THEORETICAL AND EXPERIMENTAL INVESTIGATION OF A TWO-PHASE/TWO-COMPONENT EJECTOR FOR COLD PRODUCTION

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ABSTRACT

In order to make use of low-temperature heat (solar heat, residual heat), the possibility was investigated of using a two-phase/two-component ejector which operates with ammonia/water as the working mixture. Both the primary, which is under high pressure, as well as the suction flow comprise two-phase flows with specific vapour contents. The purpose of this is to align the velocities in the mixing section in order to reduce the losses. The objective is to improve the efficiency of the ejector. The calculation of the two-phase/two-component ejector will be discussed in this paper. On the basis of the theoretical calculations, an ejector refrigeration plant with a cold capacity of 20 kW was constructed. The results will be reported in detail.

KEYWORDS

Ejector, two-phase flow, cold production, supersonic flow, ammonia

1. INTRODUCTION

The use of low-temperature heat (solar energy, residual heat) possesses great potential for the production of cold which can be used by means of absorption processes. In view of the reduction in the apparatus needed, ejector processes are an economic alternative although the COP - factor is smaller in this case.

The objective of the process presented here consisted in the development of a solar-driven refrigeration plant for air-conditioning and refrigerating purposes. The main component is a two-phase/two-component ejector which operates with ammonia/water as the working mixture. Both the primary, which is under high pressure, as well as the suction flow comprise two-phase flows with specific vapour contents. The reason for this lies in the fact that after the expansion in the primary or suction nozzle, smaller differences in velocity at the entry to the mixing section can be achieved. In thermodynamic terms, this means that the change in enthalpy on the primary or suction side is almost the same. The impact losses are reduced as a result so that consequently efficiency is improved. Two-phase flows also prevail in the mixing section and in the diffuser. It can even occur that after mixing in the mixing section there is a supersonic flow. In this case a Laval nozzle is used in place of a diffuser to achieve an efficient conversion.

Following extensive pre-calculations, a test plant with about 20 kW cold capacity was constructed on which comprehensive measurements were carried out. The comparison of the theoretical calculations with the measurements recorded is in work.

2. PROCESS DESCRIPTION

The design of the ejector refrigeration plant can be seen in Figure 1. The main components of the plant are: Evaporator 1, Evaporator 2, Two-phase/two-component ejector, Condenser, Storage container, Throttle valve and Pump. Ammonia/water is the working mixture used. The thermodynamic states are indicated with numbers.

In Evaporator 1, the cold is absorbed by the two-phase flow. As a result, the vapour content increases. This mix then flows into the suction side of the ejector and is brought, with the aid of the primary flow, which is under high pressure and high temperature, to a higher pressure (condenser pressure). At the exit of the ejector there is also a two-phase flow which is completely condensed in the condenser by heat emission. Behind the condenser, the liquid stream is separated into two flows. One part undergoes an increase in pressure by means of a high-pressure pump and then enters the Evaporator 2. Here the low-temperature heat (at about 90°C) is fed in so that again a two-phase stream is formed which is supplied to the ejector as primary stream.

The other part of the liquid stream behind the condenser is expanded to a lower pressure and

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supplied to Evaporator 1 to absorb the cold. After expansion, a two-phase flow is also formed. Thus there are two-phase flows both at the entry and the exit of the evaporator as well as at the entry and the exit of the ejector.

The heat required by Evaporator 2 is provided by hot water whereby the conditions of simple flat solar collectors (maximum heating temperature 90°C) are simulated. The heat (cold performance) needed to supply Evaporator 1 is provided by cold water. The cooling of the condenser is effected by means of cooling water. The setting of the temperature and the pressure at the entries to the ejector is carried out by means of control valves.
The investigation of the processes in the ejector was the main interest of this work. A special focus lay in the calculation of the sound velocity of the two-phase/two-component working mixture which was examined in comprehensive tests.

3. MATHEMATICAL MODEL FOR THE DESCRIPTION OF THE PROCESSES IN THE EJECTOR

For the dimensioning, the ejector was divided into individual sections such as primary nozzle, suction nozzle, mixing section, diffuser, Laval nozzle and calculated with the aid of prescribed empirical efficiencies and empirical flow equations. Knowledge of the two-phase sound velocity is highly critical here since it is greatly relevant whether the flow occurs at supersonic or subsonic velocity. In order to calculate the necessary mass and energy balances and the states in the ejector, a calculation programme was developed which establishes the parameters required with the relevant material data. In order to determine the material data in the two-phase state, the AWMix [1] programme routine was used. The calculation procedure is described in the following section. Figure 2 shows the design of the ejector with the relevant state indicators. The following data were assumed as known:

- composition of the mixture : z
- evaporation temperature : T_4
- evaporation pressure : p_4
- heating temperature : T_1
- heating pressure : p_1
- condensation temperature : T_9
- condensation pressure : p_9
- Cold capacity : $\dot{Q}_4$
- efficiencies of the nozzle parts:
  - convergent part of the driving nozzle : $\eta_{T,konv}$
  - divergent part of the driving nozzle : $\eta_{T,div}$
  - suction nozzle : η_S
  - mixing section : η_M
  - convergent part of the Laval nozzle : $\eta_{L,konv}$
  - divergent part of the Laval nozzle : $\eta_{L,div}$

First of all the suction mass flow $\dot{m}_4$ is calculated with the enthalpies of the states of the evaporator and the cold performance.

$$\dot{m}_4 = \frac{\dot{Q}_4}{h_4 - h_9} \tag{1}$$

Then the reversible enthalpy at the exit of the ejector is determined.

$$h_{8,rev} = \left(\dot{m}_{1,rev} \cdot h_1 + \dot{m}_4 \cdot h_4\right) / \left(\dot{m}_{1,rev} + \dot{m}_4\right) \tag{2}$$

With this reversible enthalpy $h_{8,rev}$, the exit pressure p_8 of the ejector (corresponding to the condensation pressure p_9) and the composition z, further material data of the reversible state 8 can be established. With the aid of the reversibility condition

$$\dot{m}_{1,rev} \cdot s_1 + \dot{m}_4 \cdot s_4 = \left(\dot{m}_{1,rev} + \dot{m}_4\right) \cdot s_{8,rev} \tag{3}$$

the reversible primary mass flow $\dot{m}_{1,rev}$ is examined. This factor is continuously varied until the reversibility condition is fulfilled. Then the actual primary mass flow can be calculated with the overall efficiency η_{ges} of the ejector, which is formed from individual empirical efficiencies of nozzle sections, and with the aid of the energy balance:

$$\dot{m}_1 = \frac{\left[\left(\dot{m}_{8,rev} \cdot h_{8,rev} / \eta_{ges}\right) - \dot{m}_4 \cdot h_4\right]}{h_1} \tag{4}$$

with

$$\eta_{ges} = \eta_{T,konv} \cdot \eta_{T,div} \cdot \eta_S \cdot \eta_M \cdot \eta_{L,konv} \cdot \eta_{L,div} \tag{5}$$

$$\dot{m}_{8,rev} = \dot{m}_{1,rev} + \dot{m}_4 \tag{6}$$

On the basis of the parameters thus obtained, the calculation of the actual enthalpy at the exit of the ejector can now take place:

$$h_{8,real} = \frac{\left(\dot{m}_1 \cdot h_1 + \dot{m}_4 \cdot h_4\right)}{\dot{m}_8} \tag{7}$$

with $\dot{m}_8 = \dot{m}_1 + \dot{m}_4$ (8)

Once the mass flows of the ejector have been established, the material data for the individual nozzle sections are determined step by step using the material data routines. With this material data, the velocities at the individual states can be calculated which are necessary for the dimensioning of the ejector.

3.1 Calculation of the suction nozzle

High differences in velocity cause high losses upon impact of the primary and suction flows in the mixing section. In order to reduce these impact losses, the flow velocity of the suction flow must be increased prior to impact [2]. The pressure p_6 is assumed for the exit of the suction nozzle (state 6). The pressure is continuously varied until at this state the velocity of sound, i.e. Ma - number =1, is reached.

The procedure is as follows. First of all the reversible enthalpy is determined with the isentropic entropy $s_4 = s_{6,isen}$, the pressure p_6 and the composition z. Then the actual enthalpy at state 6 with the prescribed efficiency of the suction nozzle is established:

$$h_{6,tat} = h_4 - \eta_S \cdot \left(h_4 - h_{6,isen}\right) \tag{9}$$

The actual material data of state 6 can be determined with the actual enthalpy $h_{6,real}$, the pressure p_6 and the composition z. Calculation of the velocity at state 6 is carried out with

$$w_6 = \sqrt{\left(2 \cdot \left(h_4 - h_{6,real}\right) + w_4^2\right)} \tag{10}$$

The aim of the calculation is that sound velocity prevails at the exit of the suction nozzle. Sound velocity is calculated for a homogenous phase distribution without phase change according to an equation described by *Deichsel* and *Winter* for two-phase sound velocity [3]. In this model envisages that the gas phase is regarded as fluid with elastic surrounding formed through the liquid phase. The sound velocity of the two-phase flow therefore results from the overlapping of the sound velocities of the individual phases:

$$c_6 = \frac{1}{\left(1-\varepsilon_6\right) \cdot \sqrt{\left(\frac{\left(1-\varepsilon_6\right)}{c_{F,6}^2} + \frac{\rho_{F,6}}{\rho_{D,6}} \cdot \frac{\varepsilon_6}{c_{D,6}^2}\right)} + \varepsilon_6 \cdot \sqrt{\left(\frac{\varepsilon_6}{c_{D,6}^2} + \frac{\rho_{D,6}}{\rho_{F,6}} \cdot \frac{\left(1-\varepsilon_6\right)}{c_{F,6}^2}\right)}} \tag{11}$$

The calculation of the individual sound velocities for liquid and gas phase is made with the programme which was developed, whereby temperature, density and the composition of the phases in question are prescribed. The vapour content was converted into volume fraction according to [4]:

$$\varepsilon_6 = \frac{\varphi_{6,real} \cdot \rho_{D,6,real}}{\rho_{6,real}} \quad . \tag{12}$$

The Mach number is used to examine whether there is sound flow at the exit of the suction nozzle.

$$Ma_6 = \frac{w_6}{c_6} \quad . \tag{13}$$

The iteration of the pressure p_6 is continued until the condition Ma = 1 is fulfilled.

3.2 Calculation of driving nozzle

The procedure for calculating the narrowest section of the primary nozzle (state 2) and the exit of the primary nozzle (state 3) is carried out according to the process for calculating the suction nozzle. In the primary nozzle, the working medium is expanded from high temperature T_1 and high heating pressure p_1 to a low evaporation pressure p_3. Due to the high pressure drop from p_1 to p_3, the velocity of the primary nozzle increases dramatically. It has been proven that an ejector operates the most economically when an even pressure prevails in the nozzle mouth [5]. The pressure at the exit of the primary nozzle is therefore set at the same value as the exit pressure of the suction nozzle p_3=p_6. At this pressure, the flow reaches a supersonic velocity at state 3. As a result, a powerful suction effect is produced which leads to an under-pressure in the suction nozzle. The primary nozzle tears the refrigerant out of the suction nozzle and mixes with it in the mixing section. In the primary nozzle, pressure energy of the primary flow is therefore transformed into kinetic energy.

Calculation of the primary nozzle is carried out in two stages, the first is from the entry to the narrowest section and then from the narrowest point to the exit of the primary nozzle. To calculate the narrowest section, the reversible enthalpy is established assuming a certain pressure and then the actual state is calculated. The sound velocity is then determined and compared with the requirement Ma = 1. If the condition is not fulfilled, the pressure at the narrowest point is varied continuously until Ma = 1 is reached. On the basis of these values – assuming that the pressure at the exit of the primary nozzle corresponds to that at the exit of the suction nozzle – the state at the exit of the primary nozzle can be established. The algorithm does not need to be presented here.

3.3 Calculation of mixing tube

For the calculation of the mixing section, constant pressure over the length of the mixing section was assumed ($p_6 = p_3 = p_7$). The effects of this assumption are described in [6]. With the energy equation, first of all the reversible entropy is calculated:

$$s_{7,rev} = \frac{\left(s_{3,real} \cdot \dot{m}_1 + s_{6,real} \cdot \dot{m}_4\right)}{\dot{m}_8} \tag{14}$$

With the reversible entropy $s_{7,rev}$, the pressure p_7 and the composition z, the material data for the reversible state 7 can be established. With the reversible enthalpy $h_{7,rev}$, the reversible velocity $w_{7,rev}$ at the exit of the mixing section is calculated via the energy equation:

$$w_{7,rew}^2 = 2 \cdot \left(\frac{\dot{m}_1}{\dot{m}_8} \cdot \left(h_{3,real} + \frac{w_3^2}{2} \right) + \frac{\dot{m}_4}{\dot{m}_8} \cdot \left(h_{6,real} + \frac{w_6^2}{2} \right) - h_{7,rew} \right) \quad (15)$$

Via the definition of the mixing section efficiency $\eta_{M,}$ the actual velocity w_7 at state 7 can be calculated:

$$w_7^{\,2} = \eta_M \cdot w_{7,rev}^2 \quad . \quad (16)$$

Calculation of the actual enthalpy at State 7 is carried out with the energy balance:

$$h_{7,real} = \left(\frac{\dot{m}_1}{\dot{m}_8} \cdot \left(h_{3,real} + \frac{w_3^2}{2} \right) + \frac{\dot{m}_4}{\dot{m}_8} \cdot \left(h_{6,real} + \frac{w_6^2}{2} \right) - \frac{w_7^2}{2} \right) \quad (17)$$

With the actual enthalpy at the exit of the mixing section $h_{7,real}$, the pressure p_7 and the composition z, the thermodynamic state at the end of the mixing section is determined. The sound velocity in this state is calculated in exactly the same way as described in the previous chapters.

3.4 Laval nozzle

The efficiency of a diffuser is all the lower poorer, the greater the differences in velocity of the flow upon entry into the diffuser [7, 8]. Therefore it is important to balance out the differences in velocity between the primary flow and the suction flow so that the subsequent rise in pressure in the diffuser can be executed with a high rate of efficiency. Since the calculations showed that supersonic velocity prevails at the end of the mixing section, a Laval nozzle was used after the mixing section in place of the diffuser. It serves to slow down the flow to sound velocity and then, like a diffuser, to bring the mix with as little losses as possible to a higher pressure, the condensation pressure. The stages for calculating the parameters of the Laval nozzle correspond to those for calculating the primary nozzle.

4. CALCULATION RESULTS

Parameter variations were carried out for the design and calculation of the ejector. With these calculations it was important to design the two-phase/two-component ejector in such a manner that high COP - factor at acceptable mass flow ratios could be obtained. The COP - factor, as a key feature of the efficiency of refrigeration plants, was considered for the evaluation. In figures 3 to 5, the calculated COP - factors dependent on the suction vapour contents at various primary vapour contents and different compositions are presented.

An evaporation temperature of ϑ_1 = 4 °C, a heating temperature of ϑ_2 = 90 °C and a condensation temperature of ϑ_4 = 40 °C were set as framework parameters. These values correspond to the primary objective of producing cold water with a temperature of 6°C with solar heat of 90°C at common ambient temperatures in Europe of about 30 – 35°C. Heating temperatures around 90°C can be achieved with the common solar collectors available on the market.

The results show that, depending on the parameters chosen such as the temperature of the cold and the temperature of the condensation, an optimum COP - factor ensues at specific vapour contents on the primary and on the suction side. The scale of the optimal vapour content on the suction side is between 0.7 - 0.8 mass fraction. Furthermore the COP - factor rises as the ammonia concentration increases.

On the basis of the calculation results, a test plant was built in which measuring sensors are foreseen in the ejector to measure the course of the temperature and the pressure. The measuring results will be compared to the calculation results, in order to validate the mathematical model later.
From the diagrams it can be seen that the primary vapour content regarding the COP - factor is insignificant. To keep the mass flow ration (primary side / suction side) in an acceptable range, then vapour contents on the primary side should not fall below 0.5.
Individual efficiencies for nozzles and nozzle sections were sufficiently investigated in the literature. The common values, which have been empirically confirmed, were included in these calculations. Only for the efficiency of the mixing section are there no precise indications in literature. However, from known or measured overall efficiencies (on the scale of 0.3 to 0.6), it is possible to arrive at a mixing section efficiency of the ejector designed here. On the basis of this and under consideration of the advantages of the two-phase flow, a mixing section efficiency of η_M = 0.7 was selected from which results an overall efficiency of the ejector of 0.55.

5. CONCLUSIONS

The objective of this work lay in the improvement of the efficiency of an ejector refrigeration plant by using the two-phase/two-component working mixture of ammonia/water. The theoretical calculations showed that pure ammonia is better than the ammonia/water mixture with regard to increasing the efficiency. The maximum scale of the COP - factor which can be achieved is 0.3 – 0.35. However, through the two-phase state of the working medium, a significant improvement of the COP - factor can be achieved, compared to one-phase ejectors. The suction vapour content has a major influence here. At suction vapour contents of 0.7 – 0.8, an optimum COP - factor is achieved. Furthermore it became evident that the primary vapour content has almost no influence on the COP - factor.

6. LITERATUR

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Nomenclature

A	Surface (m^2)
c	Sonic speed (m/s)
d	Diameter (m)
h	Specific Enthalpy (kJ/kg)
L	Length (m)
Ma	Mach number
$\dot{m}$	Mass flow (kg/s)
p	Pressure (bar)
$\dot{Q}$	Cold Capacity (kW)
s	Specific Entropy (kJ/kg K)
T	Temperature (K)
w	Velocity (m/s)
z	Content (%)

Greek letters

α	Angle (°)
ε	Volume fraction
η	Efficiency
ϑ	Temperature (°C)
φ	Vapour content
μ	Mass flow ratio
ρ	Density (kg/m^3)

Indices

D	Vapour
div	Divergent
F	Liquid
konv	Convergent
L	Laval nozzle
M	Mixing room
rev	Reversible
S	Secondary nozzle
real	real
T	Primary nozzle
0	State after pump
1	Entry of primary nozzle
2	Narrowest diameter of driving nozzle
3	Exit of the primary nozzle
4	Entrance the suction nozzle
6	Exit of the suction nozzle
7	Exit of mixing section
7‘	Throat diameter of Laval nozzle
8	Exit of Laval nozzle
9	Condensor
10	Exit throttle valve

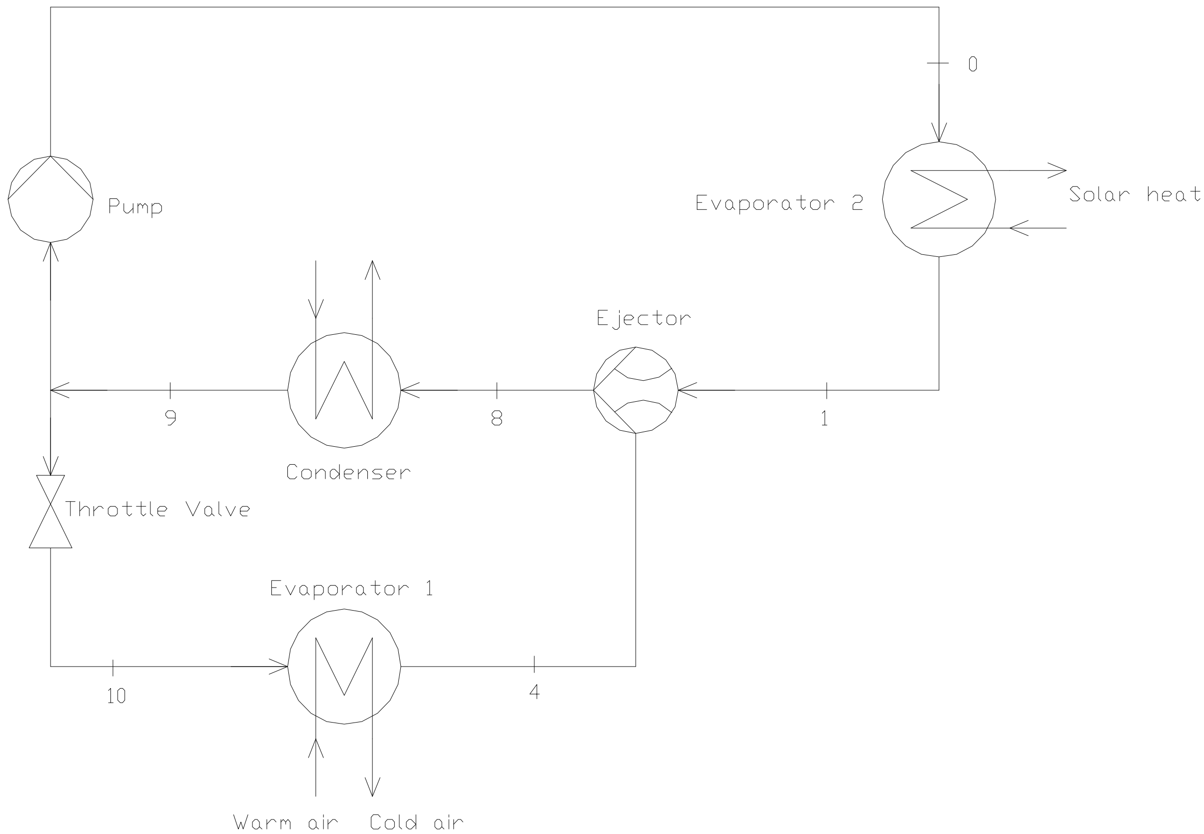

Fig. 1: Simplified process scheme of the process

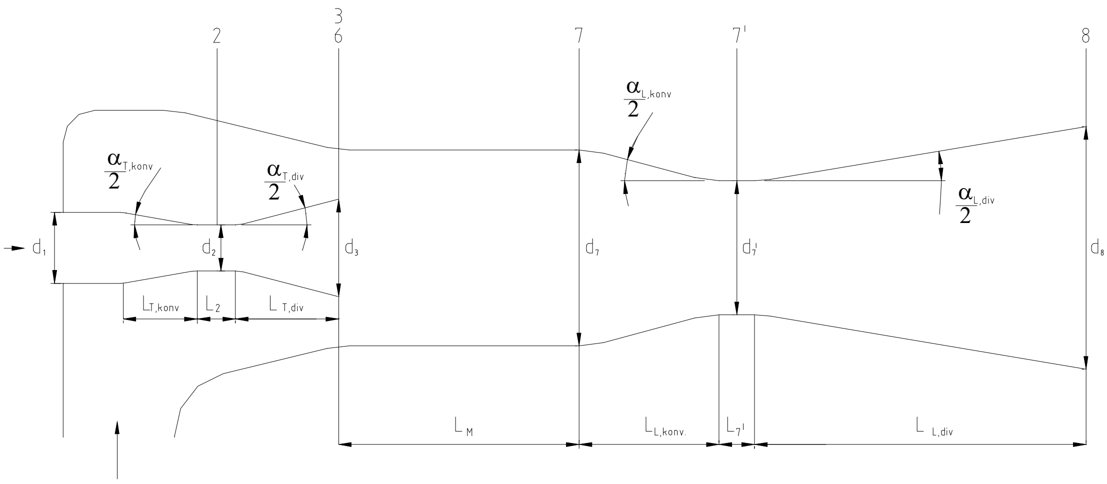

Fig. 2: Structure of the ejector with status designations

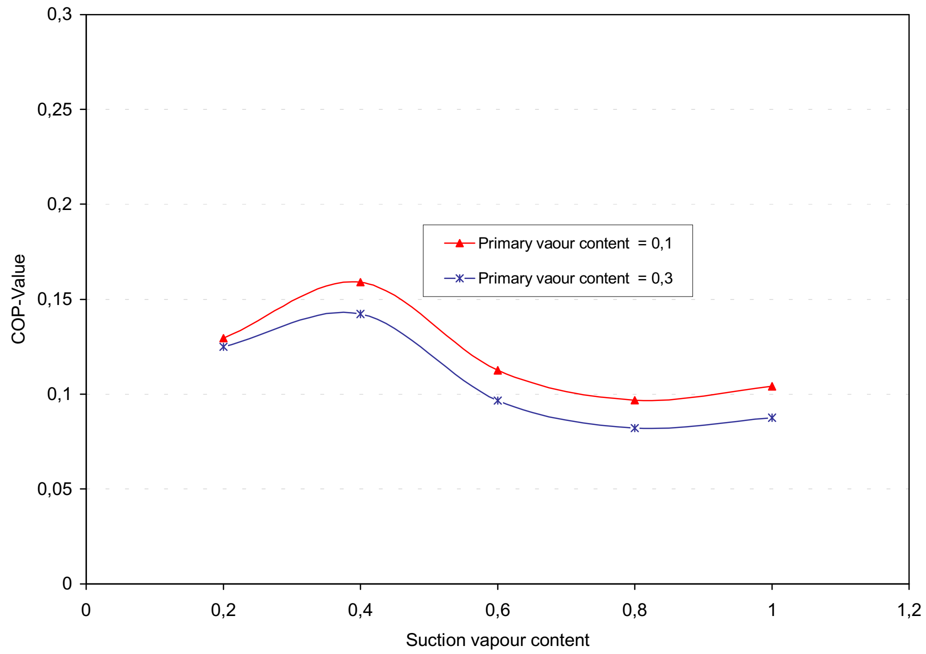

Figure 3: **Cold ratio (COP-factor) in dependence on the suction vapour conten t at various primary vapour contents for a composition of z= 70% and a mixing section efficiency of $\eta_M = 0,7$**

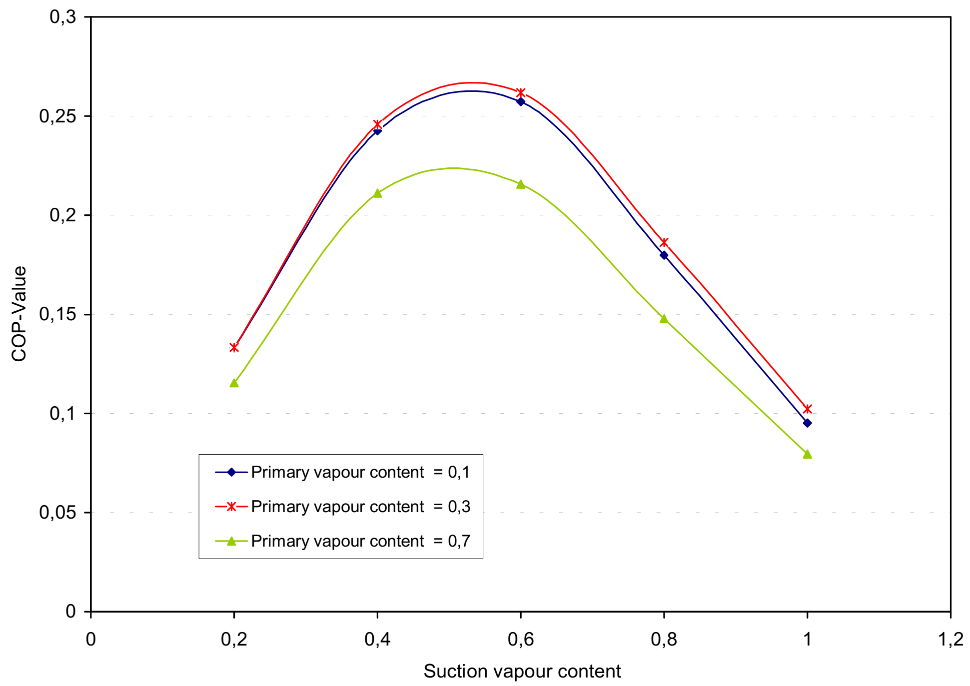

Figure 4: **Cold ratio (COP - factor) in dependence on the suction vapour content at various primary vapour contents for a composition of z = 90 % and a mixing section efficiency of $\eta_M = 0.7$.**

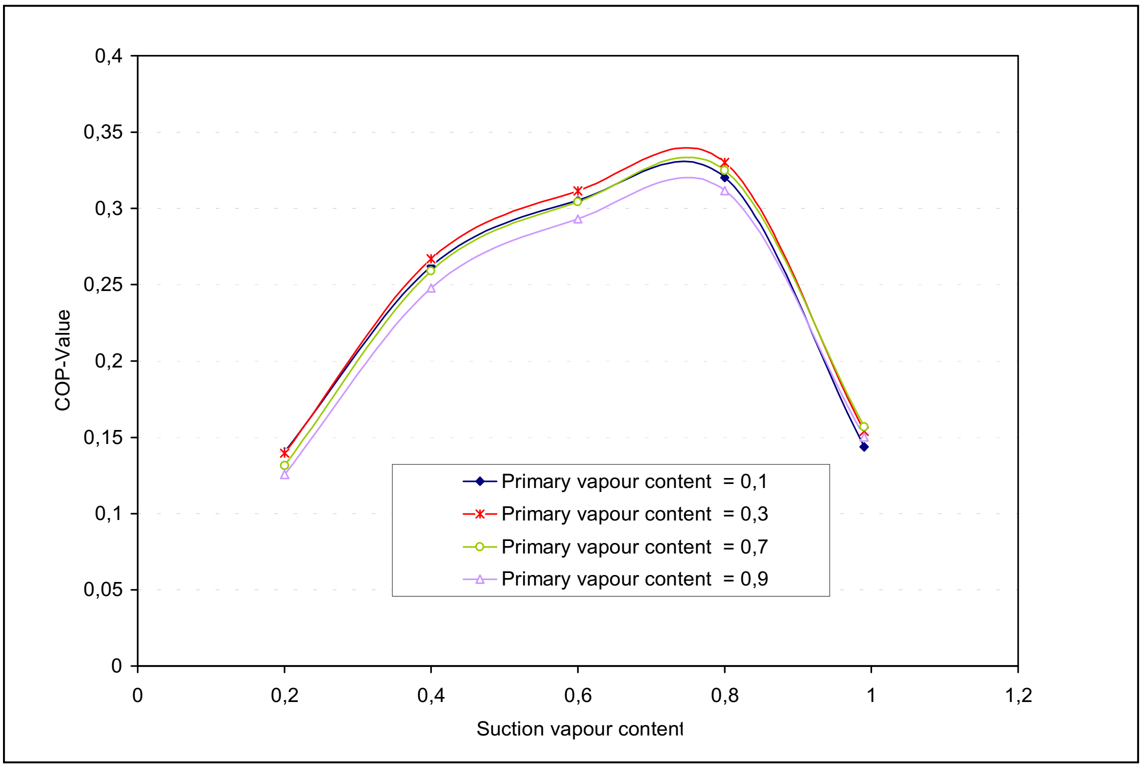

Figure 5: **Cold ratio (COP - factor) in dependence on the suction vapour content at various primary vapour contents for a composition of z = 99 % and a mixing section efficiency of $\eta_M = 0.7$.**