corresponding to points 4 and 5 on the P-h diagram of figure3.

The method employed here for calculating the cross-section area of the mixing channel is characteristic of similar techniques of two-phase ejector calculation given by Fisenko [4]. From the known values of the velocity of the mixing stream and across mixing section area(cross-section area of the cylindrical channel at the mixing chamber outlet), it is possible to calculate the pressures of the working fluid flow at the mixing chamber outlet, P_{mix}, and at the ejector outlet after the diffuser, P_d.

In this event the following equations were applied:

$$P_b\, a_m + \left(\frac{1}{(1+w)}\right) *u_{mb} + \left(\frac{1}{(1+w)}\right) *u_{sb} = P_m\, a_m + u_{mix} \quad 5.6$$

$$P_d = P_{mix} + \rho\, (u_{mix}^2 - u_d^2) / 2 \quad 5.7$$

The former being the momentum conservation equation, whereas the latter is the energy conservation equation in the form of Bernoulli equation. In these equations u_{mb}, u_{sb} – velocities of the liquid and vapor flows (motive and suction) at the mixing section inlet, u_{mix}, u_d -mixture flow velocity at the diffuser inlet and outlet.

It needs to emphasize that in our case, the mixture velocity in the mixing chamber has to be somewhat higher than local sonic speed of this two-phase flow because in this case the efficiency of the vapor-liquid ejector increases [4]. In its turn, according to the known data [1], the speed of sound propagation α in a two-phase medium can be as low as only 20-50 m/s and for its estimate one can apply the equation:

$$\alpha^2 = kP / \rho_{mix} \text{ or } a^2 = P // \rho_f\, \beta(1- \beta) \quad 5.8$$

where k is isentropic coefficient, P, ρ_{mix} is pressure and density of the two-phase flow, ρ_f is density of the liquid phase and β is the volumetric content of vapor in the mixture.

5.1 Governing Equations

A. Compressor

The compressor is assumed to be non-isentropic. Process 1-2s is an isentropic compression process, while process 1-2 is the actual compression process. The actual enthalpy of state 2 is expressed by:

$h_3 = h_2 + (h_{3s} – h_2)/\eta_c$ 5.9

where hc is the isentropic efficiency of the compression process.

The enthalpy and entropy of the refrigerant at state 1 are determined by the temperature and pressure at the compressor inlet as:

h_2 , s_2= $f(T_2,P_2)$ 5.10

The refrigerant enthalpy at state 2s for the isentropic process is:

h_{3s}= $f(s_{3s},p_3)$ 5.11

Where $s_{3s}=s_2$

The quantity of energy needed for the compression of the vapor flow m_v by the compressor with the performance η_c is determined by the expression

$\ell_c = m_v(h_3 – h_2) / \eta_c$ 5.12

B. Condenser/Generator

In the regenerative system shown in Fig. 2, low temperature fluid, which generates vapor by absorbing heat from the high temperature compressor discharge, becomes the working fluid to drive the ejector. Note that the condensing temperature in the basic refrigeration cycle is lower than the evaporating temperature of the ejector cycle. Therefore, only part of the sensible heat can be used to vaporizing the refrigerant of the ejector cycle in the generator. The total energy balance in the vapor generator is:

$m_3(h_3 – h_4) = m_{10}(h_{10} – h_9)$ 5.13

For the heat exchanger design, there is a minimum temperature difference at the generator's two sides:

$T_3 > T_{10} + \Delta T_g,\ T_4 = T_9 + \Delta T_g$ 5.14

The fluid state at the generator exit is:

$T_4, s_4= f(P_4, h_4)$ 5.15

$T_{10}, h_{10}\ s_{10}= f(P_3)$ 5.16

$Q_g= m(h_3 – h_4)$ 5.17

C. Pump:

In the regenerative system the pump is also used to raise the pressure of mixing fluid. The total mass balance at the pump is: $m_4 + m_5 = m_{10} + m_{11}$ 5.18

h_7, s_7 =$f(T_7,P_7)$ 5.19

The quantity of energy ℓ_p, consumed by a pump in compressing a working fluid is calculated with the formula

$\ell_p= m_f\, \Delta P_{7-6}/(\rho_{mix}*\eta) = m_f(h_7 – h_6) /(\rho_{mix}*\eta)$ 5.20

where η is efficiency (coefficient of efficiency) of the pump.

D. Evaporator:

The function of the evaporator in the basic refrigeration system differs from that in the hybrid system. For the evaporator , assume that the refrigerant at the exit (state 1) is super heated . The governing equations for the evaporator are then

$P_1=f_{sat}(T_{evp.})$ 5.21 $T_2= T_{evp.} + \Delta T_e$ 5.22

$h_2,s_2= f(T_2, P_2)$ 5.23

Refrigerating capacity of the system Qo

$Q_o = m_v(h_2 – h_1)$ 5.24

E. Ejector:

The ejector works as a compression device where the high pressure primary flow (state 13) entrains the low-pressure secondary flow (state 7) into the ejector. Previous studies have shown that the ejector performance is influenced by both the ejector geometry and the operating conditions. The ejector performance is usually evaluated based on the combined mass flow rates of the two flows. The mass flow rate of the primary flow of the ejector is determined by the ejector's structure and the thermodynamic properties of the primary flow. Assuming isentropic flow, the mass flow rate of the primary flow through the nozzle,m_{11}, when choked can be expressed by (Huang et al., 1999; Zhu et al., 2007).

$$m_{11}= A_t[\, \Psi_{ej}\gamma P_{11}\rho_{11}\,]^{1/2}\, (\rho/(1+\gamma))^{(\gamma+1)/2(\gamma-1)} \quad 5.25$$

where Ψej represents a coefficient related to the isentropic efficiency of the compressible flow in the nozzle and P_{11} and T_{11} are the pressure and temperature of the primary flow, respectively, at the ejector inlet.