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MARTIN MARIETTA

Investigation of an Ejector Heat Pump by Analytical Methods

OPERATED BY
MARTIN MARIETTA ENERGY SYSTEMS, INC.
FOR THE UNITED STATES
DEPARTMENT OF ENERGY

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
NTIS price codes—Printed Copy: A05 Microfiche A01

ORNL/CON-144
Dist. Category UC-95d

Energy Division

INVESTIGATION OF AN EJECTOR HEAT PUMP BY ANALYTICAL METHODS

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Date of Issue—July 1984

Prepared by the
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37831
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the
U.S. DEPARTMENT OF ENERGY
under Contract No. DE-AC05-84OR21400

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NOMENCLATURE

A	area
C_p	specific heat at constant pressure
COP	coefficient of performance
h	enthalpy
k	specific heat ratio
M	Mach number
$\dot{m}$	mass flow rate
P	pressure
Q	heat transfer
R	gas constant for refrigerant vapor
s	entropy
T	temperature
V	velocity
v	specific volume
W	work
η	efficiency
ρ	density

Subscripts

b	boiler
c	condenser, cooling
d	diffuser
E	ejector
e	evaporator
h	heating
n	nozzle
p	pump, pressure
s	isentropic

ACKNOWLEDGMENTS

The author would like to thank Dr. E. G. Keshock and Dr. R. Arimilli of the University of Tennessee for their initial effort in organizing the project activity and in overseeing the completion of the contractual objectives.

This research was a part of the buildings equipment research program performed at the Oak Ridge National Laboratory for the DOE Buildings Equipment Division. The author would like to thank F. C. Chen, J. W. Michel, and F. A. Creswick for their encouragement and assistance. Additional thanks go to K. S. Gaddis for typing the report, L. D. Gilliam for drawing the figures, and C. A. Srite for editing.

ABSTRACT

The objective of this study is to investigate the efficiency of an ejector heat pump by analytical methods. Using existing theories of ejector design, the optimum geometry of a high-efficiency ejector—including mixing section cross-sectional area, mass flow entrainment rate, ejector efficiency, and overall COP—for a heat pump cycle was determined.

A parametric study was performed to evaluate the COP values for different operating conditions. A sensitivity study determined the effects of nozzle efficiency and diffuser efficiency on the overall ejector heat pump COP. The off-design study estimated the COP for an ejector heat pump operating at off-design conditions.

Refrigerants 11, 113, and 114 are three of the halocarbons which best satisfy the criteria for an ejector heat pump system. The estimated COPs were 0.3 for the cooling mode and 1.3 for the heating mode at standard operating conditions: a boiler temperature of 93.3°C (200°F), a condenser temperature of 43.3°C (110°F), and an evaporator temperature of 10°C (50°F). Based on the same operating conditions, an optimum ejector geometry was estimated for each of the refrigerants R-11 and R-113. Since the COP values for heating obtained in this analysis are greater than unity, the performance of an ejector heat pump operating in the heating mode should be competitive with that of oil- or gas-fired furnaces or electrical resistance heaters.

1. INTRODUCTION

An ejector is a device which employs a high-velocity primary motive gas to entrain and accelerate a slower moving secondary suction gas, as shown in Fig. 1. The resulting kinetic energy of the mixture is subsequently used for self-compression to a higher pressure, thus fulfilling the function of a compressor.

A so-called "steam-jet" ejector was first used before 1901 by LeBlanc of France and Parsons of England. Steam is the motive fluid, and water is used as the refrigerant; the cooling effect is produced in the steam-jet refrigeration cycle by the continuous vaporization of a part of the water in the evaporator at a low absolute pressure level. This is of particular advantage in applications where direct vaporization is used for concentration or drying of foods and chemicals.[1]

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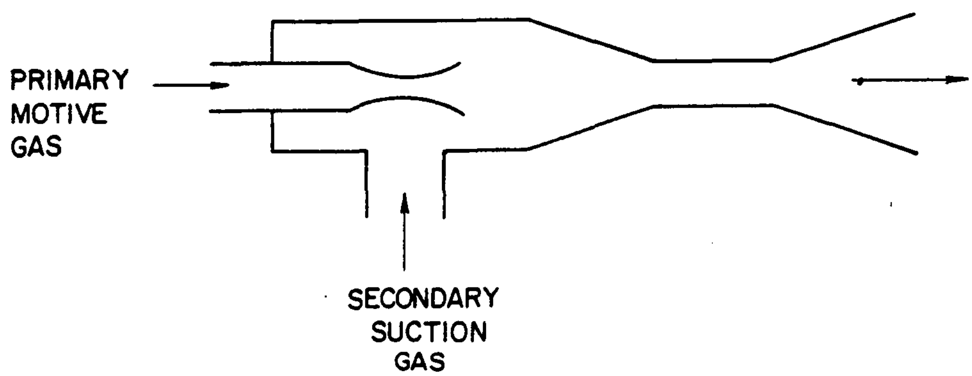

Fig. 1. Ejector.

In applications such as steam power plants, ejectors are used to evacuate noncondensable gases that enter the condenser because of the low vacuum. In such ejectors, steam is used as the motive gas, since it is available from the steam generator at a great range of pressures.[2] An ejector may also be used to produce a vacuum to refine edible oils. Volatiles are drawn off without requiring an elevation of their temperatures; rather, the pressure is reduced to bring the saturation temperature down to a lower value, which produces the same effect.[3] In aerodynamic applications, ejectors are used in "jet augmentor wings" (for aircraft that perform short takeoffs and landings), high-performance jet pumps, and certain types of combustors in which recirculation is stabilized by a jet mixing mechanism.[4] In the field of comfort air conditioning (cooling and heating), ejectors may be driven by automobile waste heat[5] or solar energy[6] to replace the mechanical compressor in an air conditioning system.

The ejector heat pump is similar to a conventional refrigeration cycle whose basic system components are a compression device, a condenser, an expansion device, and an evaporator. Instead of a mechanical compression device, however, an ejector is employed to compress the refrigerant vapor to the condenser pressure level as illustrated in Fig. 2. The ejector is driven by high-pressure refrigerant vapor supplied by a boiler. This is of particular advantage in using a low-grade thermal resource such as industrial waste heat. Alternately, the ejector can be driven by a gas-fired furnace.

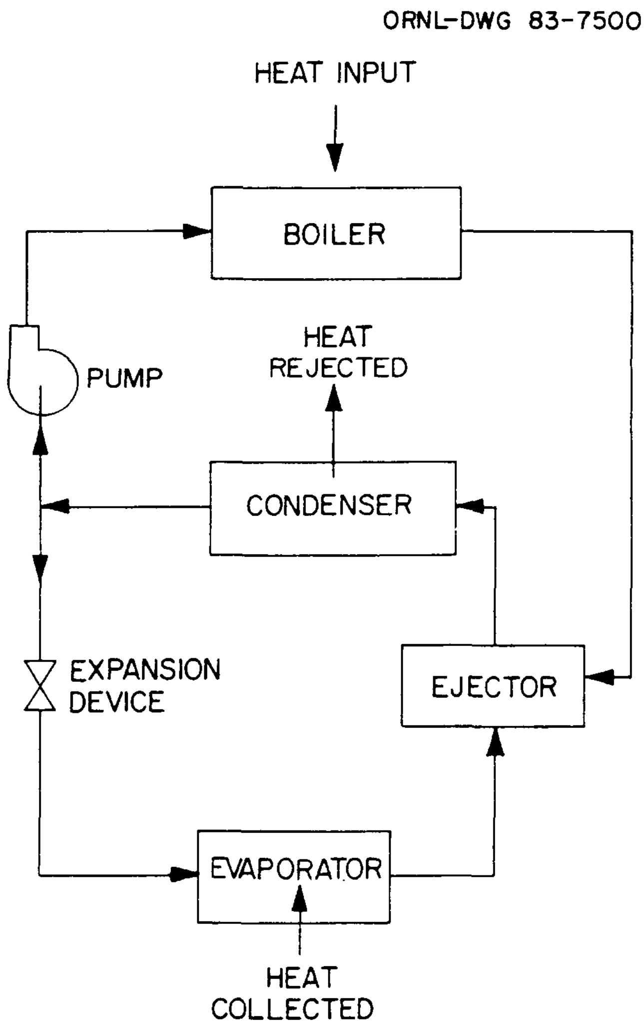

Fig. 2. Ejector refrigeration cycle.

The objective of this study is to analyze an ejector-compression cycle for heat pump application. It includes an investigation of the ejector heat pump from the analytical viewpoint. A thermodynamic analysis of the cycle and a gas dynamic analysis of the flow in the ejector itself will be carried out to obtain the optimum geometry of the ejector for the best heat pump performance.

A parametric study will calculate the optimum values of the coefficient of performance (COP), entrainment rate, and entrainment efficiency for different operating conditions, such as boiler temperatures, condenser temperatures, and evaporator temperatures. A sensitivity study will determine the effects of nozzle efficiency and diffuser efficiency on the overall ejector heat pump COP. An off-design study will estimate the COP for an ejector heat pump operating at off-design conditions.

A comparison will be made for different fluids which fall within the thermodynamic and physical criteria for use in actual ejector heat pump systems driven by low-grade thermal energy. Methods to improve the ejector heat pump efficiency will also be studied.

2. LITERATURE REVIEW

A good discussion of steam-jet ejector refrigeration systems may be found in the *ASHRAE Guide and Data Book.*[7] The steam-jet ejector has existed since the early 1900s. Its application in refrigeration is particularly useful for processing foods and chemical products that are sensitive to heat and require direct vaporization.

Articles have been written concerning analyses of the ejector from both the theoretical and experimental standpoints. Many of these articles were devoted to the applications of steam-jet or air-jet ejectors. The initial work of Keenan and Neumann[8] investigated a simple air ejector with a constant mixing area and without the diffuser section. Their work was to develop an analytical method for ejector performance and compare the results with experimental data. The comparison was made by presenting the entrainment rate as a function of pressure ratios for various area ratios. In a later work by Keenan, Neumann, and Lustwerk,[9] a diffuser section was added to the ejector, and the analysis was extended to include constant pressure mixing.

DeFrate and Hoerl[10] investigated the optimum design of ejectors by constant-area mixing analysis. The results showed the optimum entrainment rates and compression ratios for various ejector area ratios as a function of the pressure ratio of the secondary flow to the primary flow. Their study also included the effect on ejector performance if the molecular weights, temperatures, and specific heat ratios of the primary and secondary gases were dissimilar. Hamner's study[11] applied an approach similar to that of Keenan, Neumann, and Lustwerk[9] as far as the fundamental equations and assumptions were concerned. Hamner's work extended jet ejector theory to a heat pump system by producing both theoretical and experimental values of COP for a heat pump cycle using a single-component fluid, Trichloromonofluoromethane (R-11), operating in a closed loop.

The above literature may be summarized as (1) simplifying the mixing process by assuming it to occur at constant pressure or (2) simplifying the mixing process by assuming it to occur at constant cross-sectional area. The literature also formulates the conservation equations for mass, momentum, and energy. In these equations the optimum entrainment rate is obtained by iterating one of the conditions, such as an initially assumed entrainment rate or the pressure at the mixing point, and then the best result is selected.

Elrod[2] extended the ejector theory by introducing a new equation to form a closed set of equations. Therefore, if the inlet conditions and outlet pressure of the working fluids are prescribed, the optimum entrainment and the optimum area ratio may be determined. Based on the same theory, Chen[5] incorporated the ejector into a heat-driven mobile refrigeration cycle. Trichlorotrifluoroethane (R-113) was used as the working fluid. A similar approach is applied in the present study to determine the optimum mixing-section area of an ejector.

To further define the geometry of an optimum ejector (i.e., the cross-sectional areas of the convergent and divergent sections), a gas dynamic analysis of the refrigerant vapor through the ejector is performed. Gilbert and Hill[4] prepared a program based on the finite difference method to solve two-dimensional flow properties inside an ejector with a low pressure ratio. Thupvongsa[12] modified that program to make it suitable for using a refrigerant, R-113 or Dichlorotetrafluoroethane (R-114), as the working fluid instead of air. The latter program was tried in this study, but no satisfactory result was obtainable. In an ejector heat pump system, the ejector has to provide a pressure ratio of 3 to 4. But Gilbert and Hill's program was only able to simulate ejectors with pressure ratios around 1.1.

Coleman[13] of Conductron Corporation had a super ejector program to test a series of ejectors with different configurations. The results showed that the best performance for a mixing chamber is obtained with the parallel-convergent configuration, which will serve as the basic geometry for the present analysis.

The present study applies the ejector principle to a heat pump system which is powered by a heat source. The refrigerant is a single-component fluid operating in a closed loop. Both the primary and secondary fluids are in the vapor phase in the ejector. The work found in the literature is extended to calculate theoretically the optimum COP value for the ejector heat pump system. Types of studies performed include parametric study, off-design study, sensitivity study, analysis of possible methods to improve efficiency, and gas dynamic analysis of ejector geometry.

3. THEORETICAL ANALYSIS OF THE EJECTOR

The schematic diagram of an ejector with numerical symbols is shown in Fig. 3. The high-pressure motive gas enters the ejector at point 0, then expands through a nozzle to become supersonic flow. The secondary low-pressure gas is pumped in at point 4. The mixing takes place at station x, which is between 1 and 2 of the mixing section. From then on the mixture flows through the diffuser section. Kinetic energy contained in the flow is converted to pressure head; therefore, the mixture reaches a higher pressure and lower velocity at exit point 3. This process may also be described on a Mollier diagram (Fig. 4) as follows.

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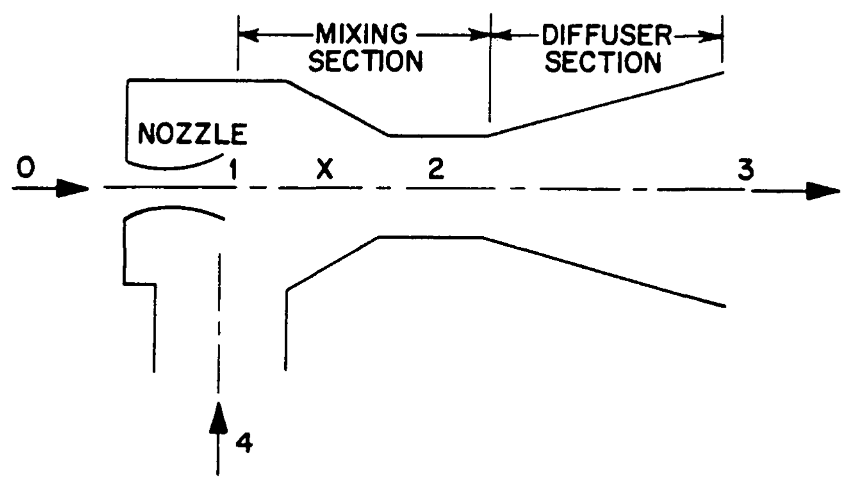

Fig. 3. Ejector model.

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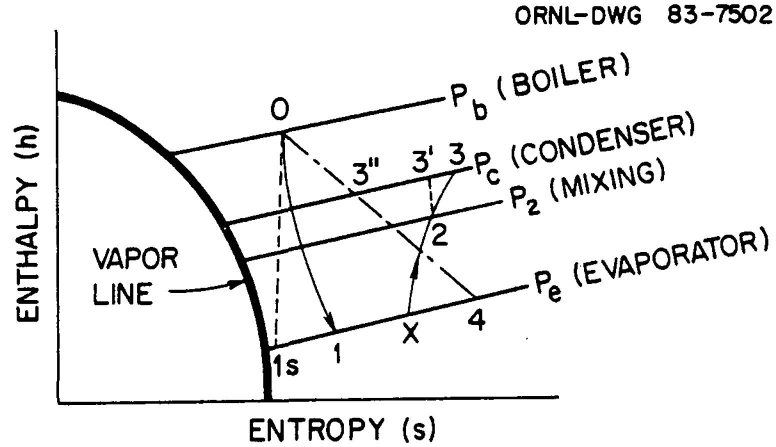

Fig. 4. Mollier diagram.

In an ejector heat pump system (described in Sect. 4) with refrigerant as the working fluid, the high-pressure motive gas is generally the refrigerant vapor supplied from the boiler. At state 0 the total pressure corresponds to the boiler pressure, P_b. From there, the primary refrigerant vapor expands through the nozzle to a static pressure, P_e (evaporator pressure), at state 1, which is located to the right of the isentropic expansion location 1_s. The low-pressure refrigerant vapor from the evaporator enters the ejector at a total pressure P_e corresponding to state 4. The two streams are merged at x and completely mixed at state 2. The combined refrigerant vapor is self-compressed through the diffuser section and then leaves the ejector at state 3, which is located to the right of state 3′ of the isentropic compression. The total pressure at state 3 corresponds to the condenser pressure, P_c.

3.1 DEFINITION OF EJECTOR EFFICIENCY

The ratio of the secondary mass flow rate to the primary mass flow rate may be defined as the entrainment rate. For given inlet conditions and outlet pressure of the working fluids, there is a maximum value of entrainment rate (ideal entrainment rate) attainable by an ejector. This ideal entrainment rate is a logical criterion of ejector performance. Therefore, the efficiency of an ejector can be defined by

$$\eta_E = \frac{\text{actual entrainment rate}}{\text{ideal entrainment rate}} . \tag{3.1}$$

The ideal entrainment rate may be derived by assuming the reversible processes of expansion, mixing, and compression of the refrigerant vapor throughout the ejector. If the inlet and outlet conditions of the refrigerant are prescribed and the overall changes of kinetic and potential energy are negligible, the conservation of energy requires that

$$\dot{m}_0 h_0 + \dot{m}_4 h_4 = (\dot{m}_0 + \dot{m}_4) h_3 . \tag{3.2}$$

The isentropic process requires that

$$\dot{m}_0 s_0 + \dot{m}_4 s_4 = (\dot{m}_0 + \dot{m}_4) s_3 . \tag{3.3}$$

If both primary and secondary fluids are the same substance, the ideal entrainment rate can be determined from the Mollier diagram, Fig. 4. According to Eqs. 3.2 and 3.3, state 3″ is the ideal exit condition, and the ideal entrainment rate can be expressed as

$$(\dot{m}_4/\dot{m}_0)_s = \frac{\overline{03''} \text{ length}}{\overline{3''4} \text{ length}} . \tag{3.4}$$

3.2 GOVERNING EQUATIONS

The following assumptions were applied for deriving the conservation of mass, momentum, and energy equations for each section in the ejector.

1. There is no external heat transfer.

2. The primary fluid expands through the nozzle from the boiler pressure, P_b, to the evaporator pressure, P_e.

3. The pressure drop and momentum of the secondary flow are negligible.
4. There is no wall friction.
5. All fluid properties are uniform over the cross section after complete mixing at section 2.
6. Potential energy is negligibly small in the energy equations.
7. The exit velocity at the ejector outlet may be ignored.

3.2.1 Flow Through the Nozzle

One-dimensional gas dynamic theory for perfect gas[14] can be applied to obtain the primary mass flow rate through the nozzle. If P_b/P_e is greater than the critical-flow pressure ratio across the nozzle, then the choking mass flow rate of unit nozzle-throat area is

$$\dot{m}_0 = P_b\left(\frac{k}{RT_b}\right)^{1/2}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)} , \tag{3.5}$$

where k is the specific heat ratio (assumed a constant), T is an absolute temperature, and the subscript b refers to the boiler.

The velocity of the primary fluid at the nozzle exit, assuming zero entrance velocity, is

$$V_1 = \left[2\eta_n\left(h_0 - h_{1s}\right)\right]^{0.5} , \tag{3.6}$$

where η_n is the nozzle efficiency.

3.2.2 Flow Through the Mixing Section

$$\text{Mass: } \dot{m}_0 + \dot{m}_4 = A_2V_2/v_2 , \tag{3.7}$$

$$\text{Momentum: } \dot{m}_0V_1 + P_4A_2 = (\dot{m}_0 + \dot{m}_4)V_2 + P_2A_2 , \tag{3.8}$$

$$\text{Energy: } \dot{m}_0h_0 + \dot{m}_4h_4 = (\dot{m}_0 + \dot{m}_4)(h_2 + V_2^2/2) , \tag{3.9}$$

where $P_4 = P_e = P_1$.

3.2.3 Flow Through the Diffuser

$$\text{Energy: } h_2 + V_2^2/2 = h_3 , \tag{3.10}$$

$$\text{Diffuser efficiency equation: } h_{3'} - h_2 = \eta_d(h_3 - h_2) , \tag{3.11}$$

where η_d is the diffuser efficiency and state 3′ has coordinates P_3 and s_2 (on Fig. 4, $P_3 = P_c$).

3.2.4 The Mixing Section Area

Let M_2 = Mach number at section 2, and for perfect gas

$$M_2 = V_2/(kP_2v_2)^{0.5} . \tag{3.12}$$

From Eqs. 3.7, 3.8, and 3.12, the cross-sectional area, A_2, may be solved by

$$A_2 = \frac{\dot{m}_0 V_1}{P_2(kM_2^2 + 1) - P_4} , \tag{3.13}$$

for the unit area of the nozzle throat.

3.3 THE OPTIMUM EQUATIONS

Actual entrainment rate depends both on operating conditions and the ejector geometry. For given conditions, the optimum mixing section area, A_2, can be found by maximizing $\dot{m}_4$ subject to the above governing Eqs. 3.5–3.12.

Thus, differentiating Eqs. 3.7–3.11 with $d\dot{m}_0 = d\dot{m}_4 = 0$, the following equations are obtained. From Eq. 3.7

$$dA_2/A_2 + dV_2/V_2 - dv_2/v_2 = 0 . \tag{3.14}$$

From Eq. 3.8

$$(\dot{m}_0 + \dot{m}_4)dV_2 + (P_2 - P_4)dA_2 + A_2 dP_2 = 0 . \tag{3.15}$$

From Eqs. 3.9 and 3.10

$$dh_2 + V_2 dV_2 = 0 . \tag{3.16}$$

From Eq. 3.11

$$dh_{3'} - dh_2 + \eta_d dh_2 = 0 . \tag{3.17}$$

With substitution of dh_2 from Eq. 3.16 and $dh_{3'} = T_{3'}ds_{3'} + v_{3'}dP_{3'} = T_{3'}ds_2$ (where $ds_{3'} = ds_2$ because of isentropic process and $dP_{3'} = 0$ because $P_{3'}$ is a known fixed value), Eq. 3.17 becomes

$$T_{3'}ds_2 + (1 - \eta_d)V_2 dV_2 = 0 . \tag{3.18}$$

There are two additional thermodynamic relations:

$$dh_2 = T_2 ds_2 + v_2 dP_2 , \tag{3.19}$$

$$dv_2 = -\left(\frac{\partial v_2}{\partial P_2}\right)_s dP_2 + \left(\frac{\partial v_2}{\partial s_2}\right)_P ds_2 . \tag{3.20}$$

With the realization that Pv^k is constant for perfect gas and isentropic processes, Eq. 3.20 becomes

$$dv_2 = -\left(\frac{v_2}{kP_2}\right)dP_2 + \left(\frac{dv_2}{ds_2}\right)_P ds_2 . \tag{3.21}$$

Equations 3.14, 3.15, 3.16, 3.18, 3.19, and 3.21 constitute six simultaneous equations containing six unknown differentials, ds_2, dV_2, dh_2, dv_2, dA_2, and dP_2. Solution of the simultaneous equations then yields[2,5]

$$M_2^2 \left[1 + (1 - \eta_d)\frac{T_2}{T_{3'}}\left(\left(\frac{d\left[\ln(T_2/v_2)\right]}{d(\ln v_2)}\right)_s + \frac{k}{1 - P_4/P_2}\right)\right] = 1 . \tag{3.22}$$

3.4 THE OPTIMUM MIXING SECTION AREA

For given conditions, the optimum criterion to operate an ejector is Eq. 3.22. Applying perfect gas and isentropic process relations

$$Pv^k = \text{constant} \ , \tag{3.23}$$

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{k-1}{k}} \ , \tag{3.24}$$

Eq. 3.22 may be simplified as

$$M_2^2 \left[1 + (1 - \eta_d)(P_2/P_3)^{\frac{k-1}{k}} \left(\frac{k}{1 - P_4/P_2} - k\right)\right] = 1 \ . \tag{3.25}$$

Combination of Eqs. 3.10 and 3.11 gives

$$h_{3'} - h_2 = \eta_d(V_2^2/2) \ . \tag{3.26}$$

With Eq. 3.24 and additional equations for perfect gas as follows:

$$h = C_p T \ ,$$

$$C_p = \frac{kR}{k-1} \ ,$$

$$M = \sqrt{kRT} \ ,$$

Eq. 3.26 becomes

$$(P_3/P_2)^{\frac{k-1}{k}} - \eta_d M_2^2(k-1)/2 = 1 \ . \tag{3.27}$$

Equations 3.25 and 3.27 will yield explicit solutions for M_2 and P_2 with values of η_d, k, P_3, and P_4 provided. Therefore, the optimum mixing section area, A_2, with respect to a unit nozzle-throat area may be obtained by substituting the M_2 and P_2 values into Eq. 3.13.

4. EJECTOR HEAT PUMP CYCLE ANALYSIS

A schematic diagram of the ejector heat pump is shown in Fig. 5. The high-pressure refrigerant vapor is supplied by the boiler, then enters the ejector to entrain the low-pressure vapor from the evaporator. The mixed-refrigerant vapor is self-compressed by converting the kinetic energy into static pressure, then discharges into the condenser, where the heat is taken away by the circulating air or water. The condensed liquid refrigerant from the condenser flows partly through an expansion valve to the evaporator, where the liquid refrigerant vaporizes by absorbing heat, and partly back to the boiler through a pump to complete a closed cycle.

4.1 IDEAL CYCLE ANALYSIS

From a thermodynamic point of view, the Carnot cycle gives the maximum efficiency for an engine and the maximum COP for a refrigerator. Using Bosnjakovic's method,[5] the maximum attainable COP for an ejector heat pump system may be established. Considering the ejector heat pump cycle as a whole system, the amount of heat added to the system at the boiler is Q_b, and the amount added at the evaporator is Q_e. The pump also adds work, W_p, to the system. The system rejects heat, Q_c, to the surroundings at the condenser. According to the first law of thermodynamics,

$$Q_c = Q_b + Q_e + W_p \; . \tag{4.1}$$

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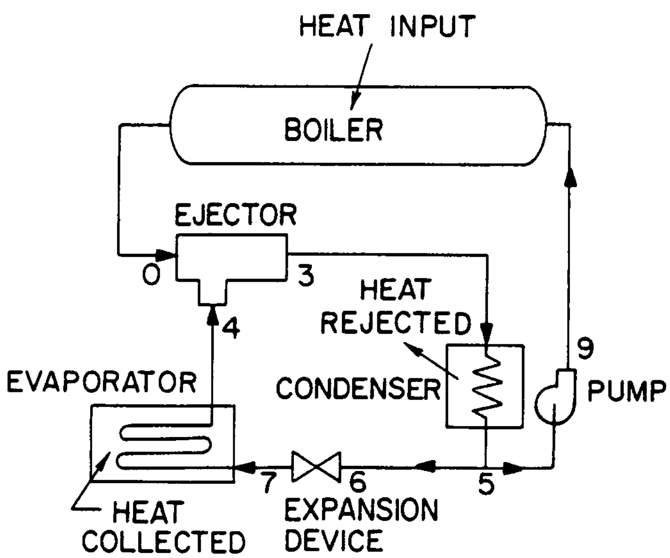

Fig. 5. Ejector heat pump.

If all three components (boiler, evaporator, and condenser) are operating at the constant temperature reservoirs T_b, T_e, and T_c, respectively, the maximum attainable COP for the system would be achieved in the isentropic process. Therefore, the entropy change for the boiler is $\Delta s_b = Q_b/T_b$, the change for the evaporator is $\Delta s_e = Q_e/T_e$, and the change for the condenser is $\Delta s_c = -Q_c/T_c$. Accordingly, the total entropy change of the system would be

$$\Delta s = \Delta s_b + \Delta s_e + \Delta s_c = \frac{Q_b}{T_b} + \frac{Q_e}{T_e} - \frac{Q_c}{T_c} = 0 . \tag{4.2}$$

If the work W_p is assumed negligible, substitution of Eq. 4.1 into Eq. 4.2 results in

$$Q_b\left(\frac{1}{T_b} - \frac{1}{T_c}\right) = Q_e\left(\frac{1}{T_c} - \frac{1}{T_e}\right) . \tag{4.3}$$

Therefore, the maximum COP for an ejector heat pump cooling cycle would be

$$(\mathrm{COP})_{c\text{-}ideal} = \frac{Q_e}{Q_b} = \frac{(T_b - T_c)}{T_b} \times \frac{T_e}{(T_c - T_e)} , \tag{4.4}$$

which is equal to the efficiency of a Carnot engine operating between the temperatures T_b and T_c multiplied by the COP for a Carnot refrigerator operating between the temperatures T_c and T_e as illustrated in Fig. 6.

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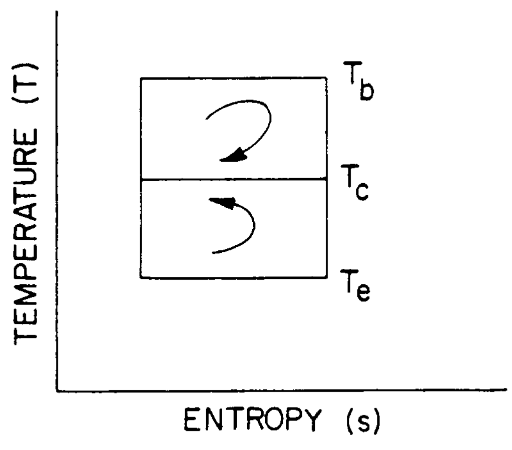

Fig. 6. T-s diagram of the ideal system for an ejector heat pump cooling cycle.

Similarly, the maximum COP for an ejector heat pump heating cycle could be derived as

$$(\mathrm{COP})_{h\text{-}ideal} = \frac{(T_b - T_e)}{T_b} \times \frac{T_c}{(T_c - T_e)} = 1 + (\mathrm{COP})_{c\text{-}ideal} , \tag{4.5}$$

which is equal to the efficiency of a Carnot engine operating between the temperatures T_b and T_e multiplied by the COP for a Carnot heat pump operating between the temperatures T_c and T_e.

4.2 CALCULATION SCHEME FOR ACTUAL CYCLE

The following specific numerical problem will demonstrate the calculation scheme for an ejector heat pump cooling cycle.

An ejector heat pump was designed for cooling to operate at a boiler temperature of 93.3°C (200°F), a condenser temperature of 43.3°C (110°F), and an evaporator temperature of 10°C (50°F). The refrigerant R-11 was selected as the working fluid. A T-s diagram of the refrigeration cycle is shown in Fig. 7. The numerical points correspond to those in Figs. 3 and 5.

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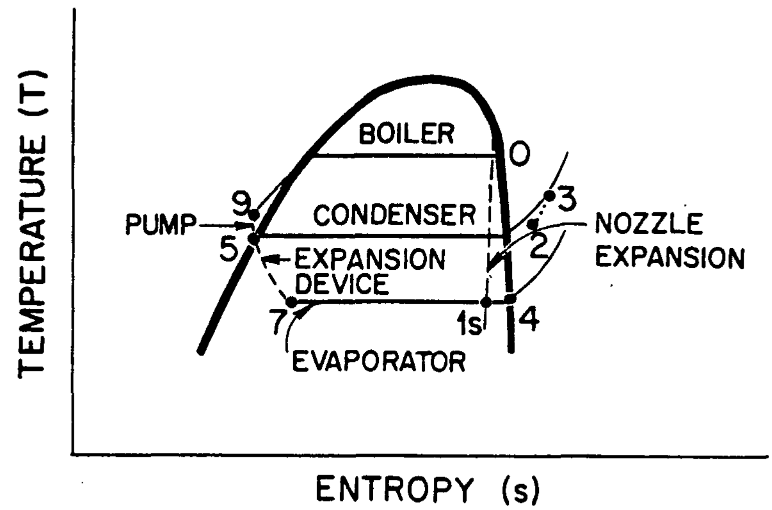

Fig. 7. T-s diagram of R-11 refrigeration cycle.

Boiler, condenser, and evaporator operated at the saturation pressures corresponding to the design temperatures. Table 1 shows thermodynamic properties at initial states for the refrigerant R-11. These properties were calculated from the saturation property subroutine[15] (see Appendix).

Table 1. The initial thermodynamic states of R-11

State	Temperature		Pressure		Enthalpy		Entropy	
	°C	°F	kPa	psia	J/kg	Btu/lb	J/(kg·K)	Btu/(lb·°R)
0	93.3	200	706.7	102.5	2.67×10^5	114.98	803.96	0.192
4	10	50	60.5	8.78	2.28×10^5	97.998	820.71	0.196
3			192.3	27.89				

It is assumed that the refrigerant vapor, R-11, will behave as a perfect gas with a constant specific heat ratio $k = 1.14$ (real k value does not significantly affect COP result). For a diffuser efficiency η_d of 0.75, solving Eqs. 3.25 and 3.27 simultaneously will yield

$$M_2 = 0.906 \ ,$$

$$P_2 = 136.4 \text{ kPa } (19.78 \text{ psia}) \ .$$

Assuming the nozzle efficiency $\eta_n = 0.97$, the primary flow velocity V_1 may be calculated by Eq. 3.6, where h_{1s} is determined from two known properties s_0 (isentropic expansion) and P_4 ($P_1 = P_4$) by the superheat vapor subroutines[15] (see Appendix):

$$h_{1s} = 2.236 \times 10^5 \text{ J/kg } (96.10 \text{ Btu/lb}) ,$$
$$V_1 = 291.9 \text{ m/s } (957.80 \text{ ft/s}) .$$

For a perfect gas, the critical pressure, denoted by P^* is

$$P^* = \left(\frac{2}{k+1}\right)^{k/(k-1)} P_0 = 407.4 \text{ kPa } (59.08 \text{ psia}) .$$

Since the back pressure [$P_1 = P_4 = 60.5$ kPa (8.78 psia)] is less than the critical pressure, which is 407.4 kPa (59.08 psia) in this case, the flow is choked at the nozzle. The mass flow rate of unit nozzle-throat area (1 ft^2) may be calculated by Eq. 3.5 with the gas constant R = 60.536 J/(kg·K) [11.248 ft-lbf/(lbm·°R)].

$$\dot{m}_0 = 327.49 \text{ kg/s } (721.98 \text{ lbm/s}) \tag{4.6}$$

The optimum cross-sectional area at the mixing section calculated by Eq. 3.13 is

$$A_2 = 0.469 \text{ m}^2 \ (5.048 \text{ ft}^2) \tag{4.7}$$

with nozzle-throat area equal to 0.093 m^2 (1 ft^2).

The calculation of entrainment rate requires a trial solution of Eqs. 3.2, 3.7, 3.10, and 3.11. First assuming $s_2 = 838.715$ J/(kg·K) [0.2003 Btu/(lb·°R)], with $P_2 = 136.4$ kPa (19.78 psia) known, the properties at state 2 may be determined from the superheat vapor subroutine[15] as follows:

$$h_2 = 2.4797 \times 10^5 \text{ J/kg } (106.597 \text{ Btu/lb}) ,$$
$$v_2 = 0.1364 \text{ m}^3/\text{kg } (2.185 \text{ ft}^3/\text{lb}) . \tag{4.8}$$

The enthalpy at state 3′, $h_{3'}$ is then determined from s_2 and P_3 [$P_3 = 192.4$ kPa (27.9 psia)] by the superheat vapor subroutines:

$$h_{3'} = 2.545 \times 10^5 \text{ J/kg } (109.379 \text{ Btu/lb}) .$$

Substitution of h_2 and $h_{3'}$ into Eq. 3.11 results in

$$h_3 = 2.5660 \times 10^5 \text{ J/kg } (110.306 \text{ Btu/lb}) .$$

Substitution of h_2 and h_3 into Eq. 3.10 results in

$$V_2 = 131.4 \text{ m/s } (431.09 \text{ ft/s}) . \tag{4.9}$$

From Eq. 3.2, the secondary mass flow rate $\dot{m}_4$ is determined:

$$\dot{m}_4 = 124.49 \text{ kg/s } (274.45 \text{ lb/s}) . \tag{4.10}$$

From Eqs. 4.6, 4.7, and 4.10

$$(\dot{m}_0 + \dot{m}_4)/A_2 = 963.7 \text{ kg/(s·m}^2) \ [197.31 \text{ lb/(s·ft}^2)] ,$$

while from Eqs. 4.8 and 4.9

$$V_2/v_2 = 963.3 \text{ kg/(s·m}^2) \ [197.30 \text{ lb/(s·ft}^2)] ,$$

which may be considered satisfying Eq. 3.7. If Eq. 3.7 is not satisfied, another s_2 must be assumed to carry out the above calculations again until Eq. 3.7 is satisfied. Of course the tolerance of convergence may be set appropriately for the computer calculation.

Therefore, the actual entrainment rate may be calculated from Eqs. 4.6 and 4.10 as

$$\dot{m}_4/\dot{m}_0 = 0.38 \ . \tag{4.11}$$

The compression ratio is calculated by

$$P_3/P_4 = 27.89/8.78 = 3.18 \ .$$

The ideal entrainment rate $(\dot{m}_4/\dot{m}_0)_s$ may be calculated from Eq. 3.4 by using Fig. 4. However, it is cumbersome to construct a Mollier diagram and measure the relevant lengths. Instead it is much simpler using a trial solution with the existing superheat vapor subroutines. Since the vapor properties of states 0 and 4 were listed in Table 1, the corresponding points (h_0,s_0) and (h_4,s_4) on Fig. 4 are determined. Therefore, the slope of the line $\overline{04}$ may be calculated as follows:

$$\text{slope} = (h_0 - h_4)/(s_0 - s_4) \ . \tag{4.12}$$

With an assumed $s_{3''}$ value and the pressure P_3 known, $h_{3''}$ may be determined from the superheat vapor subroutines. The slope of the line $\overline{3''4}$ is calculated accordingly:

$$\text{slope} = (h_{3''} - h_4)/(s_{3''} - s_4) \ . \tag{4.13}$$

If Eqs. 4.12 and 4.13 are not equal, a new $s_{3''}$ value must be assumed to repeat the calculations until the tolerance of convergency is met. For this particular case, the ideal entrainment rate is calculated as

$$(\dot{m}_4/\dot{m}_0)_s = 1.193 \ . \tag{4.14}$$

The ejector efficiency is calculated from Eqs. 4.11 and 4.14 by the definition of Eq. 3.1:

$$\eta_E = 0.319 \ .$$

The thermodynamic properties of different states of an ejector heat pump cycle are shown in Table 2. The numerical states correspond to Figs. 5 and 7. The ideal COP may be calculated from Eq. 4.4 for cooling and Eq. 4.5 for heating as follows:

$$(\text{COP})_{c\text{-}ideal} = 1.16 \ , \tag{4.15}$$

$$(\text{COP})_{h\text{-}ideal} = 2.16 \ . \tag{4.16}$$

Table 2. Thermodynamic properties of an ejector heat pump cycle

State	Temperature		Pressure		Enthalpy		Entropy	
	°C	°F	kPa	psia	J/kg	Btu/lb	J/(kg·K)	Btu/(lb·°R)
0	93.3	200	706.7	102.5	2.67×10^5	114.98	803.96	0.192
3	62.8	145.12	192.3	27.89	2.57×10^5	110.31	845.83	0.202
4	10	50	60.5	8.78	2.28×10^5	97.998	820.71	0.196
5	43.3	110	192.3	27.89	7.16×10^4	30.798	263.8	0.063
7	10	50	60.5	8.78	7.16×10^4	30.798	267.99	0.064
9	43.7	110.72	706.7	102.5	7.20×10^4	30.953	263.8	0.063

The actual COPs are calculated according to the definition of coefficient of performance:

$$(\mathrm{COP})_c = \frac{\text{energy sought}}{\text{energy that costs}} = \frac{\dot{m}_4}{\dot{m}_0} \times \frac{(h_4 - h_7)}{(h_0 - h_5)} = 0.307 \ , \tag{4.17}$$

$$(\mathrm{COP})_h = \frac{\text{energy sought}}{\text{energy that costs}} = \frac{(\dot{m}_4 + \dot{m}_0)}{\dot{m}_0} \times \frac{(h_3 - h_5)}{(h_0 - h_5)} = 1.307 \ . \tag{4.18}$$

From Eqs. 4.15 and 4.17, the actual COP is only 26% of ideal COP for the cooling cycle, partially because of the low entrainment rate ($\dot{m}_4/\dot{m}_0 = 0.38$). From Eqs. 4.16 and 4.18, the actual COP is 60% of the ideal COP for the heating cycle; therefore, the ejector heat pump appears to be more attractive for the heating mode than the cooling mode.

4.3 PARAMETRIC STUDY

The calculation scheme of an actual ejector heat pump cycle using R-11 as the working fluid has been demonstrated in the previous section. A computer program which is listed in the Appendix was written to perform all the calculations. The available refrigerant property subroutines (see Appendix) for several refrigerants including R-11, R-12, R-22, R-113, R-114, and R-502 are incorporated in the program. For a selected refrigerant and a given set of the input values including diffuser efficiency, nozzle efficiency, specific heat ratio, gas constant, and operating conditions for the ejector heat pump (i.e., boiler temperature, evaporator temperature, and condenser temperature), the program will calculate all the results presented in the previous section. Care has been taken in the program to handle those refrigerants such as R-11, R-12, R-22, and R-502, for which saturated vapor entropy increases as the pressure decreases. With these kinds of refrigerants as the working fluid, the refrigerant vapor will become a two-phase (liquid and vapor) mixture after the nozzle expansion. Therefore, the vapor quality must be calculated first in order to determine the enthalpy and the specific volume at state point 1s (refer to Fig. 7).

The boiler, evaporator, and condenser of a heat pump system usually operate at a range of temperature and pressure conditions instead of operating at a very specific condition. The temperature change of any of these three components will affect the performance (COP) of an ejector heat pump. It is of interest to evaluate the COP for an ejector heat pump operating at different conditions. Table 3 shows the T_b, T_c, and T_e parametric study for R-11 as the working fluid. In the first group of results, boiler temperatures are increased from 82.2°C (180°F) to 148.9°C (300°F) with constant temperatures for the condenser [T_c = 43.3°C (110°F)] and evaporator [T_e = 10°C (50°F)]. The entrainment rates, $\dot{m}_4/\dot{m}_0$, are listed in column four of Table 3. The ideal entrainment rates, $(\dot{m}_4/\dot{m}_0)_s$, are listed in column five. The ejector efficiencies, η_E, are listed in column six. The optimum mixing section cross-sectional areas on the basis of unit nozzle-throat area, A_2, are listed in column seven. The COPs for cooling, $(\mathrm{COP})_c$, are listed in column eight. The COPs for ideal cycle, $(\mathrm{COP})_{c\text{-}ideal}$, are listed in column nine. The results indicate that the entrainment rate increases as the boiler temperature increases since there is more motive energy. The ideal entrainment rate also increases, resulting in a decrease in ejector efficiency. From Eq. 4.17 the $(\mathrm{COP})_c$ is directly proportional to the entrainment rate.

In the second group of results, condenser temperatures are varied from 32.2°C (90°F) to 54.4°C (130°F) with constant temperatures for the boiler [T_b = 93.3°C (200°F)] and the

Table 3. T_b, T_c, T_e parametric study for R-11[a]

T_b (°F)	T_c (°F)	T_e (°F)	$\dot{m}_4/\dot{m}_0$	$(\dot{m}_4/\dot{m}_0)_s$	η_E	A_2 m^2	A_2 ft^2	$(COP)_c$	$(COP)_{c\text{-}ideal}$
					Group 1				
180	110	50	0.314	0.953	0.33	0.34	3.651	0.257	0.93
190	110	50	0.350	1.074	0.326	0.4	4.305	0.283	1.046
200	110	50	0.385	1.193	0.322	0.47	5.048	0.307	1.159
205	110	50	0.399	1.248	0.3195	0.51	5.455	0.316	1.214
210	110	50	0.416	1.307	0.319	0.55	5.889	0.328	1.269
220	110	50	0.447	1.416	0.316	0.64	6.839	0.349	1.375
240	110	50	0.501	1.628	0.308	0.85	9.117	0.382	1.579
260	110	50	0.551	1.827	0.302	1.11	12.00	0.413	1.771
280	110	50	0.595	2.01	0.296	1.45	15.66	0.438	1.953
300	110	50	0.634	2.18	0.29	1.89	20.3	0.46	2.13
					Group 2				
200	90	50	0.604	2.192	0.276	0.79	8.512	0.488	2.125
200	100	50	0.475	1.591	0.298	0.6	6.44	0.381	1.545
200	105	50	0.427	1.372	0.311	0.53	5.681	0.342	1.335
200	110	50	0.385	1.193	0.322	0.47	5.048	0.307	1.159
200	115	50	0.35	1.037	0.337	0.42	4.513	0.278	1.01
200	120	50	0.323	0.906	0.356	0.38	4.06	0.256	0.883
200	130	50	0.275	0.692	0.397	0.31	3.32	0.216	0.676
					Group 3				
200	110	30	0.352	0.861	0.408	0.45	4.839	0.271	0.835
200	110	40	0.359	1.002	0.358	0.46	4.908	0.281	0.974
200	110	50	0.385	1.193	0.322	0.47	5.048	0.307	1.159
200	110	55	0.405	1.31	0.31	0.48	5.155	0.326	1.277
200	110	60	0.433	1.457	0.297	0.49	5.297	0.352	1.418
200	.110	70	0.511	1.851	0.276	0.53	5.736	0.423	1.807

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9.

evaporator [T_e = 10°C (50°F)]. As expected, the $(COP)_c$ decreases as the condenser temperature increases. In the third group of results, evaporator temperatures are increased from −1.1°C (30°F) to 21.1°C (70°F) with constant temperatures for the boiler [T_b = 93.3°C (200°F)] and the condenser [T_c = 43.3°C (110°F)]. The $(COP)_c$ increases accordingly.

Figure 8 shows the results of parametric study for R-11 as the working fluid. The coordinates are $(COP)_c$ vs temperatures. The upper set of curves is for the ideal cycle. Three curves intersect at the standard operating conditions; i.e., T_b = 93.3°C (200°F), T_c = 43.3°C (110°F), and T_e = 10°C (50°F). The lower set of curves is for the optimum-designed ejector heat pump cycle. There is a set of optimum ejector geometrical parameters for each operating condition. Obviously, the optimum $(COP)_c$ is less than the ideal $(COP)_{c\text{-}ideal}$, although there is the same trend between these two sets of curves. It should be reemphasized that the low entrainment rate affixed to the ejector affects the overall performance of the ejector heat pump. The large discrepancies between ideal and actual entrainments and COPs are due to the low ejector efficiency. When the high-velocity primary flow entrained the secondary low-velocity gas, a great deal of kinetic energy was lost in the mixing process. Besides, in flowing through the shock wave, the gas experienced a

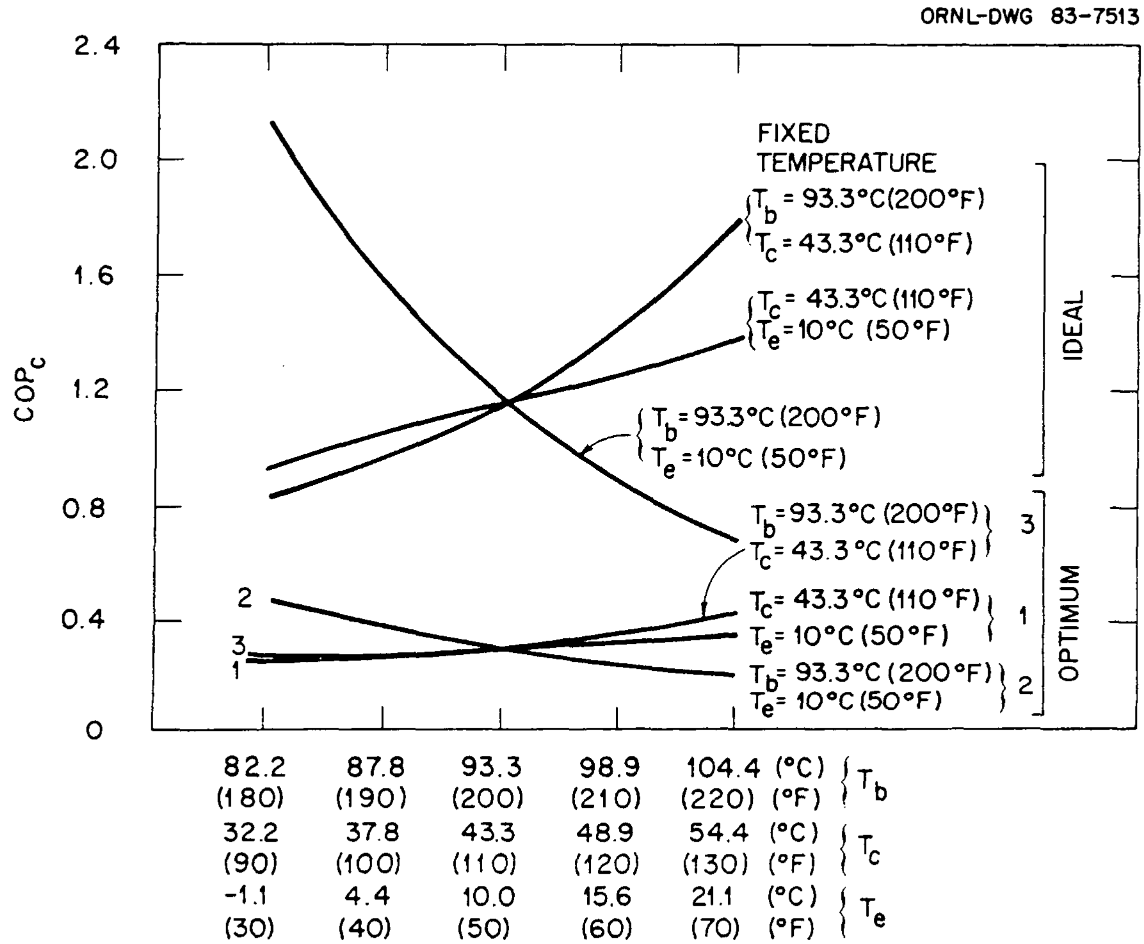

Fig. 8. Parametric study for R-11.

decrease in its available energy and, accordingly, an increase in its entropy. Both mixing and shock processes are of high irreversibility.

4.4 SENSITIVITY STUDY

In the previous calculations, nozzle efficiency and diffuser efficiency were assumed constant values (η_n = 0.97 and η_d = 0.75). According to Eq. 3.6, the kinetic energy (velocity) of the motive gas is affected by the nozzle efficiency. According to Eq. 3.11 and Fig. 4, more kinetic energy is required for compression in the diffuser with low efficiency.

Table 4 shows the sensitivity study of R-11 based on the following standard operating conditions: T_b = 93.3°C (200°F), T_c = 43.3°C (110°F), and T_e = 10°C (50°F). The values of A_2, $\dot{m}_4/\dot{m}_0$, and $(COP)_c$ are listed according to the variations of η_n and η_d independently. The results indicate that $(COP)_c$ increases as η_n and η_d increase.

4.5 OFF-DESIGN STUDY

Table 3 shows that there is an optimum mixing section area A_2 for each operating condition. However, an ejector heat pump is not always operating at the design conditions.

Table 4. Sensitivity study of R-11 at fixed temperatures T_b = 93.3°C (200°F), T_c = 43.3°C (110°F), and T_e = 10°C (50°F) and at varying efficiencies of nozzle and diffuser

η_n	η_d	A_2 m^2	A_2 ft^2	$\dot{m}_4/\dot{m}_0$	$(COP)_c$
0.97	0.65	0.45	4.820	0.366	0.292
0.97	0.70	0.46	4.931	0.374	0.298
0.97	0.75	0.47	5.048	0.385	0.307
0.97	0.80	0.48	5.173	0.392	0.313
0.97	0.85	0.49	5.309	0.404	0.323
0.85	0.75	0.44	4.725	0.286	0.228
0.90	0.75	0.45	4.862	0.329	0.263
0.95	0.75	0.46	4.996	0.371	0.297
0.97	0.75	0.47	5.048	0.385	0.307
0.98	0.75	0.47	5.074	0.39	0.311

Therefore, the off-design study should be performed to determine the effect on COP for the ejector heat pump.

For example, in an ejector heat pump operating at the temperature conditions T_b = 93.3°C (200°F), T_c = 43.3°C (110°F), and T_e = 10°C (50°F), using R-11 as the working fluid, the optimum mixing section cross-sectional area was determined to be 0.47 m^2 (5.048 ft^2) per unit nozzle-throat area (see Table 3). With the designed cross-sectional area A_2 = 0.47 m^2 (5.048 ft^2), the $(COP)_c$ may be calculated for different operating conditions as follows:

First, $\dot{m}_0$ and V_1 may be calculated from Eqs. 3.5 and 3.6 respectively. Solving Eqs. 3.13 and 3.27 simultaneously with known values of A_2, k, P_4, P_3, and η_d results in values for P_2 and M_2. When state 2 conditions (P_2 and M_2) are determined, the rest of the calculations may be carried out in a manner similar to the method illustrated in Sect. 4.2 to obtain the entrainment rate $\dot{m}_4/\dot{m}_0$ and the COP. When the boiler temperature is higher than 93.3°C (200°F) or the condenser temperature is less than 43.3°C (110°F) for the present case, the flow will be choked at the mixing section. There are no solutions to Eqs. 3.13 and 3.27 simultaneously. Thus, M_2 equals unity and P_2 may be solved from Eq. 3.13.

The results of off-design study for R-11 when A_2 equals 0.47 m^2 (5.048 ft^2) are listed in Table 5. For the first group T_b is varied from 82.2°C (180°F) to 104.4°C (220°F) while T_c and T_e are maintained constant. For the second group T_c is varied from 32.2°C (90°F) to 48.9°C (120°F) while T_b and T_e are maintained constant. For the third group T_e is varied from −17.8°C (0°F) to 21.1°C (70°F) while T_b and T_c are maintained constant. The resulting $(COP)_c$ values are also plotted in Fig. 9 in addition to the parametric study results. When T_b varies, the $(COP)_c$ decreases as T_b deviates from the standard temperature of 93.3°C (200°F). The $(COP)_c$ decreases to 0 when T_b reaches below 82.2°C (180°F) or beyond 104.4°C (220°F). When T_c varies, the $(COP)_c$ also decreases from that of the standard temperature 43.3°C (110°F). The $(COP)_c$ decreases to 0 when T_c reaches 32.2°C (90°F) or beyond 48.9°C (120°F). The $(COP)_c$ decreases most for these three groups of temperature

Table 5. Off-design study for R-11 with A_2 = 0.47 m^2 (5.048 ft^2)[a]

T_b (°F)	T_c (°F)	T_e (°F)	M_2	P_2		$\dot{m}_4$		$\dot{m}_0$		$\dot{m}_4/\dot{m}_0$	$(COP)_c$
				kPa	psia	kg/s	lb/s	kg/s	lb/s		
						Group 1					
180	110	50	0.345	182.7	26.5	0	0	250.3	551.9	0	0
190	110	50	0.611	164.4	23.84	75.7	167.0	286.4	631.3	0.265	0.214
195	110	50	0.746	152.4	22.1	106.4	234.6	305.9	674.3	0.348	0.279
200	110	50	0.91	136.7	19.82	125.6	276.8	326.5	719.7	0.385	0.307
205	110	50	1	131.5	19.07	108.5	239.3	348.2	767.7	0.312	0.247
210	110	50	1	139.6	20.25	66.5	146.5	371.2	818.3	0.179	0.141
220	110	50	1	157.6	22.86	0	0	420.9	927.9	0	0
						Group 2					
200	90	50	1	123.8	17.95	0	0	326.5	719.7	0	0
200	100	50	1	123.8	17.95	24.4	53.8	326.5	719.7	0.075	0.06
200	105	50	1	123.8	17.95	87.6	193.1	326.5	719.7	0.268	0.215
200	110	50	0.91	136.7	19.82	125.6	276.8	326.5	719.7	0.385	0.307
200	115	50	0.69	171.3	24.84	100.0	220.5	326.5	719.7	0.306	0.244
200	120	50	0.512	203.4	29.5	44.1	97.2	326.5	719.7	0.135	0.107
						Group 3					
200	110	0	0.873	140.0	20.31	123.6	272.6	326.5	719.7	0.379	0.275
200	110	10	0.848	142.5	20.67	116.1	255.9	326.5	719.7	0.356	0.263
200	110	20	0.836	143.7	20.84	111.8	246.5	326.5	719.7	0.343	0.259
200	110	30	0.84	143.3	20.78	112.7	248.4	326.5	719.7	0.35	0.266
200	110	40	0.86	141.1	20.47	116.9	257.7	326.5	719.7	0.358	0.281
200	110	45	0.88	139.2	20.19	119.9	264.3	326.5	719.7	0.367	0.291
200	110	50	0.91	136.7	19.82	125.6	276.8	326.5	719.7	0.385	0.307
200	110	55	0.94	133.2	19.32	131.1	289	326.5	719.7	0.402	0.323
200	110	60	0.985	128.7	18.66	136.6	301.1	326.5	719.7	0.418	0.34
200	110	70	1	130.7	18.95	133.5	294.4	326.5	719.7	0.41	0.338

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9.

variations. When T_e varies, the $(COP)_c$ deviates only a little from the optimum line, which indicates that T_e variation has the least effect on the off-design performance.

For the heating mode the evaporator component will be exposed to much lower outdoor temperatures. Therefore, the parametric study and off-design study for T_e variation are extended down to −17.8°C (0°F) for the evaporator temperature. $(COP)_h$ vs T_e is plotted in Fig. 10 for T_b equal to 93.3°C (200°F) and T_c equal to 26.7°C (80°F) or 43.3°C (110°F). The results indicate that values of $(COP)_h$ are not affected very much by the low temperatures. This is of particular advantage for an ejector heat pump utilized in the heating mode.

4.6 METHODS TO IMPROVE COP

There are two possible methods to improve the $(COP)_c$ for an ejector heat pump.

4.6.1 Regeneration

An ejector heat pump with two heat exchangers is shown in Fig. 11. The heat exchanger on the right is for the preheating of the refrigerant liquid before it enters the boiler. The

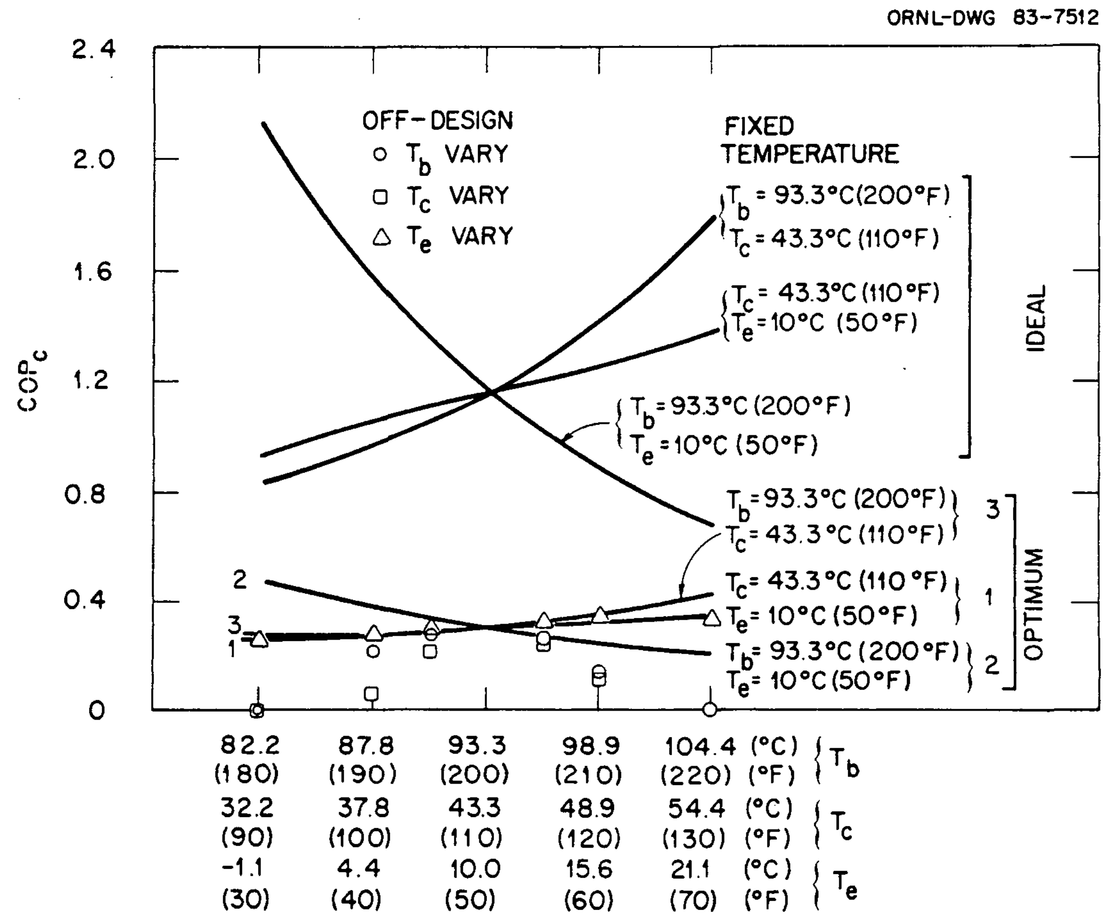

Fig. 9. Off-design and parametric study results for R-11.

heat exchanger on the left is for the subcooling of refrigerant liquid before it enters the expansion valve.

Table 6 illustrates the regeneration results for R-113 as the working fluid with T_b equal to 76.1°C (169°F), T_c equal to 50°C (122°F), and T_e equal to 7.1°C (44.7°F). The calculation method is based on the temperature increments of T_4 and T_{10}. The result indicates that $(COP)_c$ increases from 0.181 to 0.232, which is a 28% improvement.

Table 7 illustrates the regeneration results for R-113 as the working fluid with T_b equal to 104.4°C (220°F), T_c equal to 50°C (122°F), and T_e equal to 7.1°C (44.7°F). The $(COP)_c$ increases from 0.277 to 0.393, which is a 42% improvement.

Table 8 illustrates the regeneration results for R-11 as the working fluid with T_b equal to 93.3°C (200°F), T_c equal to 43.3°C (110°F), and T_e equal to 10°C (50°F). The $(COP)_c$ increases from 0.307 to 0.36, which is a 17% improvement.

Table 9 illustrates the regeneration results for R-11 as the working fluid with T_b equal to 137.8°C (280°F), T_c equal to 43.3°C (110°F), and T_e equal to 10°C (50°F). The $(COP)_c$ increases from 0.438 to 0.559, which is a 28% improvement.

Therefore, the improvement in $(COP)_c$ depends on different refrigerants and different operating conditions. Notice the entrainment rate, $\dot{m}_4/\dot{m}_0$, decreased slightly as the temperature T_4 increased due to the regeneration.

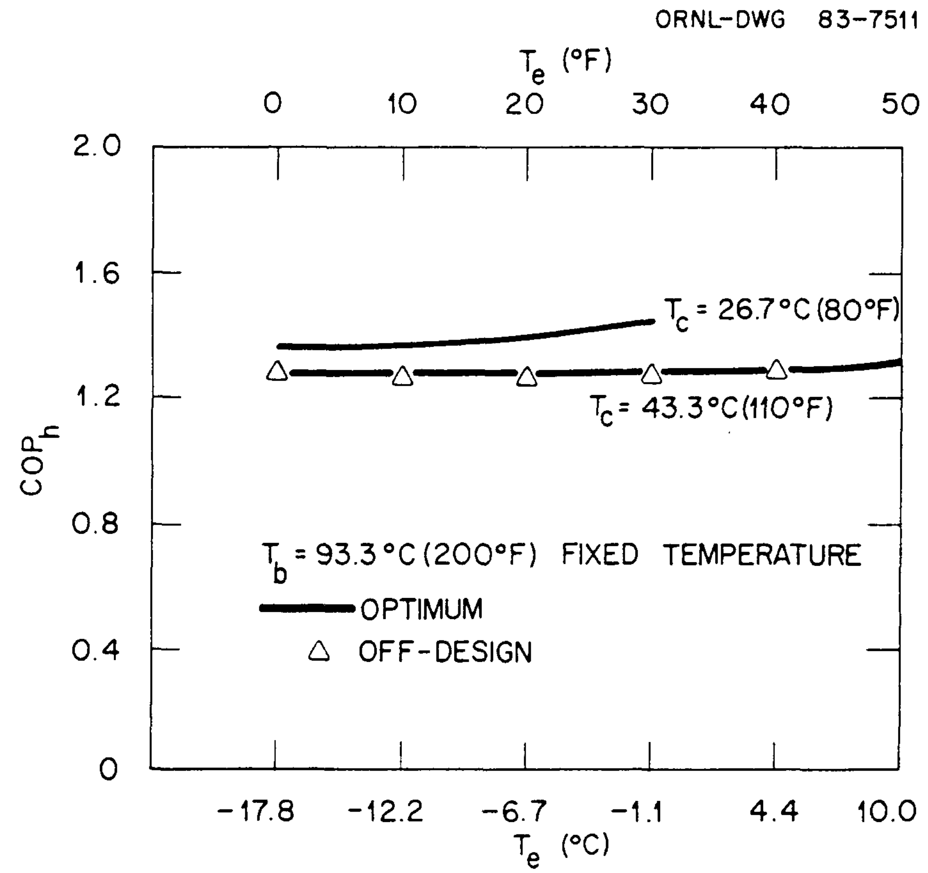

Fig. 10. Coefficient of performance for ejector heat pump at heating mode.

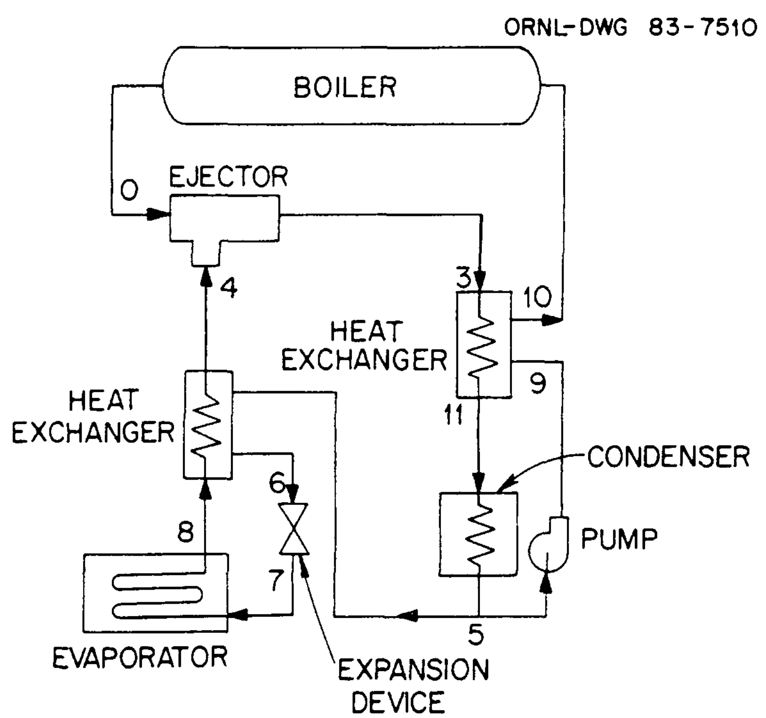

Fig. 11. Ejector heat pump with regeneration.

Table 6. Regeneration comparisons for R-113 with T_b = 76.1°C (169°F), T_c = 50°C (122°F), and T_e = 7.1°C (44.7°F)[a]

T8 (°F)	T4 (°F)	T5 (°F)	T6 (°F)	T9 (°F)	T10 (°F)	T3 (°F)	T11 (°F)	$(COP)_c$	$\dot{m}_4/\dot{m}_0$
44.7	44.7	122	122	122.1	122.1	142.62	142.62	0.181	0.245
44.7	44.7	122	122	122.1	123.18	142.62	141.36	0.1816	0.245
44.7	44.7	122	122	122.1	131.1	142.62	132.03	0.1866	0.245
44.7	44.7	122	122	122.1	139	142.62	122.63	0.1919	0.245
44.7	64.7	122	109.24	122.1	122.1	146.63	146.63	0.1871	0.239
44.7	64.7	122	109.24	122.1	126.58	146.63	141.36	0.1899	0.239
44.7	64.7	122	109.24	122.1	134.44	146.63	132.03	0.195	0.239
44.7	64.7	122	109.24	122.1	142.38	146.63	122.63	0.201	0.239
44.7	84.7	122	96.18	122.1	122.1	150.19	150.19	0.1989	0.241
44.7	84.7	122	96.18	122.1	129.61	150.19	141.36	0.2040	0.241
44.7	84.7	122	96.18	122.1	137.48	150.19	132.03	0.2098	0.241
44.7	84.7	122	96.18	122.1	145.43	150.19	122.63	0.2159	0.241
44.7	104.7	122	82.81	122.1	122.1	154.18	154.18	0.2028	0.233
44.7	104.7	122	82.81	122.1	125.11	154.18	150.64	0.2048	0.233
44.7	104.7	122	82.81	122.1	132.94	154.18	141.36	0.2105	0.233
44.7	104.7	122	82.81	122.1	140.75	154.18	132.03	0.2164	0.233
44.7	104.7	122	82.81	122.1	148.66	154.18	122.63	0.2227	0.233
44.7	120	122	72.30	122.1	122.1	157.01	157.01	0.209	0.231
44.7	120	122	72.30	122.1	127.51	157.01	150.64	0.213	0.231
44.7	120	122	72.30	122.1	135.33	157.01	141.36	0.219	0.231
44.7	120	122	72.30	122.1	143.12	157.01	132.03	0.225	0.231
44.7	120	122	72.30	122.1	151.06	157.01	122.63	0.232	0.231

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9.

4.6.2 Two-Stage Cooling

The majority of steam-jet refrigeration units currently being installed are in two or more stages. When temperature change is large, the efficiency in cooling of multiple-stage ejector heat pumps will increase.

For example, a material may need to be cooled from 26.7°C (80°F) to 4.4°C (40°F). If the temperature allowance for heat transfer is 5.6°C (10°F), the evaporator must operate at −1.1°C (30°F) for one-stage cooling. Using R-11 as the working fluid, the ejector heat pump operating at temperature conditions T_b equal to 93.3°C (200°F), T_c equal to 43.3°C (110°F), and T_e equal to −1.1°C (30°F) will achieve a $(COP)_c$ of 0.271 (see Table 3).

For two-stage cooling, the material is first cooled from 26.7°C (80°F) to 15.6°C (60°F), then from 15.6°C (60°F) to 4.4°C (40°F). The first-stage evaporator is operating at 10°C (50°F); T_b is equal to 93.3°C (200°F), and T_c is equal to 43.3°C (110°F). With R-11 as the working fluid, the $(COP)_c$ equals 0.307. The second-stage $(COP)_c$ would be 0.271 for the evaporator operating at −1.1°C (30°F). Therefore, the resulting $(COP)_c$ is calculated as follows:

$$(COP)_c = \frac{1}{\dfrac{0.5}{0.307} + \dfrac{0.5}{0.271}} = 0.288$$

The efficiency has thus been improved 6.3%.

Table 7. Regeneration comparisons for R-113, with T_b = 104.4°C (220°F), T_c = 50°C (122°F), and T_e = 7.1°C (44.7°F)[a]

T8 (°F)	T4 (°F)	T5 (°F)	T6 (°F)	T9 (°F)	T10 (°F)	T3 (°F)	T11 (°F)	$(COP)_c$	$\dot{m}_4/\dot{m}_0$
44.7	44.7	122	122	122.3	122.3	163.6	163.6	0.277	0.414
44.7	44.7	122	122	122.3	126	163.6	159.9	0.28	0.414
44.7	44.7	122	122	122.3	134.9	163.6	150.6	0.288	0.414
44.7	44.7	122	122	122.3	143.9	163.6	141.4	0.296	0.414
44.7	44.7	122	122	122.3	152.8	163.6	132	0.306	0.414
44.7	44.7	122	122	122.3	161.9	163.6	122.6	0.315	0.414
44.7	64.7	122	109.24	122.3	122.3	169.4	169.4	0.2887	0.408
44.7	64.7	122	109.24	122.3	122.7	169.4	169	0.289	0.408
44.7	64.7	122	109.24	122.3	132	169.4	160	0.297	0.408
44.7	64.7	122	109.24	122.3	140.5	169.4	150.6	0.306	0.408
44.7	64.7	122	109.24	122.3	149.4	169.4	141.4	0.315	0.408
44.7	64.7	122	109.24	122.3	158	169.4	132	0.325	0.408
44.7	64.7	122	109.24	122.3	167.5	169.4	122.6	0.335	0.408
44.7	84.7	122	96.18	122.3	122.3	175.1	175.1	0.2999	0.402
44.7	84.7	122	96.18	122.3	128	175.1	169	0.305	0.402
44.7	84.7	122	96.18	122.3	137	175.1	160	0.314	0.402
44.7	84.7	122	96.18	122.3	146	175.1	151	0.323	0.402
44.7	84.7	122	96.18	122.3	155	175.1	141	0.333	0.402
44.7	84.7	122	96.18	122.3	164	175.1	132	0.344	0.402
44.7	84.7	122	96.18	122.3	173	175.1	122.6	0.355	0.402
44.7	104.7	122	82.8	122.3	122.3	180.9	180.9	0.31	0.394
44.7	104.7	122	82.8	122.3	125	180.9	178	0.312	0.394
44.7	104.7	122	82.8	122.3	134	180.9	169	0.321	0.394
44.7	104.7	122	82.8	122.3	143	180.9	160	0.33	0.394
44.7	104.7	122	82.8	122.3	151	180.9	151	0.34	0.394
44.7	104.7	122	82.8	122.3	160	180.9	141	0.351	0.394
44.7	104.7	122	82.8	122.3	169	180.9	132	0.362	0.394
44.7	104.7	122	82.8	122.3	178	180.9	122.6	0.374	0.394
44.7	120	122	72.3	122.3	122.3	185	185	0.321	0.393
44.7	120	122	72.3	122.3	129	185	178	0.328	0.393
44.7	120	122	72.3	122.3	138	185	169	0.337	0.393
44.7	120	122	72.3	122.3	147	185	160	0.347	0.393
44.7	120	122	72.3	122.3	155	185	151	0.358	0.393
44.7	120	122	72.3	122.3	164	185	141	0.369	0.393
44.7	120	122	72.3	122.3	173	185	132	0.381	0.393
44.7	120	122	72.3	122.3	182	185	122.6	0.393	0.393

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9.

Table 8. Regeneration comparisons for R-11, with T_b = 93.3°C (200°F), T_c = 43.3°C (110°F), and T_e = 10°C (50°F)[a]

T8 (°F)	T4 (°F)	T5 (°F)	T6 (°F)	T9 (°F)	T10 (°F)	T3 (°F)	T11 (°F)	$(COP)_c$	$\dot{m}_4/\dot{m}_0$
50	50	110	110	110.7	110.7	144.9	144.9	0.307	0.385
50	50	110	110	110.7	114	144.9	141	0.31	0.385
50	50	110	110	110.7	124	144.9	131	0.318	0.385
50	50	110	110	110.7	133	144.9	121	0.326	0.385
50	70	110	97.3	110.7	110.7	150.3	150.3	0.314	0.377
50	70	110	97.3	110.7	119	150.3	141	0.321	0.377
50	70	110	97.3	110.7	129	150.3	131	0.329	0.377
50	70	110	97.3	110.7	138	150.3	121	0.338	0.377
50	90	110	84.3	110.7	110.7	155.7	155.7	0.319	0.369
50	90	110	84.3	110.7	115	155.7	151	0.322	0.369
50	90	110	84.3	110.7	124	155.7	141	0.331	0.369
50	90	110	84.3	110.7	134	155.7	131	0.339	0.369
50	90	110	84.3	110.7	143	155.7	121	0.348	0.369
50	110	110	71.1	110.7	110.7	161	161	0.325	0.363
50	110	110	71.1	110.7	120	161	151	0.333	0.363
50	110	110	71.1	110.7	129	161	141	0.342	0.363
50	110	110	71.1	110.7	139	161	131	0.351	0.363
50	110	110	71.1	110.7	148	161	120.8	0.360	0.363

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9.

4.7 SELECTION OF REFRIGERANT

An extensive discussion about the selection of refrigerant for the ejector heat pump may be found in ref. 11. The general requirements are for a nontoxic, nonflammable, chemically stable or inert, simple compressible substance.[16] The desirable thermodynamic properties would be a high enthalpy of vaporization and a saturation region in the range necessary for the application.

With these physical and thermodynamic requirements as a basis, Hamner[11] has selected a list of refrigerants, which is shown in Table 10. The extremely high heat of vaporization of water along with its inherent stability make it attractive, but water will freeze below 0°C (32°F), and the pressure range is too low. Ammonia also has a high enthalpy of vaporization, but the operating pressure is too high.

Refrigerants 11, 113, and 114, three of the halocarbons, satisfy the criteria best for an ejector heat pump system. Therefore, calculations of COP for these three kinds of refrigerants were made at the same operating conditions: T_b equal to 93.3°C (200°F), T_c equal to 43.3°C (110°F), and T_e equal to 10°C (50°F). Results show that for R-11 the $(COP)_c$ is 0.307, for R-113 the $(COP)_c$ is 0.308, and for R-114 the $(COP)_c$ is 0.252. It is obvious that R-11 and R-113 give better performance than R-114. Therefore, R-11 and R-113 will be used for the gas dynamic analysis of the ejector in the next section.

Table 9. Regeneration comparisons for R-11, with T_b = 137.8°C (280°F), T_c = 43.3°C (110°F), and T_e = 10°C (50°F)[a]

T8 (°F)	T4 (°F)	T5 (°F)	T6 (°F)	T9 (°F)	T10 (°F)	T3 (°F)	T11 (°F)	$(COP)_c$	$\dot{m}_4/\dot{m}_0$
50	50	110	110	112.1	112.1	163.85	163.85	0.438	0.595
50	50	110	110	112.1	115.04	163.85	161.19	0.441	0.595
50	50	110	110	112.1	126.02	163.85	151.15	0.453	0.595
50	50	110	110	112.1	136.92	163.85	141.08	0.466	0.595
50	50	110	110	112.1	147.74	163.85	130.98	0.479	0.595
50	50	110	110	112.1	158.48	163.85	120.84	0.493	0.595
50	70	110	97.3	112.1	112.1	171.2	171.2	0.449	0.586
50	70	110	97.3	112.1	123.0	171.2	161.2	0.461	0.586
50	70	110	97.3	112.1	133.9	171.2	151.2	0.473	0.586
50	70	110	97.3	112.1	144.7	171.2	141.1	0.487	0.586
50	70	110	97.3	112.1	155.4	171.2	130.98	0.501	0.586
50	70	110	97.3	112.1	166	171.2	120.8	0.516	0.586
50	90	110	84.3	112.1	112.1	178.5	178.5	0.458	0.575
50	90	110	34.3	112.1	120.1	178.5	171.2	0.467	0.575
50	90	110	84.3	112.1	130.9	178.5	161.2	0.479	0.575
50	90	110	84.3	112.1	141.6	178.5	151.2	0.493	0.575
50	90	110	84.3	112.1	152.3	178.5	141.1	0.507	0.575
50	90	110	84.3	112.1	163	178.5	131	0.522	0.575
50	90	110	84.3	112.1	173.4	178.5	120.8	0.538	0.575
50	110	110	71.1	112.1	112.1	185.8	185.8	0.466	0.564
50	110	110	71.1	112.1	117.2	185.8	181.2	0.472	0.564
50	110	110	71.1	112.1	127.9	185.8	171.2	0.484	0.564
50	110	110	71.1	112.1	138.6	185.8	161.2	0.498	0.564
50	110	110	71.1	112.1	149.2	185.8	151.2	0.512	0.564
50	110	110	71.1	112.1	159.7	185.8	141.1	0.527	0.564
50	110	110	71.1	112.1	170.1	185.8	131	0.542	0.564
50	110	110	71.1	112.1	180.5	185.8	120.8	0.559	0.564

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9.

Table 10. Selection of refrigerants

Refrigerant number	Toxicity group[a]	1977 cost		T		P		Enthalpy of vaporization	
		\$/kg	\$/lb	°C	°F	kPa	psia	J/kg	Btu/lb
11	5a	1.1	0.50	7.2	45	54.2	7.863	1.86×10^5	80.105
12	6	1.1	0.50	7.2	45	388.7	56.373	1.90×10^5	81.937
21	4–5	31.8	14.45	7.2	45	94.6	13.725	2.43×10^5	104.53
113	4–5	1.4	0.62	7.2	45	20.8	3.021	1.58×10^5	68.18
114	6	1.9	0.87	7.2	45	115.4	16.737	1.35×10^5	58.022
142b	5a	4.4	2.00	7.2	45	189.7	27.506	2.08×10^5	89.766
152a	6	4.8	2.16	7.2	45	342.1	49.619	2.94×10^5	126.84
C318	6	18.3	8.30	7.2	45	169.4	24.572	1.11×10^5	48.017
717 (NH_3)	2	0.4	0.18	7.2	45	558.2	80.96	1.23×10^6	531.8
718 (H_2O)	6			7.2	45	1.02	0.148	2.48×10^6	1068.4
22				7.2	45	625.4	90.7	1.99×10^5	85.6
502				7.2	45	710	103	1.42×10^5	61

[a]Toxicity groups are rated from 1 to 6. Group 2 is defined as gases or vapors which, in concentrations of the order of 0.5 to 1% for durations of exposure of the order of 0.5 h, are lethal or produce serious injury. Group 4 is defined as gases or vapors which, in concentrations of the order of 2 to 2.5% for durations of exposure of the order of 2 h, are lethal or produce serious injury. Group 6 is defined as gases or vapors which, in concentrations of up to at least 20% by volume for durations of exposure of the order of 2 h, do not appear to produce injury. Groups 4–5 and 5a are defined in between groups 4 and 6.

5. GAS DYNAMIC ANALYSIS OF THE EJECTOR GEOMETRY

Gilbert and Hill[4] developed a finite difference computer program for calculating the detailed performance of air-to-air flows in a two-dimensional ejector with a symmetrical variable area mixing section. However, this finite difference flow model program is limited to the ideal or perfect gas as the working fluid. For perfect gas, the thermodynamic properties may be treated as constants or as functions of temperature only. When refrigerant vapor is used as a working fluid, the fluid properties depend on both pressure and temperature. Thus modifications must be made in order to analyze the real gas flow in the ejector.

Thupvongsa[12] made the main modifications in the fluid's properties and the energy equation of the finite difference program. The real gas properties of R-113 and R-114 were used in the program calculations. The energy equation was written in terms of the unknown time-averaged enthalpy, h, since the time-averaged temperature, T, could no longer be solved explicitly from the energy equation. Instead, the temperature was determined from the enthalpy and the pressure of the refrigerant vapor. Given the initial temperature, pressure, and velocity at the ejector entrance, with the specified ejector geometry and entrainment rate, the finite difference program would calculate the temperature, enthalpy, and velocity profiles in detail in the radial direction. When the tolerance of the cross-sectional area was converged, the average pressure of that axial location was also determined. The calculations then marched down the axial direction with very small steps.

Thupvongsa's study resulted in a low compression ratio (P_c/P_e) such as 1.02, which failed the purpose of using the ejector as a thermal compressor. Further efforts were made in the present study. Several attempts were made to improve Thupvongsa's program:

1. Increase the iteration times of the pressure gradient in order to meet the convergent criteria which would improve the accuracy of the average pressure at an axial location.
2. Increase the axial marching step size to reduce the computation time.
3. Replace the iteration method by Gilbert and Hill's interpolation method for convergence of the pressure gradient.
4. Use the optimum mixing section cross-sectional area (obtained from Sect. 3.2.4) and Conductron's optimum mixing section configuration[13] as the ejector geometry for the finite difference calculations.
5. Assume uniform velocity of the primary and secondary flows at the ejector entrance to reduce the mixing loss.
6. Assume uniform velocity and thermodynamic properties of the primary and secondary flows at the ejector entrance to minimize the mixing loss.

These attempts still result in low compression ratios. Therefore, a one-dimensional ideal gas dynamic analysis is employed to further estimate the optimum ejector geometry.

5.1 EJECTOR GEOMETRY

A typical ejector configuration is shown in Fig. 12. The motive gas nozzle has a characteristic convergent-divergent shape. Its throat is sized to give the maximum flow rate (usually choked flow) at the design inlet vapor conditions and depends on the cooling or heating capacity. The nozzle's divergent section is typically conical, and the theoretical area ratio (cone outlet area to the throat area) is found through conventional supersonic flow equations. The cone angles (total included angle) employed in the divergent nozzle cone range from 8 to 15 degrees, with 10 to 12 degrees as the most common range.[7] The theoretical area ratio and an assumed divergent cone angle design will define the theoretical length.

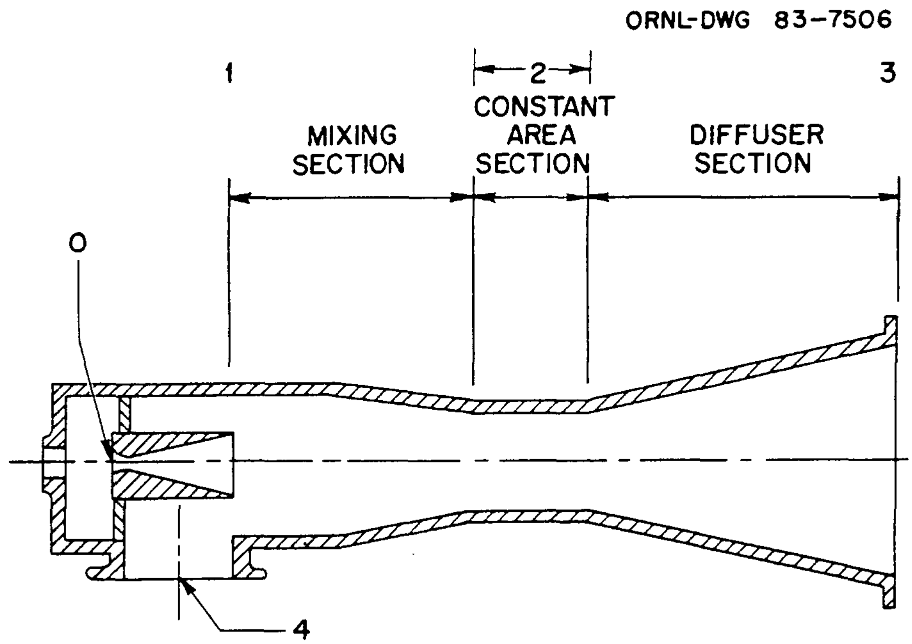

Fig. 12. Typical ejector configuration.

The mixing section is conical. According to Conductron's study[13] the parallel-convergent configuration gives the best performance. The theoretical area ratio is found through conventional gas dynamic equations. The cone angle employed will determine the axial length of the convergent section. The constant-area section is for shock diffusion.[7] The subsonic diffuser section is always conical in shape with an included angle range of 5 to 12 degrees, although 8 to 10 degrees is most common. The area ratio may be determined from states 2 and 3 by the isentropic compression equation.

5.2 DESIGN CALCULATIONS

An example of the design calculations is presented here, using R-11 as the working fluid. Assumed conditions are boiler temperature of 93.3°C (200°F), boiler pressure of 706.7 kPa (102.5 psia), condenser temperature of 43.3°C (110°F), condenser pressure of

192.4 kPa (27.9 psia), evaporator temperature of 10°C (50°F), and evaporator pressure of 60.5 kPa (8.78 psia). The design will normally proceed from the evaporator's viewpoint since that is where the load is imposed. Based on the calculation procedures in Sect. 4.2, the primary mass flow rate is 3.55 kg/m (7.83 lb/m), the secondary mass flow rate is 1.35 kg/m (2.98 lb/m) for 1 ton of cooling capacity. At the mixing section the throat Mach number (M_2) is determined to be 0.906, the pressure (P_2) is determined to be 136.4 kPa (19.78 psia), and the cross-sectional area (A_2) is determined to be 0.000085 m^2 (0.00091 ft^2). The general configuration of an ejector is shown in Fig. 12. According to one-dimensional gas dynamic theory there must be a shock existing in the mixing section to convert the supersonic flow into subsonic flow.

The following assumptions are made in order to utilize some of the gas dynamic equations:

1. The refrigerant vapor is treated as perfect gas.
2. Isentropic expansion and compression processes occur in the convergent and divergent sections of the ejector.
3. Friction effect is neglected in the flow processes.
4. There is a normal shock appearing at the cross section between the parallel and convergent sections as illustrated in Fig. 13.
5. There is an adiabatic process for the flow.

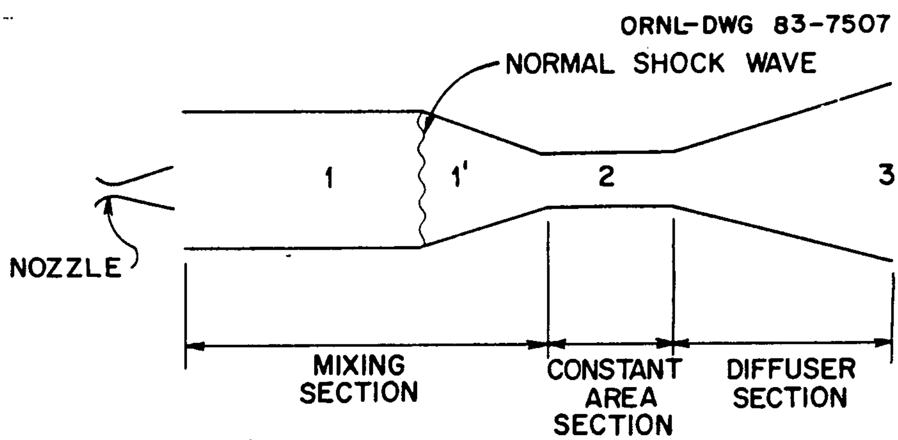

Fig. 13. Normal shock wave in the ejector.

Thus the known quantities are P_1 (P_e), P_2, and M_2; the unknown quantities are M_1, $M_{1'}$, and $P_{1'}$. The subscript 1 is designated for the state before the normal shock wave, and 1′ is for the state right after the normal shock wave. Two equations of normal shock waves in perfect gas are[14]

$$M_{1'} = \left(\frac{M_1^2 + \dfrac{2}{k-1}}{\dfrac{2k}{k-1} M_1^2 - 1} \right)^{1/2} , \tag{5.1}$$

$$\frac{P_{1'}}{P_1} = \frac{1 + kM_1^2}{1 + kM_{1'}^2} \tag{5.2}$$

The isentropic equation between states 1′ and 2 is

$$\frac{P_{1'}}{P_2} = \frac{\left\{1 + \dfrac{k-1}{2}M_2^2\right\}^{k/(k-1)}}{\left\{1 + \dfrac{k-1}{2}M_{1'}^2\right\}^{k/(k-1)}} . \tag{5.3}$$

Solving Eqs. 5.1 through 5.3 simultaneously results in M_1 equal to 1.65, $M_{1'}$ equal to 0.625, and $P_{1'}$ equal to 172.6 kPa (25.03 psia) for the present example. From the isentropic compression equation between states 2 and 3 (with M_2, P_2, and P_3 known), M_3 is solved to be 0.44. The cross-sectional areas of A_1 and A_3 may be solved from the isentropic Eqs. 5.4 and 5.5 as follows:

$$\frac{A_1}{A_2} = \frac{M_2\left[\dfrac{2}{(k+1)}\left\{1 + \dfrac{k-1}{2}M_{1'}^2\right\}\right]^{(k+1)/2(k-1)}}{M_{1'}\left[\dfrac{2}{(k+1)}\left\{1 + \dfrac{k-1}{2}M_2^2\right\}\right]^{(k+1)/2(k-1)}} , \tag{5.4}$$

$$\frac{A_3}{A_2} = \frac{M_2\left[\dfrac{2}{(k+1)}\left\{1 + \dfrac{k-1}{2}M_3^2\right\}\right]^{(k+1)/2(k-1)}}{M_3\left[\dfrac{2}{(k+1)}\left\{1 + \dfrac{k-1}{2}M_2^2\right\}\right]^{(k+1)/2(k-1)}} . \tag{5.5}$$

The cross-sectional areas of the nozzle throat A_t and the nozzle exit A_j may be solved from the continuity equations, $A_j = \dot{m}_0/\rho_j V_j$ and $A_t = \dot{m}_0/\rho_t V_t$. The axial lengths of the convergent and divergent sections are calculated by a total angle of 10 degrees. The axial lengths of the constant area sections have no effect on the flow because of the no-friction assumption. The resultant cross-sectional areas and diameters are listed in Table 11 and illustrated in Fig. 14. For the same operating conditions, an ejector geometry for R-113 with a 1-ton cooling capacity is shown in Fig. 15.

Table 11. Resultant cross-sectional areas and diameters of an ejector

State	Area		Diameter	
	cm^2	ft^2	cm	ft
t	0.17	0.00018	0.46	0.01517
j	0.55	0.00059	0.83	0.02733
1	0.98	0.00106	1.12	0.03677
2	0.85	0.00091	1.04	0.03410
3	1.25	0.00135	1.26	0.04148

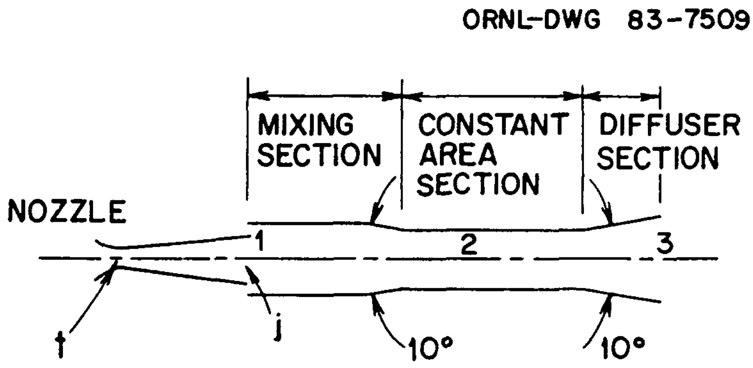

Fig. 14. An ejector geometry for R-11 with 1-ton cooling capacity.

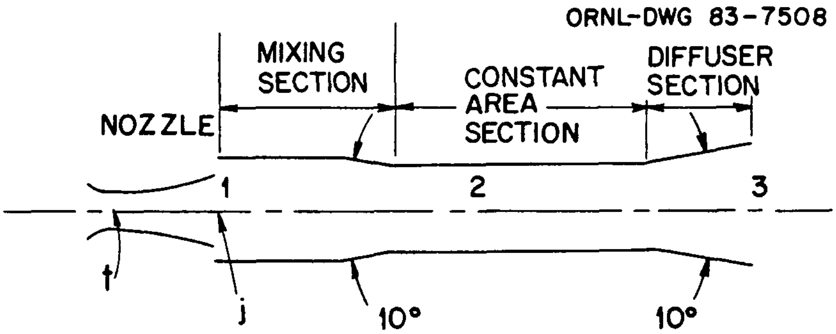

Fig. 15. An ejector geometry for R-113 with 1-ton cooling capacity.

6. SUMMARY AND CONCLUSIONS

An analysis of the ejector-compression heat pump cycle is developed which yields COP values for the ideal cycle and the real gas cycle. The ejector performance is optimized on the basis of Elrod's theory.[2] Accordingly, the optimum area ratio at the mixing section throat may be determined. The calculation scheme for an ejector heat pump cycle utilizing refrigerant vapor as the working fluid follows Chen's study.[5] Additionally, the parametric study which calculates the COP, entrainment rate, and ejector efficiency for different operating conditions; the sensitivity study which determines the effects of nozzle efficiency and diffuser efficiency on the overall ejector heat pump COP; and the off-design study which estimates the COP for an ejector heat pump operating at the off-design conditions are included in the investigation.

The analyses are independent of fluid species, but emphasis is placed on the halocarbon refrigerants, particularly R-11, R-113, and R-114, which fall within the thermodynamic and physical criteria for use in ejector heat pumps driven by low-grade thermal energy. Neither the azeotropic nor the nonazeotropic mixture refrigerants are analyzed. Regeneration and two-stage cooling are the possible methods to improve the ejector heat pump COP.

The parametric study results presented in Fig. 8 indicate that the COP is proportional to the boiler temperature. In order to achieve high COP, the boiler temperature may be extended as high as the critical temperature, which is 198°C (388.4°F) for R-11 and 214°C (417.4°F) for R-113. Calculated optimum $(COP)_c$ results are shown in Table 12. For R-113, the boiler temperature increases from 82.2°C (180°F) to 204.4°C (400°F); the $(COP)_c$ also increases from 0.261 to 0.48, which is an 84% increase. For R-11, the boiler temperature increases from 82.2°C (180°F) to 182.2°C (360°F), and the $(COP)_c$ increases from 0.257 to 0.514, which is a 100% increase. However, the boiler pressures for both refrigerants increase substantially, particularly at high operating temperatures.

The off-design study indicates that the COP of the ejector heat pump decreases greatly when it operates at off-design conditions. However, the boiler temperature can be controlled by a thermostat at the heat supply. In the heating mode the condenser is on the warm temperature side, which does not vary much. The evaporator may be exposed to cold weather and temperature variation, which does not affect the overall efficiency much (see Fig. 10). Therefore, the ejector heat pump is more attractive in the heating mode operation.

In the future geometrical design of the ejector, the perfect gas equations may be replaced by real gas equations. Friction should be considered for the flow in the ejector. The axial length of the constant-area section could be calculated by the Fanno line equation (adiabatic flow with friction) and the momentum equation. The location and the kind of shock wave would also affect the design of the ejector geometry.

In conclusion, the ejector heat pump device may be a good substitute for compression refrigeration where relatively low-temperature [~93°C (~200°F)] heat sources are readily

Table 12. Results of entrainment rate, ejector efficiency, and $(COP)_c$ by varying boiler temperatures for R-113 and R-11[a]

Refrigerant number	T_b (°F)	P_b (psia)	T_e (°F)	P_e (psia)	T_c (°F)	P_c (psia)	$\dot{m}_4/\dot{m}_0$	η_E	$(COP)_c$
113	180	41	50	3.43	110	12.75	0.35	0.35	0.261
113	200	55	50	3.43	110	12.75	0.43	0.34	0.308
113	220	71	50	3.43	110	12.75	0.49	0.32	0.338
113	240	91	50	3.43	110	12.75	0.55	0.31	0.369
113	260	114	50	3.43	110	12.75	0.61	0.31	0.394
113	280	142	50	3.43	110	12.75	0.66	0.297	0.412
113	300	174	50	3.43	110	12.75	0.70	0.29	0.43
113	320	210	50	3.43	110	12.75	0.75	0.285	0.444
113	340	252	50	3.43	110	12.75	0.78	0.278	0.454
113	360	298	50	3.43	110	12.75	0.81	0.272	0.461
113	380	350	50	3.43	110	12.75	0.84	0.27	0.475
113	400	408	50	3.43	110	12.75	0.87	0.27	0.48
11	180	79.4	50	8.78	110	27.9	0.314	0.33	0.257
11	200	102.5	50	8.78	110	27.9	0.385	0.322	0.307
11	220	130.3	50	8.78	110	27.9	0.447	0.316	0.349
11	240	163.3	50	8.78	110	27.9	0.501	0.308	0.382
11	260	202.2	50	8.78	110	27.9	0.55	0.30	0.413
11	280	247.6	50	8.78	110	27.9	0.595	0.296	0.438
11	300	300.2	50	8.78	110	27.9	0.634	0.29	0.46
11	320	360.7	50	8.78	110	27.9	0.665	0.287	0.478
11	340	430	50	8.78	110	27.9	0.698	0.286	0.499
11	360	509	50	8.78	110	27.9	0.715	0.285	0.514

[a]Temperatures may be converted by the equation °C = (°F − 32) × 5/9. Pressures may be converted by the equation kPa = psia × 6.895.

available. The ejector compressor is simple and easy to fabricate, requiring the least maintenance effort. Despite the low COP values obtained in this analysis, the ejector heat pump operating in the heating mode should offer better performance than direct furnace heating or electrical resistance type heating.

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APPENDIX

Computer Program Listing

For a selected refrigerant (R-11, R-12, R-22, R-113, R-114, or R-502) and a given set of the input values, including diffuser efficiency, nozzle efficiency, specific heat ratio, gas constant, and the operating conditions for the ejector heat pump (i.e., boiler temperature, evaporator temperature, and condenser temperature), the main program will calculate the entrainment rate, the ideal entrainment rate, the ejector efficiency, the thermodynamic states of the ejector heat pump cycle, and the coefficient of performance [$(COP)_{c\text{-}ideal}$, $(COP)_c$, and $(COP)_h$] on the basis of the optimum mixing section area of the ejector from Elrod's theory.[2]

The subroutine AREA calculates the cross-sectional areas of the ejector geometry for 1 ton of cooling capacity using the one-dimensional gas dynamic equations. Based on the obtained ejector geometry, the subroutine MACH uses conservation equations of momentum and energy for mixing to determine the state 1 flow velocity, enthalpy, and Mach number, which may be compared with the value obtained from the perfect gas equations (results obtained from subroutine AREA).

The subroutine TABLES provides the correct values for constants used in the thermodynamic properties subprograms.[17] The author incorporated the constants for R-113 in the subroutine.

The following subprograms used to calculate the refrigerant thermodynamic properties may be found in ref. 18:

- Subroutine SATPRP is used to evaluate the saturation thermodynamic properties of a specified refrigerant given the saturation temperature.
- Subroutine TRIAL is used to determine remaining superheated vapor properties, given the pressure and one other property of a specified refrigerant.
- Function TSAT is used to evaluate the saturation temperature of a specified refrigerant given the saturation pressure.
- Subroutine VAPOR is used to evaluate the thermodynamic properties of the superheated vapor phase of a specified refrigerant given the temperature and pressure.
- Function SPVOL is used to evaluate the specific volume of the vapor phase of a specified refrigerant given the pressure and temperature.
- Subroutine SPFHT is used to calculate specific heat at constant volume, specific heat at constant pressure, and specific heat ratio.

```
C     INVESTIGATION OF EJECTOR HEAT PUMP BY ANALYTICAL METHOD
C
C     PURPOSE:
C          BASED ON ELROD'S THEORY TO DETERMINE THE OPTIMUM EJECTOR MIXING
C     SECTION AREA, TO EVALUATE THE ENTRAINMENT RATE, EJECTOR EFFICIENCY,
C     COP, AND THERMODYNAMIC STATES FOR AN EJECTOR HEAT PUMP CYCLE.
C          AN OPTIMUM EJECTOR GEOMETRY IS ESTIMATED UTILIZING ONE
C     DIMENSIONAL GAS DYNAMIC EQUATIONS.
C
C     INPUT PARAMETERS
C     1.    NR  -  REFRIGERANT NUMBER (11,12,22,113,114,502)
C           ED  -  DIFFUSER EFFICIENCY
C           EN  -  NOZZLE EFFICIENCY
C           RK  -  INITIAL VALUE OF SPECIFIC HEAT RATIO FOR REFRIGERANT VAPOR
C           RR  -  GAS CONSTANT FOR REFRIGERANT, (FT-LBF/LBM-R)
C     2.    TB  -  SATURATION TEMPERATURE OF BOILER (F)
C           TE  -  SATURATION TEMPERATURE OF EVAPORATOR (F)
C           TC  -  SATURATION TEMPERATURE OF CONDENSER (F)
C     3.    TO  -  SUPERHEAT VAPOR TEMPERATURE AT STATE 0 (F)
C           T4  -  SUPERHEAT VAPOR TEMPERATURE AT STATE 4 (F)
C     4.    CP6 -  CONST. PRESS. HEAT CAPACITY OF LIQUID REFRIGERANT
C                  THE AVERAGE VALUE OF STATES 6 AND 5 (BTU/LBM-R)
C           CP9 -  CONST. PRESS. HEAT CAPACITY OF LIQUID REFRIGERANT
C                  THE AVERAGE VALUE OF STATES 9 AND 5 (BTU/LBM-R)
C           CP10-  CONST. PRESS. HEAT CAPACITY OF LIQUID REFRIGERANT
C                  THE AVERAGE VALUE OF STATES 10 AND 9 (BTU/LBM-R)
C
C     OUTPUT PARAMETERS
C     (EC)P = ESTIMATED ENTRAINMENT EFFICIENCY BASED ON CONDUCTRON'S
C             PERFECT GAS EQUATION
C     CR     = COMPRESSION RATIO, PC/PE
C     RMF    = ENTRAINMENT RATE
C     RMFS   = IDEAL ENTRAINMENT RATE
C     EC     = EJECTOR EFFICIENCY
C     (COP)-WORK = ACTUAL COP INCLUDING PUMP WORK FOR COOLING
C     (COP)-NEG WORK = COP NEGLECTING PUMP WORK FOR COOLING
C     (COP)IDL = COP FOR IDEAL EJECTOR HEAT PUMP CYCLE FOR COOLING
C     (COP)LOW = COP FOR COOLING MODE WITHOUT REGENERATION, THE VALUE
C     IS NOT AFFECTED BY CP6, CP9, CP10 VALUES.
C     (COP)HT = COP FOR HEATING MODE WITHOUT REGENERATION.
C     XM = MACH NUMBER
C     A = CROSS-SECTIONAL AREAS OF THE EJECTOR
C     D = DIAMETERS OF THE EJECTOR
C     XL = AXIAL LENGTHS OF THE CONVERGENT AND DIVERGENT SECTIONS BASED
C          ON 10 DEGREE ANGLE.
C     DT = DIAMETER OF THE NOZZLE THROAT.
C     DJ = DIAMETER OF THE NOZZLE EXIT.
C
C     COMMENTS:
C     1. IF INPUT PARAMETERS TO=0., T4=0., THE PROGRAM WILL EXECUTE
C        AS TO=TB, T4=TE.
C     2. APPROXIMATE VALUES OF CP6, CP9, CP10 ARE ENOUGH.
C
C
      COMMON/REFRIG/NR
      READ(3,10)NR,ED,EN,RK,RR
      READ(3,30)TB,TE,TC
      READ(3,33)TO,T4
      READ(3,32)CP6,CP9,CP10
10    FORMAT(I10,4F10.5)
```

```
      CALL TABLES(NR)
      WRITE(6,20)NR,ED,EN,RK,RR
20    FORMAT(//,5X,'R-',I3,/,5X,'ED=',F10.5,5X,'EN=',F10.5,/,
     &5X,'K=',F10.5,5X,'R=',F10.5,/)
30    FORMAT(3F10.5)
      CALL SATPRP(TB,PB,VFB,SVB,HFB,HFG,HB,SFB,SB,IFLG)
      CALL SATPRP(TE,PE,VFE,SVE,HFE,HFG,H8,SFE,SQ8,IFLG)
      CALL SATPRP(TC,PC,SV5,VG,H5,HFG,HGC,S5,SGC,IFLG)
      P0=PB
      P4=PE
32    FORMAT(3F10.5)
33    FORMAT(2F10.5)
      IF(T4 .GT. TE) GO TO 34
      T4=TE
      SV4=SVE
      H4=H8
      S4=S8
      GO TO 35
34    CALL VAPOR(T4,P4,SV4,H4,S4,IERROR)
35    IF(TO .GT. TB) GO TO 36
      TO=TB
      SV0=SVB
      HO=HB
      SO=SB
      GO TO 37
36    CALL VAPOR(TO,PO,SVO,HO,SO,IERROR)
37    H7=H5+H8-H4
      P3=PC
      WRITE(6,31)TE,PE,TC,PC,H7,H8,H5
31    FORMAT(5X,'TE=',F10.5,5X,'PE=',F10.5,5X,'TC=',F10.5,5X
     &,'PC=',F10.5,/,5X,'H7=',F10.5,5X,'H8=',F10.5,5X,'H5=',F10.5)
      WRITE(6,40)TO,PO,HO,SO,T4,P4,H4,S4,P3,SVO
40    FORMAT(5X,'TO=',F10.5,5X,'PO=',F10.5,5X,'HO=',F10.5,5X,'SO=',
     &F10.5,/,5X,'T4=',F10.5,5X,'P4=',F10.5,5X,'H4=',F10.5,5X,
     &'S4=',F10.5,/,5X,'P3=',F10.5,5X,'SVO=',F10.5,/)
      WRITE(6,41)CP6,CP9,CP10
41    FORMAT(5X,'CP6=',F10.5,5X,'CP9=',F10.5,5X,'CP10=',F10.5)
      RK1=RK
      PT=PO*(2./(RK1+1.))**(RK1/(RK1-1.))
      TTS=TSAT(PT,IFLAG)
      TT=2.*(TO+460.)/(RK1+1.)-460.
      CALL SPFHT(TT,PT,CVT,CPT,RK3)
      WRITE(6,134)PT,TT,RK3,TTS
134   FORMAT(5X,'PT=',F10.5,5X,'TT=',F10.5,5X,'RK3=',F10.5,
     &5X,'TTS=',F10.5)
132   VT=(32.2*RK1*RR*2.*(TO+460.)/(RK1+1.))**0.5
      SVT=SVO/((2./(RK1+1.))**(1./(RK1-1.)))
      RMO=VT/SVT
      WRITE(6,61)P2,PT,SVT,VT,RK1
61    FORMAT(5X,'P2=',F10.5,5X,'PT=',F10.5,5X,'SVT=',F10.5,
     &5X,'VT=',F10.5,/,5X,'K1=',F10.5,/)
      RK2=RK
      CP=RK2*RR/778./(RK2-1.)
      WRITE(6,50)RK2,CP,CP2
50    FORMAT(5X,'K2=',F10.5,5SX,'CP=',F10.5,5X,'CP2=',F10.5,
     &/,5X,'##################################################')
      P43=P4/P3
      P3P2=1.0
      WRITE(6,80)
80    FORMAT(5X,'P3/P2',5X,'P4/P3',5X,'(K2M2**2+1)/(P3/P2)')
```

```
C
C     DO LOOP TO DETERMINE STATE 2 PRESSURE, P2, OF MIXER SECTION
C
      PTOL=0.00001
      DP=0.1
      DO 120 I=1,40
      P3P2=P3P2+DP
      SQM=778.*CP*(P3P2**((RK2-1.)/RK2)-1.)*2./ED/RK2/RR
      P4P3=(1.-RK2/((1./SQM-1.)/((1.-ED)*(1./P3P2)**((RK2-1.)
     &/RK2))+RK2))/P3P2
      ANS=(RK2*SQM+1.)/P3P2
      WRITE(6,90)P3P2,P4P3,ANS
90    FORMAT(3F10.5)
      DIFF1=P4P3-P43
      IF(ABS(DIFF1).LE.PTOL) GO TO 110
      IF(DIFF1) 100,110,120
100   P3P2=P3P2-DP
      DP=DP/2.0
120   CONTINUE
110   RM2=SQM**0.5
      P2=P3/P3P2
      WRITE(6,130)RM2,P2
130   FORMAT(5X,'M2=',F10.5,5X,'P2=',F10.5,//)
      TI=TSAT(P4,IFLAG)
      IF(SO .LT. S8) GO TO 148
      CALL TRIAL(TI,30.,P4,4,SO,1.0E-05,SV1,H1S,S1S,T1S,IEROR)
      GO TO 149
148   X=(SO-SFE)/(S8-SFE)
      SV1=(1-X)*VFE+X*SVE
      H1S=(1-X)*HFE+X*H8
149   V1=223.8*EN**0.5*(HO-H1S)**0.5
      A2=RMO*V1/(32.2*P3*144.*(ANS-P4P3))
      WRITE(6,150)P4,SO,H1S,X
150   FORMAT(5X,'P4=',F10.5,5X,'SO=',F10.5,5X,'H1S=',F10.5,5X,
     &'X=',F10.5)
      WRITE(6,160)RMO,V1,A2
160   FORMAT(5X,'MO=',F10.5,5X,'V1=',F10.5,5X,'A2=',F10.5
     &,///,5X,'****************************************************')
C
C     DO LOOP TO DETERMINE STATE 2 ENTROPY, S2, OF MIXER SECTION
C
      T2=TSAT(P2,IFLAG)
      CALL SATPRP(T2,PSAT,VF,VG,HF,HFG,HG,SF,S2,IFLAG)
      S2=S2-0.002
      STOL=0.2
      DS=0.002
      DO 220 I=1,40
      S2=S2+DS
      TI=TSAT(P2,IFLAG)
      CALL TRIAL(TI,30.,P2,4,S2,1.0E-05,SV2,H2,S2S,T2,IEROR)
      WRITE(6,170)P2,S2,S2S,T2,H2
170   FORMAT(5X,'P2=',F10.5,5X,'S2=',F10.7,5X,'S2S=',F10.5,5X,
     &'T2=',F10.5,/,5X,'H2=',F10.5)
      TI=TSAT(P3,IFLAG)
      CALL TRIAL(TI,30.,P3,4,S2,1.0E-05,V,H3S,S,T,IEROR)
      H3=(H3S-H2)/ED+H2
      RM4=RMO*(H3-HO)/(H4-H3)
      IF(RM4.LE.0.0)GO TO 499
```

```
      RIGHT=(RM0+RM4)/A2
      V2=((H3-H2)*2.*32.2*778.)**0.5
      RLEFT=V2/SV2
      WRITE(6,180)H3S,H3,RM4,RIGHT,V2,SV2,RLEFT
180   FORMAT(5X,'H3S=',F10.5,5X,'H3=',F10.5,5X,'RM4=',E12.5,
     &5X,'RIGHT=',F10.5,/,5X,'V2=',F10.5,5X,'SV2=',F10.5,
     &24X,'RLEFT=',F10.5,/,5X,'------------------------
     &----------------------------')
      DIFF2=RIGHT-RLEFT
      IF(ABS(DIFF2).LE.STOL)GO TO 230
      IF(DIFF2)210,230,220
210   S2=S2-DS
      DS=DS/2.0
220   CONTINUE
C
C     DO LOOP TO DETERMINE THE IDEAL ENTRAINMENT RATIO OF THE EJECTOR
C
230   IF(S0.LT.S4)GO TO 430
      S3SS=S0+0.001
      TOL=2.0
      DS=-0.001
      DO 300 I=1,40
      S3SS=S3SS+DS
      TI=TSAT(P3,IFLAG)
      CALL TRIAL(TI,30.,P3,4,S3SS,1.0E-05,V,H3SS,S,T,IEROR)
      RIGHT=(H3SS-H4)/(S3SS-S4)
      RLEFT=(H0-H4)/(S0-S4)
      WRITE(6,270)P3,S3SS,H3SS,RIGHT,RLEFT
270   FORMAT(5X,'P3=',F10.5,5X,'S3SS=',F10.5,5X,'H3SS=',F10.5,
     &/,5X,'RIGHT=',E12.5,5X,'RLEFT=',E12.5,/,5X,'++++++++++++++
     &++++++++++++++++++++++++++++++')
      IF(RIGHT.LT.0.0) GO TO 320
      DIFF3=RIGHT-RLEFT
      IF(ABS(DIFF3).LE.TOL)GO TO 310
      IF(DIFF3)300,310,320
320   S3SS=S3SS-DS
      DS=DS/2.0
300   CONTINUE
430   S3SS=SGC-0.001
      TOL=2.0
      DS=0.001
      DO 400 I=1,40
      S3SS=S3SS+DS
      TI=TSAT(P3,IFLAG)
      CALL TRIAL(TI,30.,P3,4,S3SS,1.0E-05,V,H3SS,S,T,IEROR)
      RIGHT=(H3SS-H4)/(S3SS-S4)
      RLEFT=(H0-H4)/(S0-S4)
      WRITE(6,270)P3,S3SS,H3SS,RIGHT,RLEFT
      IF(RIGHT.GT.0.0) GO TO 420
      DIFF3=RIGHT-RLEFT
      IF(ABS(DIFF3).LE.TOL)GO TO 310
      IF(DIFF3)420,310,400
420   S3SS=S3SS-DS
      DS=DS/2.0
400   CONTINUE
310   RMF=RM4/RM0
      RMFS=((H0-H3SS)**2.+(S3SS-S0)**2.)**0.5/((H3SS-H4)**2.+
     &(S4-S3SS)**2.)**0.5
```

```
      EC=RMF/RMFS
      ECP=RMF*(T4+460.)/(T0+460.)*((P3/P4)**((RK-1.)/RK)-1.)
     &/(1.-(P3/P0)**((RK-1.)/RK))
      CR=PC/PE
      WRITE(6,329)ECP,CR
329   FORMAT(5X,'(EC)-PERFECT GAS BASE =',F10.5,
     &/,5X,'COMPRESSION RATIO=',F10.5)
      WRITE(6,330)RMF,RMFS,EC
330   FORMAT(5X,'RMF=',F10.5,5X,'RMFS=',F10.5,5X,'EC=',F10.5,///)
C
C     TO DETERMINE THERMODYNAMIC STATES OF AN EJECTOR HEAT PUMP
C     CYCLE.
C
      TI=TSAT(P3,IFLAG)
      CALL TRIAL(TI,30.,P3,3,H3,1.0E-05,V,H,S3,T3,IEROR)
      T5=TC
335   T6=TC-(H5-H7)/CP6
      CP6=(0.2155-0.213)*((T5+T6)/2.-90.)/20.+0.213
      TT6=TC-(H5-H7)/CP6
      IF(ABS(T6-TT6) .GT. 0.3) GO TO 335
      WKP=SV5*(P0-PC)*144./778.
      H9=H5+WKP
      T9=(H9-H5)/CP9+TC
      H11=HGC
340   H11=H11+1.0
      H10=(RM0+RM4)*(H3-H11)/RM0+H9
336   T10=(H10-H9)/CP10+T9
      CP10=(0.22-0.2155)*((T10+T9)/2.-110.)/30.+0.2155
      TT10=(H10-H9)/CP10+T9
      IF(ABS(T10-TT10) .GT. 0.5) GO TO 336
      TI=TSAT(PC,IFLAG)
       CALL TRIAL(TI,30.,PC,3,H11,1.0E-05,V,H,S11,T11,IEROR)
      IF(T10.GE.T3)GO TO 340
      WRITE(6,331)
      WRITE(6,332)TB,PB,HFB,HB,SFB,SB
      WRITE(6,332)TC,PC,H5,HGC,S5,SGC
      WRITE(6,332)TE,PE,HFE,H8,SFE,S8
331   FORMAT(1X,'SATURATION THERMODYNAMIC PROPERTIES OF BOILER
     &, CONDENSER, AND EVAPORATOR',/,1X,'------------------------
     &---------------------------------------------',/
     &/,3X,'TEMP(F)',6X,'P(PSIA)',3X,'HF(BTU
     &/LB)',3X,'HG(BTU/LB)',2X,'SF(BTU/LBF)',2X,'SG(BTU/LBF)')
332   FORMAT(1X,F10.5,3X,F10.5,3X,F10.5,3X,F10.5,3X,F10.5,3X,F10.5)
      X7=(H7-HFE)/(H8-HFE)
      S7=(1-X7)*SFE+X7*S8
      T7=TE
      T8=TE
      WRITE(6,349)
      WRITE(6,350)T0,P0,H0,S0
      WRITE(6,351)T3,P3,H3,S3
      WRITE(6,352)T4,P4,H4,S4
      WRITE(6,353)TC,PC,H5,S5
      WRITE(6,354)T6,PC,H7
      WRITE(6,355)TE,PE,H7,S7
      WRITE(6,356)TE,PE,H8,S8
      WRITE(6,357)T9,P0,H9
      WRITE(6,358)T10,P0,H10
      WRITE(6,359)T11,PC,H11,S11
```

```
350   FORMAT(1X,'0',5X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
351   FORMAT(1X,'3',5X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
352   FORMAT(1X,'4',5X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
353   FORMAT(1X,'5',5X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
354   FORMAT(1X,'6',5X,F10.5,5X,F10.5,5X,F10.5)
355   FORMAT(1X,'7',5X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
356   FORMAT(1X,'8',5X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
357   FORMAT(1X,'9',5X,F10.5,5X,F10.5,5X,F10.5)
358   FORMAT(1X,'10',4X,F10.5,5X,F10.5,5X,F10.5)
359   FORMAT(1X,'11',4X,F10.5,5X,F10.5,5X,F10.5,5X,F10.5)
349   FORMAT(///,1X,'STATE',2X,'TEMP(F)',8X,'P(PSIA)',8X,'H(BTU/LB)',
     &6X,'S(BTU/LBF)')
C
C     TO CALCULATE COP FOR 1 TON COOLING CAPACITY.
C
      QE=200.
      RMF4=QE/(H8-H7)
      RMFO=RMF4/RMF
      QG=RMFO*(HO-H10)
      QC=QG+QE
      ACOP=QE/(QG+WKP*RMFO)
      COP=QE/QG
      COPM=(TE+460.)*(TO-TC)/(TO+460.)/(TC-TE)
      TG=TO+20.
      TO=TC-20.
      COPMM=(TE+460.)*(TG-TO)/(TG+460.)/(TO-TE)
      COPL=RMF*(H8-H7)/(HO-H5)
      COPHT=(1.+RMF)*(H3-H5)/(HO-H5)
      WRITE(6,360)QE,RMF4,RMFO,QG,QC,ACOP,COP,COPM,COPMM,COPL,COPHT
360   FORMAT(1X,///,'*******************************************',
     &/,5X,'FOR 1 TON COOLING CAPACITY',/,5X,'QE=',F10.5,
     &'(BTU/MIN)',/,5X,'M4=',F10.5,'(LB/MIN)',
     &/,5X,'MO=',F10.5,'(LB/MIN)',/,5X,'QG=',F10.5,'(BTU/MIN)',
     &/,5X,'QC=',F10.5,'(BTU/MIN)',/,5X,'(COP)-WORK=',F10.5,/,5X,
     &'(COP)-NEG WORK=',F10.5,/,5X,'(COP)IDL=',F10.5,
     &/,5X,'(COP)MAX=',F10.5,5X,'(COP)LOW=',F10.5,3X,'(COP)HT=',F10.5//)
      COOL=1955./QG
      COOLT=COOL*200.
      WRITE(6,361)COOLT,COOL
361   FORMAT(1X,'FOR A CONVENTIONAL 2000 CC. AUTOMOBILE THE
     &HEAT',/,5X,' ENERGY FROM ENGINE IS 1955 BTU/MIN',/,5X,'COOLING
     & EFFECT=',F10.5,'(BTU/MIN)',2X,'OR',F10.5,'(TONS)',///)
C
C     THIS SECTION IS FOR THE DESIGN OF THE EJECTOR GEOMETRY
C     ASSUMPTIOS ARE MADE:
C     1. A NORMAL SHOCK EXISTS AT THE INTERSECTION OF STRAIGHT
C        PIPE SECTION AND CONVERGENT SECTION.
C     2. BASED ON PERFECT GAS ISENTROPIC PROCESS, NORMAL SHOCK,
C        THE CROSS-SECTIONAL AREA OF THE STRAIGHT PIPE MAY BE DETERMINED.
C        AND THEN V(SEC) MAY BE DETERMINED FROM M=VA/SV.
C     3. AFTER EXPANSION R-11 BECOMES TWO-PHASE MIXTURE,
C        THE AVERAGE SPECIFIC VOLUME IS CALCULATED BY THE QUALITY
C        OF THE TWO-PHASE MIXTURE. THE INVERSE
C        OF SV1 OF THE MIXTURE SPECIFIC VOLUME IS ASSUMED AS THE
C        AVERAGE DENSITY, FOR MASS FLOW RATE = V1*A1/SV1.
      APRI=RMFO/60.*SV1/V1
      THROT=RMFO/60./RMO
      RMP=RMFO/60.
```

```
      RMS=RMF4/60.
      A2=A2*THROT
      CALL AREA(PE,P2,PC,RM2,A2,RK,A1)
      ASEC=A1-APRI
      VSEC=RMF4/60.*SVE/ASEC
      DT=(THROT/3.1416)**0.5*2.
      D1=(A1/3.1416)**0.5*2.
      DJ=(APRI/3.1416)**0.5*2.
      D2=(A2/3.1416)**0.5*2.
      XLJ=(DJ-DT)/2./0.087489
      WRITE(6,362)THROT,DT,A2,APRI,ASEC,A1,D1,DJ,D2,XLJ
362   FORMAT(1X,'----------------------------------------',/1X,
     &'EJECTOR GEOMETRY',/,1X,'THROT=',F10.5,'(FT2)',5X,'DT=',F10.5,
     &'(FT)',5X,'A2=',F10.5,'FT2',/,1X,'APRI=',F10.5,
     &'FT2',5X,/,1X,'ASEC=',F10.5,'FT2',5X,/,1X,'A1=',F10.5,'FT2',
     &/,1X,'DIAMETERS: D1=',F10.5,'FT',5X,'DJ=',
     &F10.5,'FT',5X,'D2=',F10.5,'FT',/,12X,'XLJ=',F10.5,'(FT)',/)
      RO1=1./SV1
      ROE=1./SVE
      DX=D2/0.8*5.0
      CX=D2/0.8*(10.75-5.0)
      EX=DX+CX
      FX=D2+2.*EX*0.08748866
      WRITE(6,363)SV1,SVE,T1S,RO1,ROE,VSEC,CX,DX,EX,FX
363   FORMAT(1X,'SV1=',F10.5,'FT3/LB',5X,'SVE=',F10.5,'FT3/LB'
     &,5X,'T1S=',F10.5,'F',/,5X,'RO1=',F10.5,'LB/FT3',5X,'ROE=',F10.5
     &,'LB/FT3',//,1X,'VSEC=',F10.5,' FT/S',/,1X,'LENGTH IN FT
     & CX=',F10.5,'  DX=',F10.5,'  EX=',F10.5,'FX=',F10.5)
      CALL MACH(RMP,RMS,V1,VSEC,HO,H4,PE,RK,RR)
C
C
C     THIS SECTION IS FOR CONVERGE CHECKING
C
C
      IF(ABS(DIFF1).GT.PTOL)WRITE(6,510)DIFF1,PTOL
      IF(ABS(DIFF2).GT.STOL)WRITE(6,520)DIFF2,STOL
      IF(ABS(DIFF3).GT.TOL)WRITE(6,530)DIFF3,TOL
      IF(P4.GE.P2 .OR. P2.GE.P3) WRITE (6,540)
      IF(T4.GT.T5 .OR. T8.GT.T6)WRITE(6,550)
      IF(T9.GT.T11 .OR. T10.GT.T3)WRITE(6,560)
      TSOS4=SO+RMF*S4
      TS2=(RMF+1.)*S2
      IF(TSOS4.GT.TS2 .OR. S2 .GT. S3) WRITE(6,570)TSOS4,TS2,S2,S3
510   FORMAT(5X,'***P2 TRIAL DOES NOT CONVERGE***DIFF1=',
     &F10.5,5X,'PTOL=',F10.5)
520   FORMAT(5X,'***S2 TRIAL DOES NOT CONVERGE***DIFF2=',
     &F10.5,5X,'STOL=',F10.5)
530   FORMAT(5X,'***IDEAL ENTRAINMENT TRIAL DOES NOT CONVERGE***
     &DIFF3=',F10.5,5X,'TOL=',F10.5)
540   FORMAT(5X,'***WARNING***CHECK P4<P2<P3 ? ***')
550   FORMAT(5X,'***WARNING***CHECK T4<T5 AND T8<T6 ? ***')
560   FORMAT(5X,'***WARNING***CHECK T9<T11 AND T10<T3 ? ***')
570   FORMAT(5X,'***WARNING*** TSOS4=',F10.5
     &,5X,'TS2=',F10.5,5X,'S2=',F10.5,5X,'S3=',F10.5)
      GO TO 501
499   WRITE(6,500)RM4
500   FORMAT(1X,'M4=',F10.5,2X,'THE CONDITIONS AT MIXER SECTION
     & MAKE ZERO ENTRAINMENT EFFICIENCY FOR THE EJECTOR')
501   STOP
      END
```

```
      SUBROUTINE AREA(P1,P3,P4,XM3,A3,GAMMA,A1)
      Y3=(1.+(GAMMA-1.)/2.*XM3**2.)**(GAMMA/(GAMMA-1.))
      XM1=1.5
      DXM=0.5
40    XM1=XM1+DXM
      XM2=((XM1**2.+2./(GAMMA-1.))/(2.*GAMMA/(GAMMA-1.)
     &*XM1**2.-1.))**0.5
      Y2=(1.+(GAMMA-1.)/2.*XM2**2.)**(GAMMA/(GAMMA-1.))
      P2R=Y3/Y2*P3
      W1=XM1**2.*(GAMMA-1.)/2.+1.
      W2=XM2**2.*(GAMMA-1.)/2.+1.
      P2L=(W1/W2)**0.5*XM1/XM2*P1
      WRITE(6,20)XM1,P2L,P2R
20    FORMAT(1X,'XM1=',F10.5,5X,'P2L=',F10.5,5X,'P2R=',F10.5)
      IF(ABS(P2L-P2R).LE.0.0001)GO TO 50
      IF(P2L.LT.P2R)GO TO 40
      XM1=XM1-DXM
      DXM=DXM/2.
      GO TO 40
30    CONTINUE
50    XM4=(((P3/P4*Y3)**((GAMMA-1.)/GAMMA)-1.)*2./(GAMMA-1.))**0.5
      P2=P2L
      Z1=(2./(GAMMA+1.)*(1.+(GAMMA-1.)/2.*XM1**2.))**((GAMMA
     &+1.)/2./(GAMMA-1.))
      Z2=(2./(GAMMA+1.)*(1.+(GAMMA-1.)/2.*XM2**2.))**((GAMMA
     &+1.)/2./(GAMMA-1.))
      Z3=(2./(GAMMA+1.)*(1.+(GAMMA-1.)/2.*XM3**2.))**((GAMMA
     &+1.)/2./(GAMMA-1.))
      Z4=(2./(GAMMA+1.)*(1.+(GAMMA-1.)/2.*XM4**2.))**((GAMMA
     &+1.)/2./(GAMMA-1.))
      A2=A3*XM3/XM2*Z2/Z3
      A1=A2
      A4=A3*XM3/XM4*Z4/Z3
      D1=(4.*A1/3.1416)**0.5
      D2=(4.*A2/3.1416)**0.5
      D3=(4.*A3/3.1416)**0.5
      D4=(4.*A4/3.1416)**0.5
      XL1=(D1-D3)/2./0.087489
      XL4=(D4-D3)/2./0.087489
      XL2=(D2-D3)/2./0.087489
      XL3=3.*D3
      WRITE(6,10)GAMMA,P1,XM1,A1,D1,XL1,P2,XM2,A2,D2,XL2,
     &P3,XM3,A3,D3,XL3,P4,XM4,A4,D4,XL4,XL3
10    FORMAT(1X,'GAMMA=',F10.5,/,4X,'P(PSIA)',8X,'XM',13X,'A(FT2)',
     &9X,'D(FT)',10X,'XL(FT)',4(/,1X,5(F10.5,5X))
     &,/1X,'3*D3=',F10.5,'(FT)')
      RETURN
      END
```

```
      SUBROUTINE MACH(RMP,RMS,VP,VS,HO,HS,P1,GAMMA,R)
      RM1=RMP+RMS
      V1=(RMP*VP+RMS*VS)/RM1
      H1=(RMP*HO+RMS*(HS+VS**2./2./32.2/778.))/RM1-V1**2./2.
     &/32.2/778.
      TI=TSAT(P1,IFLAG)
      CALL TRIAL(TI,30.,P1,3,H1,1.0E-05,SV1,HDUM,S1,T1,IERROR)
      WRITE(6,40)RMP,VP,HO,RMS,VS,HS,RM1,T1,P1,V1,SV1,H1,S1
40    FORMAT(5X,'RMP=',F10.5,'LBM/S',5X,'VP=',F10.5,'FT/S',
     &5X,'HO=',F10.5,'BTU/LBM',/,5X,'RMS=',F10.5,'LBM/S',
     &5X,'VS=',F10.5,'FT/S',5X,'HS=',F10.5,'BTU/LBM',/,5X,
     &'RM1=',F10.5,'LBM/S'/5X,'T1=',F10.5,'F',5X,'P1=',
     &F10.5,'PSIA',5X,'V1=',F10.5,'FT/S',/,5X,'SV1=',F10.5,
     &'FT3/LBM',5X,'H1=',F10.5,'BTU/LBM',5X,'S1=',F10.5,'BTU/LBM-R')
      C1=(GAMMA*R*32.2*(T1+460.))**0.5
      XM1=V1/C1
      WRITE(6,10)C1,XM1
10    FORMAT(/5X,'C1=',F10.5,5X,'XM1=',F10.5)
      RETURN
      END
```

```
      SUBROUTINE TABLES(NR)
C
C     PURPOSE
C        TO PROVIDE CORRECT VALUES FOR CONSTANTS IN THE
C        THERMODYNAMIC PROPERTIES SUBPROGRAMS, CORRESPONDING
C        TO THE DESIRED REFRIGERANT (11, 12, 22, 114, OR 502,113)
C
C***  AUTHORS  G.T. KARTSOUNES AND R.A. ERTH, COMPUTER                ***
C              CALCULATION OF THE THERMODYNAMIC PROPERTIES
C              OF REFRIGERANTS 12, 22, AND 502,
C              ASHRAE TRANSACTIONS, VOL. 77, PT. 2, 1971
C
C***  REVISED BY C.K. RICE AND  S.K. FISCHER
C***  FURTHER REVISED BY W.L. JACKSON
C
C***  REFRIGERANTS 13,14,21,23,113,500,& C318
C        CAN ALSO BE USED BY SUPPLYING A NEW SET OF CONSTANTS AS
C        GIVEN BY: R. C. DOWNING, REFRIGERANT EQUATIONS,
C        ASHRAE TRANSACTIONS, VOL. 80, PART 2, 1974.
C
C     DESCRIPTION OF PARAMETERS
C      INPUT
C        NR    -  REFRIGERANT NUMBER (11, 12, 22, 114, OR 502,113)
C      OUTPUT
C        ALL OF THE CONSTANTS HELD IN COMMON BLOCKS
C
      REAL K,LE10,L10E,J
C
C     DESCRIPTION OF CONSTANTS
C
C        DENSITY CONSTANTS USED TO COMPUTE THE SPECIFIC VOLUME
C        OF THE LIQUID REFRIGERANT
C
      COMMON / DENSIT / AL, BL, CL, DL, EL, FL, GL
C
C        SPECIFIC HEAT AT CONSTANT VOLUME CONSTANTS
C        ACV,BCV,CCV,DCV,FCV;
C        ENTHALPY AND ENTROPY OF VAPOR CONSTANTS  X,  Y;
C        MISCELLANEOUS CONSTANTS  L10E,  J
C
      COMMON / OTHER / ACV, BCV, CCV, DCV, FCV, X, Y, L10E, J
C
C        VAPOR PRESSURE CONSTANTS
C
      COMMON / SAT / AVP, BVP, CVP, DVP, EVP, FVP
C
C        EQUATION OF STATE CONSTANTS
C
      COMMON / STATEQ / R, B1, A2, B2, C2, A3, B3, C3, A4, B4, C4,
     &                 A5, B5, C5, A6, B6, C6, K, ALPHA, CPR
C
      COMMON / SAVSAT / PSLAST,TSLAST
C
C        CRITICAL POINT TEMPERATURE  TC
C        INITIAL APPROXIMATION CONSTANTS  A,  B
C         (A & B ARE SET TO ZERO FOR REFRIGERANTS
C          OTHER THAN 12, 22, AND 502)
C        MISCELLANEOUS CONSTANTS  TFR,  LE10
```

```
      COMMON / SUPER / TC, A, B, TFR, LE10
C
      J = .185053
      L10E = .434294
      LE10 = 2.302585
      PSLAST=0.0
      TSLAST=-460.
      IF(NR.EQ. 11) GO TO  11
      IF(NR.EQ. 12) GO TO  12
      IF(NR.EQ. 22) GO TO  22
      IF(NR.EQ.114) GO TO 114
      IF(NR.EQ.502) GO TO 502
      IF(NR.EQ.113) GO TO 113
       GO TO 999
C
C CONSTANTS FOR REFRIGERANT 12
C
   12 CONTINUE
      AL = 34.84
      BL =  53.341187
      CL =  0.0
      DL =  18.69137
      EL =  0.0
      FL =  21.98396
      GL = -3.150994
      AVP = 39.883817
      BVP = -3436.63228
      CVP = -12.471522
      DVP = 4.730442E-03
      EVP = 0.0
      FVP = 0.0
      TC = 693.3
      PC = 596.9
      VC = .02870
      A = 120.0
      B = 312.0
      TFR = 459.7
      R = .088734
      B1 = 6.509389E-03
      A2 = -3.409727
      B2 = 1.594348E-03
      C2 = -56.762767
      A3 = 6.023945E-02
      B3 = -1.879618E-05
      C3 = 1.311399
      A4 = -5.487370E-04
      B4 = 0.0
      C4 = 0.0
      A5 = 0.0
      B5 = 3.468834E-09
      C5 = -2.543907E-05
      A6 = 0.0
      B6 = 0.0
      C6 = 0.0
      K = 5.475
      ALPHA = 0.0
      CPR = 0.0
      ACV = 8.0945E-03
```

```
      BCV = 3.32662E-04
      CCV = -2.413896E-07
      DCV = 6.72363E-11
      FCV = 0.0
      X = 39.556551
      Y = -1.653794E-02
      RETURN
C
C CONSTANTS FOR REFRIGERANT 22
C
   22 CONTINUE
      AL =  32.76
      BL =  54.63441
      CL =  36.74892
      DL = -22.29257
      EL =  20.47329
      FL =  0.0
      GL =  0.0
      AVP = 29.357545
      BVP = -3845.193152
      CVP = - 7.861031
      DVP = 2.190939E-03
      EVP = .445747
      FVP = 686.1
      TC = 664.5
      PC = 721.906
      VC = .030525
      A = 0.0
      B = 0.0
      TFR = 459.69
      R = .124098
      B1 = .002
      A2 = -4.353547
      B2 = 2.407252E-03
      C2 = -44.066868
      A3 = -.017464
      B3 = 7.62789E-05
      C3 = 1.483763
      A4 = 2.310142E-03
      B4 = -3.605723E-06
      C4 = 0.0
      A5 = -3.724044E-05
      B5 = 5.355465E-08
      C5 = -1.845051E-04
      A6 = 1.363387E08
      B6 = -1.672612E05
      C6 = 0.0
      K = 4.2
      ALPHA = 548.2
      CPR = 0.0
      ACV = 2.812836E-02
      BCV = 2.255408E-04
      CCV = -6.509607E-08
      DCV = 0.0
      FCV = 257.341
      X = 62.4009
      Y = -4.53335E-02
      RETURN
```

```
C
C CONSTANTS FOR REFRIGERANT 11
C
  11  CONTINUE
      AL = 34.57
      BL = 57.63811
      CL = 43.6322
      DL = -42.82356
      EL = 36.70663
      FL = 0.0
      GL = 0.0
      AVP = 42.14702865
      BVP = -4344.343807
      CVP = -12.84596753
      DVP = 4.0083725E-03
      EVP = .0313605356
      FVP = 862.07
      TC = 848.07
      PC = 639.50
      VC = .028927
      A = 0.0
      B = 0.0
      TFR = 459.67
      R = .078117
      B1 = .00190
      A2 = -3.126759
      B2 = 1.318523E-03
      C2 = -35.76999
      A3 = -.025341
      B3 = 4.875121E-05
      C3 = 1.220367
      A4 = 1.687277E-03
      B4 = -1.805062E-06
      C4 = 0.0
      A5 = -2.358930E-05
      B5 = 2.448303E-08
      C5 = -1.478379E-04
      A6 = 1.057504E08
      B6 = -9.472103E04
      C6 = 0.0
      K = 4.50
      ALPHA = 580.0
      CPR = 0.0
      ACV = 2.3815E-02
      BCV = 2.798823E-04
      CCV = -2.123734E-07
      DCV = 5.999018E-11
      FCV = -336.80703
      X = 50.5418
      Y = -9.18395E-02
      RETURN
C CONSTANTS FOR REFRIGERANT 502
  502 AL =  35.00
      BL =  53.48437
      CL =  63.86417
      DL = -70.08066
      EL =  48.47901
      FL = 0.0
```

```
          GL = 0.0
          AVP = 10.644955
          BVP = -3671.153813
          CVP = -  .369835
          DVP = -1.746352E-03
          EVP = .816114
          FVP = 654.0
          TC = 639.56
          PC = 591.00
          VC = .028571
          A = 0.0
          B = 0.0
          TFR = 459.67
          R = .096125
          B1 = .00167
          A2 = -3.261334
          B2 = 2.057629E-03
          C2 = -24.24879
          A3 = 3.486675E-02
          B3 = -.867913E-05
          C3 = .332748
          A4 = -8.576568E-04
          B4 = 7.024055E-07
          C4 = 2.241237E-02
          A5 = 8.836897E-06
          B5 = -7.916809E-09
          C5 = -3.716723E-04
          A6 = -3.825373E07
          B6 = 5.581609E04
          C6 = 1.537838E09
          K = 4.2
          ALPHA = 609.0
          CPR = 7.E-07
          ACV = 2.0419E-02
          BCV = 2.996802E-04
          CCV = -1.409043E-07
          DCV = 2.210861E-11
          FCV =0.0
          X = 35.308
          Y = -.07444
          RETURN
C CONSTANTS FOR REFRIGERANT 114
114       AL = 36.32
          BL = 61.146414
          CL = 0.0
          DL = 16.418015
          EL = 0.0
          FL = 17.476838
          GL = 1.119828
          AVP =  27.071306
          BVP = -5113.7021
          CVP = -6.3086761
          DVP =  6.913003E-04
          EVP =  0.78142111
          FVP =  768.35
          TC = 753.95
          PC = 473.0
          VC = 0.0275
          A = 0.0
```

```
      B = 0.0
      TFR = 459.69
      R  =  0.062780807
      B1 =  0.005914907
      A2 = -2.3856704
      B2 =  1.0801207E-03
      C2 = -6.5643648
      A3 =  0.034055687
      B3 = -5.3336494E-06
      C3 =  0.16366057
      A4 = -3.857481E-04
      B4 =  0.0
      C4 =  0.0
      A5 =  1.6017659E-06
      B5 =  6.2632341E-10
      C5 = -1.0165314E-05
      A6 =  0.0
      B6 =  0.0
      C6 =  0.0
      K  =  3.0
      ALPHA = 0.0
      CPR = 0.0
      ACV =  0.0175
      BCV =  3.49E-04
      CCV = -1.67E-07
      DCV =  0.0
      FCV =  0.0
      X =  25.3396621
      Y = -0.11513718
    RETURN
C CONSTANTS FOR REFRIGERANT 113
  113 AL= 122.872
    BL= -0.0128
    CL= 6.36E-05
    DL= 0.0
    EL= 0.0
    FL= 0.0
    GL= 0.0
    AVP= 33.0655
    BVP= -4330.98
     CVP= -9.2635
    DVP= 2.0539E-03
    EVP= 0.0
    FVP= 0.0
    TC = 877.0
    PC = 498.9
    VC = 0.0278
    A=0.0
    B=0.0
    TFR= 459.6
    R= 0.05728
    B1=0.0
    A2=-4.035
    B2=2.618E-03
    C2=0.0
    A3=-0.0214
    B3=5.00E-05
    C3=0.0
    A4=0.0
```

```
      B4=0.0
      C4=0.0
      A5=0.0
      B5=0.0
      C5=0.0
      A6=0.0
      B6=0.0
      C6=0.0
      K =0.0
      ALPHA=0.0
      CPR=0.0
      ACV=0.07963
      BCV=1.159E-04
      CCV=0.0
      DCV=0.0
      FCV=0.0
      X=25.198
      Y=-0.40552
        RETURN
C     PRINT ERROR MESSAGE IF
C     'NR' DOES NOT EQUAL 12, 22, 502, OR 114,113,11
  999 WRITE(6,1000) NR
 1000 FORMAT('OTABLES: ***** SUBROUTINE NOT VALID FOR REFRIGERANT ',
     &               I3,' *****',/,11X,'*** USER MUST SUPPLY NEW',
     &               'CONSTANTS',/)
      RETURN
      END
```

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