sensitive to changes in outdoor air temperature.

In both cycles, it is assumed that water is injected at the compressor inlet only, and that the water is at the temperature of the air entering the compressor. Under some conditions, some of the water vapor will condense in the indoor coil. It was assumed that the condensate would be drained and returned to the compressor inlet and that the air entering the expander would, under those conditions, be saturated with water vapor. Thus, some of the injected water is recycled and some is exhausted from the expander. No specific mechanical configuration was envisioned, nor any particular size range. No consideration was given to means for injecting water or possible mechanical problems resulting from the presence of the water. It was assumed that consideration of those concerns would be worthwhile only if potential for high thermodynamic efficiency could be shown.

METHOD OF ANALYSIS

The principal challenge in the analysis of air cycle with water injection was to devise a simple approach to computing the work of isentropic compression and expansion of air-water mixtures. To accomplish this, an approximate expression was derived for the entropy change of the water; with this expression and another expression for the entropy change of the air, the temperature of the mixture at the end of the process was computed by an iterative trial-and-error process. The net entropy change was held to zero.

The entropy change of the air in the compression process was computed as

$$\Delta s_{air} = c_p \, \ell n(T_2/T_1) - R \, \ell n(p_2/p_1) \tag{1}$$

which is valid for an ideal gas. To approximate the entropy change of the water vapor, the following process was assumed: (1) heating the liquid from T_1 to T_2, (2) vaporization at T_2 and corresponding saturation vapor pressure and (3) expansion at T_2 from saturated vapor pressure to the final vapor pressure. The resulting expression is

$$\underset{\text{(compression)}}{\Delta s_{water}} = c_{p,\ell} \ell n(T_2/T_1) + h_{fg(T_2)}/T_2 - R \, \ell n(p_{2w}/p_{2w,sat}) \tag{2}$$

This expression is approximate because it is based on perfect-gas behavior in the vapor state; however, it proved sufficiently accurate for purposes of this investigation.

For the expansion process, Eq. 1 can again be used to compute the entropy change of the air. To compute the entropy change of the water, the following process was assumed: (1) compression of the water vapor at T_1 from the actual partial pressure to saturation pressure, (2) condensation at T_1, and cooling of the liquid from T_1 to T_2. The resulting expression is

$$\underset{\text{(expansion)}}{\Delta s_{water}} = c_{p,\ell} \ell n(T_2/T_1) - h_{fg(T_1)}/T_1 - R \, \ell n(p_{1w,sat}/p_{1w}) \; . \tag{3}$$

For the case of ice formation, the value 491.6°R (0°C, but computations were done in English units) was substituted for T_2 in Eq. 3 and the following two terms were added to the expression:

$$- h_{if}(491.6°R)/491.6 + c_{p,\ell} \ln(T_2/491.6) \quad (3a)$$

The additional terms account for the entropy change of freezing and subcooling.

To compute compressor work with losses, the isentropic work was divided by the assumed efficiency. For expansion, the isentropic temperature change was multiplied by the assumed efficiency, and the expansion work was computed therefrom. For all calculations, it was assumed that air was completely dry at the compressor inlet ahead of the water injection point. Also, it was assumed that air and water phases were in thermodynamic equilibrium at the end of both expansion and compression. In no case was a water injection rate assumed that would be large enough to exceed saturation at the compressor exit; that is, water at the compressor exit was entirely in vapor form.

Equations were needed for computing heat of vaporization, saturation vapor pressure, and saturation humidity ratio at atmospheric pressure. The following expressions were derived by assuming an appropriate functional form and computing the coefficients by hand computation methods:

$$h_{fg}(T) = 1092.04 - 0.49605\ T - 0.00036096\ T^2 \quad (4)$$

$$P_{w,sat}(T) = \log^{-1}[1.93899(\log T)^2 - 4.9281(\log T) + 1.89759] \quad (5)$$

$$W_{sat}(T) = \log^{-1}(-0.00005667\ T^2 + 0.022534\ T - 3.09797) \quad (6)$$

For most of the temperature range of interest, these expressions give results that are within 2% of standard table values.

Listings of the computational steps used to predict the thermal-performance of the cycles are presented in Ref 5.

To simplify the computational work, it was assumed that the compression and expansion efficiency were the same. For most calculations it was assumed that the internal working-fluid air left the indoor coil at 48.9°C (120 F). Pressure drop in the heat exchangers was expressed as $\Delta p/p$ and assumed to be 0.05 in most instances. (The cycle is not sensitive to where the pressure drop occurs.) With this parameter, the expansion ratio is computed as

$$p_{r,exp} = (1 - \Delta p/p)p_{r,comp} \quad (7)$$

Calculations were performed in English units using a programmable desk calculator.

PERFORMANCE OF DRY AIR CYCLES

Simple Cycle

Fig. 2 shows the theoretical coefficient of performance (COP) at 8.3°C (47 F) outdoor air temperature as a function of compressor pressure ratio and component efficiency for the simple heat-pump cycle shown in Fig. 1a. The top curve shows that the COP for the ideal cycle approaches 8.0 at a pressure ratio of about 1.6. This value is of little practical significance because the heating output approaches zero under this condition. The second curve gives the effect of 5% pressure loss in the system, still with perfect compression and expansion processes. This pressure loss drops the COP to about 3.2 at a pressure ratio of 2.5.

The remaining curves show the COP for component efficiencies of 0.95, 0.90, 0.85, 0.80, and 0.75 and a pressure drop of 5%. The peak COP for each of these efficiency levels and the pressure ratio at which each peak occurs is given by the following tabulation:

Theoretical Compressor-Only COP of the Simple Cycle as a Function of Component Efficiency

Component efficiency	Maximum compressor-only COP	Pressure ratio at maximum COP
0.95	2.28	3.0
0.90	1.87	3.5
0.85	1.62	4.0
0.80	1.46	4.5
0.75	1.35	5.0

The above COP values are for compressor power only, as no auxiliary fan power requirements are included in the calculations.

Since most commercially available air-to-air heat pumps have a rated COP in excess of 2.0 at 8.3°C, there is little incentive to consider any machine that will deliver less than this. [As a first approximation, it can be assumed that performance degradation of a conventional heat pump due to cycling, frosting, and defrosting losses will be of the same magnitude as the reduction in COP of the air-cycle machine when fan power requirements are considered. Cycling, frosting, and defrosting losses for the air-cycle system are negligible. Thus, the seasonal performance factor (SPF) of an air-cycle machine having a compressor-only COP of 2.0 might be roughly equal to the SPF of a conventional heat pump with an ARI rating of 2.0.] We can also make the judgment that there is little hope of achieving compressor and expander efficiencies greater than 0.90, which leads to the conclusion that there is no real possibility of developing a simple-cycle air-cycle heat pump having competitive thermal efficiency.

Regenerative Cycle

Fig. 3 gives the theoretical COP of the regenerative cycle for the same conditions assumed for the simple cycle, with the additional assumption that the regenerator effectiveness is 0.7. Two differences are apparent: (1) for a given component efficiency level, the regenerative system is slightly more efficient, and (2) peak efficiency is obtained at a lower pressure ratio. COP and pressure ratio for peak COP as a function of component efficiency are given in the following tabulation:

Theoretical Compressor-Only COP of the Regenerative Cycle as a Function of Component Efficiency

Component efficiency	Maximum compressor-only COP	Pressure ratio at maximum COP
0.95	2.40	1.9
0.90	2.03	2.1
0.85	1.78	2.3
0.80	1.61	2.4
0.75	1.48	2.5

The above results are for an outdoor temperature of 8.3°C. Ref 5 shows that a pressure ratio of 2.5 is best for minimum sensitivity to outdoor air temperature; and in fact, that this system loses less than 10% of its COP and capacity as outdoor air temperature is reduced from 8.3°C to −8.3°C (17 F).

The regenerative cycle provides an opportunity to achieve a compressor-only COP of 2.0 if a component efficiency of 0.90 can be achieved. Although this level of thermal performance is barely acceptable by present standards, in terms of steady-state performance at the 8.3°C rating point, the regenerative air-cycle looks more interesting if operation at very low temperatures is considered. For the regenerative cycle with 0.90 component efficiency, 5% pressure drop, and a pressure ratio of 2.5, the following variation in thermal performance with decreasing outdoor temperature is computed:

Outdoor air temperature, °C (F)	COP	Capacity, % of 8.3°C rating
13.9 (57)	2.07	102
2.8 (37)	1.95	98
−8.3 (17)	1.84	93
−19.4 (−3)	1.73	88

This computation illustrates the relative insensitivity of the cycle to outdoor air temperature and suggests that an air-cycle system could be developed having thermal performance superior to that of the best conventional units at extremely low ambients. However, no attempt was made in this study to predict a northern-climate SPF for such a machine.

PERFORMANCE WITH WATER INJECTION

Simple Cycle

The predicted effect of water injection on the simple-cycle air-cycle heat pump is shown in Figs. 4, 5, and 6 for pressure ratios of 3.5, 4.0 and 6.0, respectively.

With a pressure ratio of 3.5 (Fig. 4), water injection raises the COP of the ideal cycle from 3.32 with dry air to a peak of 6.35 with water injection at the rate of 0.022 kg/kg dry air. However, with a pressure drop of 5% and component efficiencies of 0.95 and 0.90, no significant increase in efficiency is predicted. The loss of effect of water injection on COP with nonideal component performance can be explained with the help of Table 1. For all cycles, the water injection tends to reduce the work of compression — a result of the cooling effect of the vaporization of the liquid water injected at the compressor inlet. The increased water vapor content at the expander inlet causes the expansion work to increase because latent heat is transferred to the air as the water vapor condenses and perhaps freezes. The combination of these effects causes the net work input to decrease.

Another effect of the water injection is to decrease the heat output. This effect is obvious because the work of compression is decreased; therefore less energy is available from the mixture leaving the compressor.

In the ideal cycle, the water injection reduces the net work input faster than it reduces the heat output. As shown in Table 1, the net work input decreases 86% while the heat delivered decreases 74%. This change almost doubles the COP. In the cycle with 0.90 component efficiency and 5% pressure drop (Table 1), the net work of the dry cycle is about twice that of the ideal dry cycle. The effect of water injection is to reduce net work input by about the same amount as in the ideal cycle; however, the percent reduction is less. Net work input is reduced only 38% while the heat output drops 65%, for a substantial reduction in COP.

At a pressure ratio of 4.0, (Fig. 5 and Table 2) the effect of water injection is less pronounced. Compression work is found to go through a minimum as water injection is increased; the subsequent increase is probably the result of the increased mass and volume to be compressed. Expansion work increases with water injection rate until saturation at the expander inlet is reached. (The abrupt change in slope of some of the curves in Figs. 5 and 7 is the result of the onset of condensation.) Assuming condensed water vapor at the indoor coil outlet is removed, further increases in water injection do not increase the

humidity ratio at the expander inlet. Heating effect is reduced sharply as water injection is increased, up to the point at which condensation in the indoor coil occurs; the reduction is minor beyond that point. These effects are documented in Table 2.

As shown in Fig. 6, water injection improves efficiency at all three component efficiency levels at a pressure ratio of 6.0, with maximum improvement at an injection rate of about 0.030 kg/kg dry air. At this pressure ratio, compression work is minimal at an injection rate of 0.035 kg/kg air and expansion work is not affected at injection rates above 0.013. Although water injection enhances the efficiency of the cycle at 6.0 pressure ratio for all cases investigated, the resulting efficiencies are all negligibly greater than those of the dry-air cycles at lower pressure ratios. Overall, there was no significant increase in efficiency of the simple-cycle air-cycle heat pump due to water injection for cycles with losses.

Fig. 7 summarizes the effect of water injection on heating capacity. Clearly, the effect of water injection is to reduce drastically the heating output per unit mass of dry air circulated.

Regenerative Cycle

It is unlikely that water injection could be used in the simple cycle when outdoor temperatures are below 0°C because of the complications due to freezing at the compressor inlet. The regenerative cycle avoids this limitation to a large extent; with the assumed regenerator effectiveness of 0.7, the mixture temperatures at both compressor inlet and regenerator outlet stay above the freezing point of water down to an ambient temperature of −20°C.

Fig. 8 shows the effect of water injection on the regenerative cycle for three pressure ratios at a component efficiency of 0.90, for both 8.3°C and −8.3°C outdoor air temperatures. The water injection clearly has an adverse effect on the efficiency of the cycle in most cases, and there is no significant improvement in any case.

CONCLUSIONS

The following conclusions were drawn from this investigation.

1. The regenerative cycle is superior to the simple cycle because it achieves a slightly higher COP for a given component efficiency level and its efficiency and capacity are less sensitive to changes in outdoor air temperature.

2. Compressor and expander isentropic efficiencies of 0.90 are needed if an air-cycle heat pump is to equal the SPF of the least efficient of current vapor-compression heat pumps (ARI rating of 2.0 at 8.3°C). This conclusion is based partly on the assumption that the air-cycle machine would have minimal losses due to cycling, frosting, and defrosting.

3. In general, the effects of water injection on the air heat-pump cycle energy balance are

- decrease work of compression (to some minimum value, after which a further increase in injection rate increases compression work),
- an increase in work of expansion (up to the point at which the expander inlet air is saturated),
- a reduction in heating capacity (in all cases).

4. In an ideal air-cycle heat pump, water injection reduces the net work input requirement faster than the heating output decreases. The net result is a substantial increase in theoretical COP, particularly at low pressure ratios. However, with real component efficiency (0.95 and lower) and with 5% pressure loss, no significant increase in COP can be obtained by water injection, while substantial reductions in heating capacity are incurred.

5. For the cycles and conditions investigated there is no practical incentive to develop an air-cycle heat pump using water injection.

NOMENCLATURE

c_p	Specific heat at constant pressure
h_{fg}	Latent heat of vaporization
h_{if}	Latent heat of fusion
p	Pressure
p_r	Pressure ratio
R	Gas constant
Δs	Entropy change
T	Temperature
W	Humidity ratio

Subscripts

comp	Compression
exp	Expansion
ℓ	Liquid
sat	at saturation
w	water vapor
1	initial or inlet
2	final or outlet

REFERENCES

1. T. Edwards, "New Air-Conditioning Refrigeration, and Heat Pump Cycle," abstract of ASHRAE Seminar Presentation, *ASHRAE Journal*, V. 19, no. 5, p. 45 (May 1977).

2. E. Hise and J. Wilson, "A Heat Pump Cycle with an Air-Water Working Fluid," Abstract of ASHRAE Seminar Presentation, *ASHRAE Journal*, V. 19, no. 5, p. 45 (May 1977).

3. Kenneth W. Cooper and Lee E. Simmer, Jr., "An Open Air Cycle Study Using Moist Air Thermodynamics," *ASHRAE Journal*, V. 20, pp. 68–71 (January 1978).

4. T. C. Scott and G. L. Davis, "Thermodynamics of Air/Water-Cycle Air Conditioning Systems," *Proceedings of the 1976 Purdue Compressor Technology Conference*, July 6-9, 1976, Purdue University, W. Lafayette, Indiana.

5. F. A. Creswick, "Analytical Evaluation of the Potential of Air-Cycle Machinery for Improving the Efficiency of Heat Pumps for Space Conditioning," Oak Ridge National Laboratory CON Report (to be published).

Table 1. Effect of Water Injection on the Energy Balance of a Simple-Cycle Air-Cycle Heat Pump

Energy balance parameter	Without water injection	With water injection, W=0.02
Ideal Cycle, No Losses		
Compressor work, kJ/kg	121.6	113.2
Expansion work, kJ/kg	97.3	109.8
Net work, kJ/kg	24.3	3.4
Heat delivered, kJ/kg	80.9	21.4
COP	3.32	6.35
Cycle With 0.90 Component Efficiency and $\Delta p/p = 0.05$		
Compressor work, kJ/kg	135.1	125.7
Expansion work, kJ/kg	84.6	94.3
Net work, kJ/kg	50.6	31.4
Heat delivered, kJ/kg	94.4	33.4
COP	1.87	1.06

Table 2. Effect of Water Injection on the Energy Balance of a Simple-Cycle Heat Pump at 4.0 Pressure Ratio[a]

Water injection rate, kg/kg air	Compression work, kJ/kg	Expansion work, kJ/kg	Net work, kJ/kg	Heat output, kJ/kg	COP
0	152.6	92.3	60.3	111.9	1.85
0.01	144.0	101.5	42.5	77.6	1.83
0.02	141.1	109.4	31.7	52.6	1.66
0.03	141.3	109.4	31.9	50.7	1.59

[a] With 0.90 component efficiency, 5% pressure drop.

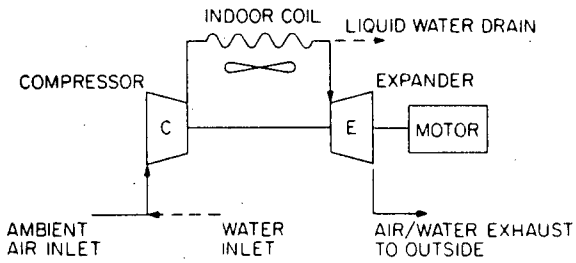

Fig. 1a Nonregenerative cycle

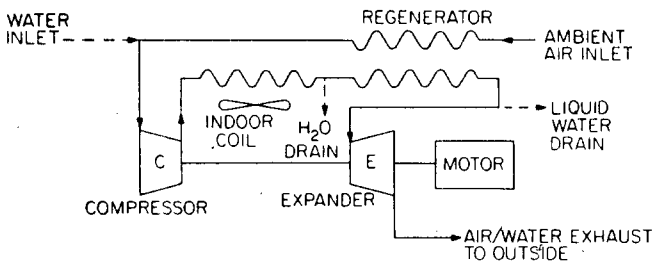

Fig. 1b Regenerative cycle

Fig. 1a,b Air cycles with water injection

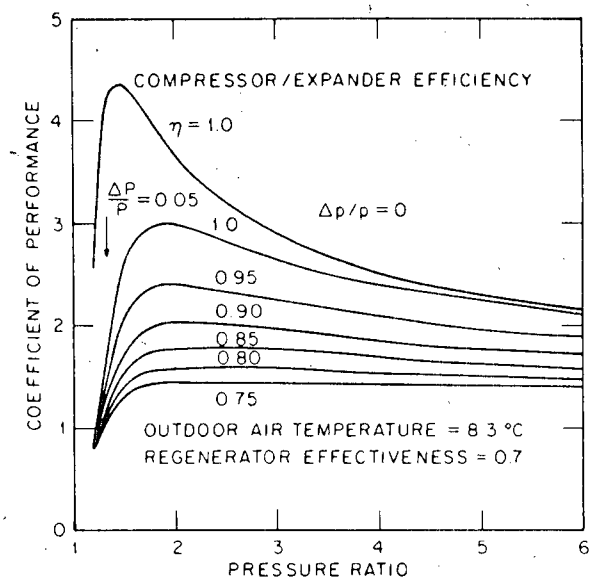

Fig. 3 COP of a high-pressure regenerative air-cycle heat pump as a function of component efficiency and pressure ratio

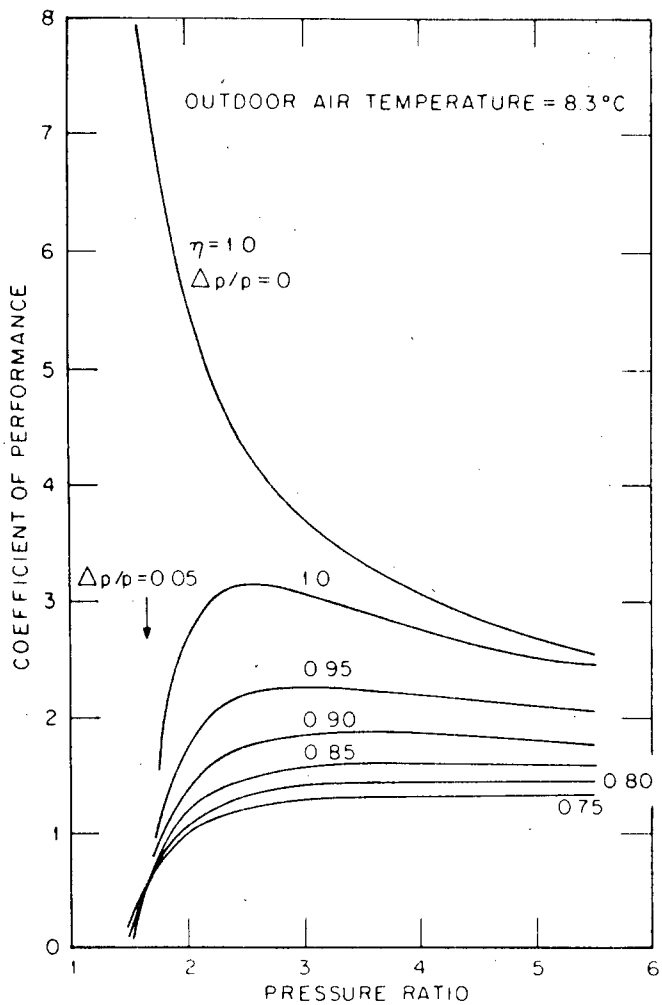

Fig. 2 COP of a simple air cycle heat pump as a function of efficiency and pressure ratio

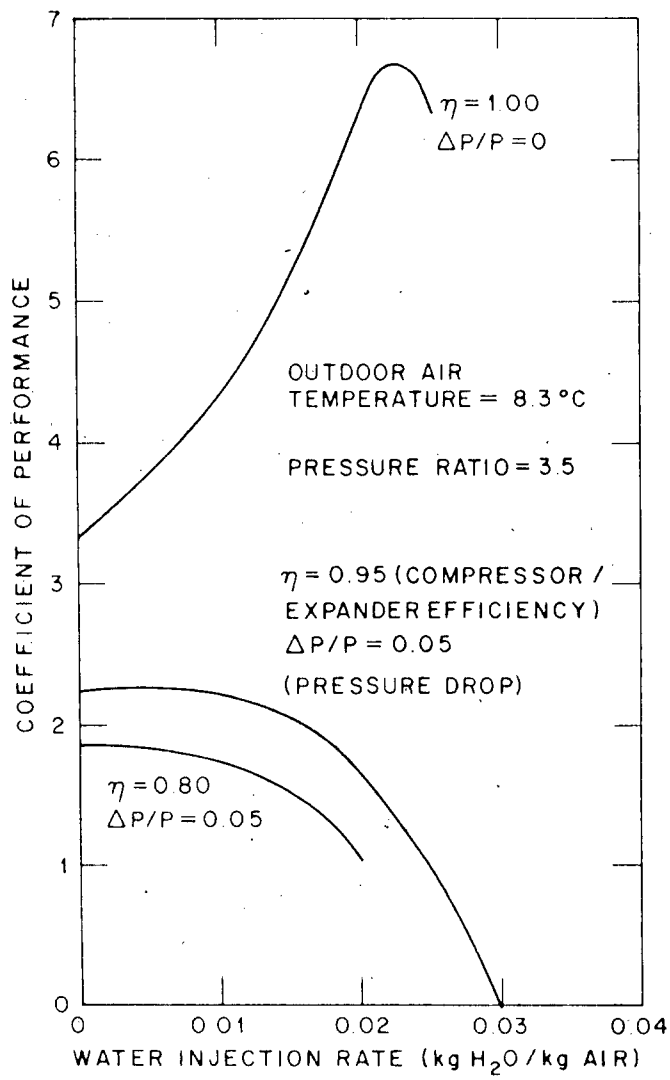

Fig. 4 Effect of water injection rate on the COP of a nonregenerative air-cycle heat pump

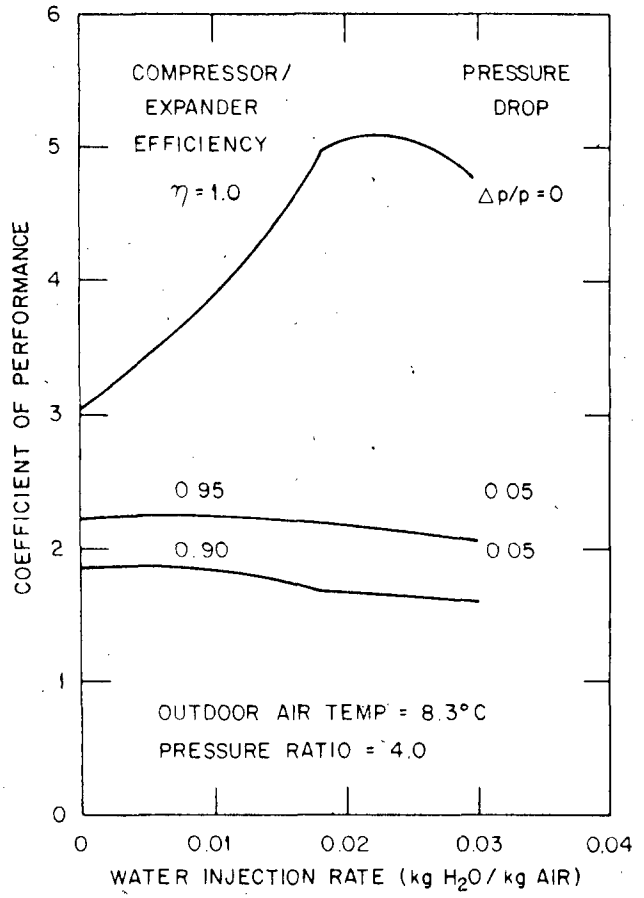

Fig. 5 Effect of water injection rate on the COP of a nonregenerative air-cycle heat pump

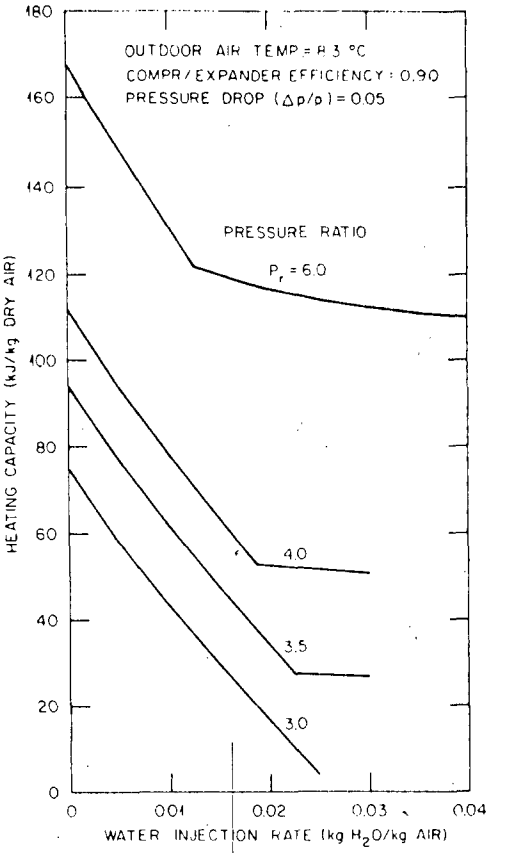

Fig. 7 Effect of water injection rate on the heating capacity of a nonregenerative air-cycle heat pump

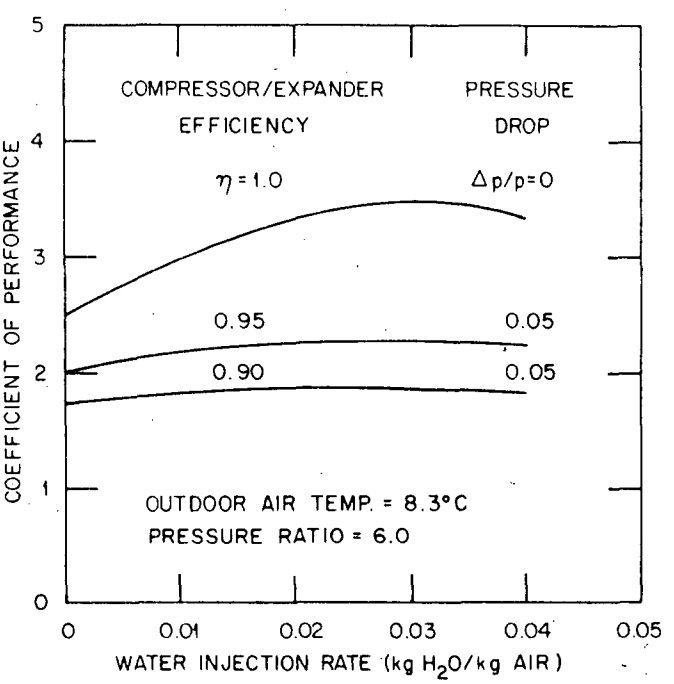

Fig. 6 Effect of water injection rate on the COP of a nonregenerative air-cycle heat pump, 6.0 pressure ratio

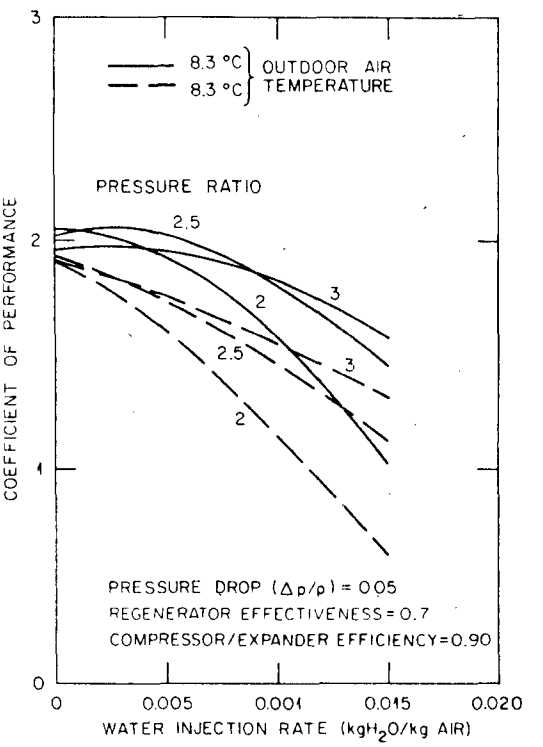

Fig. 8 Effect of water injection rate on the COP of a regenerative air-cycle heat pump, 0.90 component efficiency

DISCUSSION

DR. PRASANNA V. KADABA, Assoc. Prof. of Mech. Engg., Georgia Institute of Tech., Atlanta, GA: It was intriguing to hear the use of reversed Joule cycle for winter heating applications. Since the return air (generally at 80 F) needs to be heated to an acceptable level (e.g., 115 F), it is necessary to boost the temperature of the air flowing in the compressor stages from a design inlet temperature of about 0 F to an exit high value of about 140 F. If the water injection is used in the compressor while using the system for winter heating applications, the evaporation of droplets will quench the compression heat. Since the above temperature potentials cannot be sacrificed, the number of stages in the compressor will increase with water injection devices. Alternatively, if the design without water injection is fixed as a reference, the thermal capacity of the heat pump will drastically decrease. Recognizing this thermodynamic viewpoint, where was the incentive to consider water injection in this paper? Your results have proven this fact.

As pointed out at the meeting by M.M. Shah, water injection in the compressor stages of the reversed Joule cycle for summer cooling is advantageous since it reduces the burden on the heat exchanger situated before the turbine expansion stages, and the low temperature air leaving the turbine (expander) is utilized for cooling. However, (in the open-cycle configuration considered here) since the water is carried through the cycle, the leaving air will be saturated. Perhaps the water particles formed in the last stages of the turbine may be detrimental. One may need to consider mist eliminators at the exit from the turbine. It is useful to consider an additional psychrometric representation of moist air with water injection as it flows through the components of reversed Joule cycle which is being used for cooling applications.

Based on the above remarks, it is possible to equip the turbine with a water injection device. Here, as demonstrated in peaking power plants, the addition of water injection in the expander will produce more work. In the current application, the net work input will be smaller. Did the author investigate the advantage of injecting water in the expander (turbine) when the system is operating during heating cycle? Beware of icing problems. Energy economics of injecting water in the compressor during the cooling cycle and in the turbine during the heating cycle should be of interest to all concerned.

In a presentation of this nature, it is beneficial to participants at the meeting and to future readers if the author specifies significant base parameters on the diagram. Some of these are as follows:

- Number of stages in compressor and turbine;
- Their isentropic efficiencies;
- The conductance (UA product) and temperature effectiveness of the heat exchangers used;
- Temperature and their range of variations at selected points in the cycle;
- Mass flow rates or heat rate in certain components;
- Typical expected pressure drops in parts of the cycle.

F.A. CRESWICK: In reply to Dr. Kadaba's question regarding the incentives for this study, they are stated in the Introduction. The primary incentive was the large increase in COP of an ideal cycle reported by previous investigators.

We did not consider the possible advantage of injecting water into the expander because we judged that this would require the availability of an unlimited supply of hot water.

In regard to the final comment, the parameters Dr. Kadaba is looking for were not used because the analysis did not involve specific design parameters. All assumed performance parameters were stated in the paper.