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INVESTIGATION OF GEOMETRY, TOTAL CONDITION AND WAVES EFFECT ON TWO PHASE LIQUID-VAPOR FLOW USING EQUILIBRIUM THERMODYNAMICS

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ABSTRACT

For steam under expansion processes, a second phase can be formed. One of the main applications of steam is in steam turbines; in which fluid expands when go through turbine stages as its pressure decreases. Because of this decreasing, the liquid phase in the last stages of turbine can be formed. In this paper, a numerical method for the solution of two-dimensional two-phase flow by means of equilibrium thermodynamic is presented. The unsteady N-S equations, governing the overall behavior of compressible, inviscid flow, are combined with equations of state that are numerically solved by the Roe's FDS time marching scheme which has been recently modified to allow for two-phase effects. After describing the numerical method, the location of condensation onset and the effects of flow condensation on other thermodynamic and flow parameters have been investigated in three separate sections. In the first section, the effects of geometry on both compression and expansion processes have been modeled and changes of two phase flow for different expansion rates have been indicated. In the second section, effects of inlet total condition on two-phase flow are studied and finally two-phase flow passing through expansion and compression waves are analyzed. In order to validate the obtained results, some properties of flow in centerline of two dimensional convergent-divergent nozzle are compared with experimental results and also a recently developed analytical solution.

KEY WORDS: two phase flow, geometry, total condition, waves; equilibrium thermodynamics.

INTRODUCTION

Two-phase or multi-phase flows have been utilized by different industries. The flow in rocket motors, flow in oil transmission pipes and etc. are the forms of multi-phases flows. Naturally, most phenomena such as storm and blood circulation have been known as multi-phases flows. Gas-liquid flow is a type of multi-phases flow that could be used in some processes such as boiling process, cavitations, bubble-involved flows and steam-based flows along with condensation. Studying about condensation steam two-phase flow more than scientific attraction has a significant importance in mechanical and chemical engineering, some applications of this flow include: moisture formation in combustion, condensation of fuel cells, aerodynamic tests in wind tunnel, particularly related to two-phase flows in steam turbines and its simple model i.e. convergent-divergent nozzle.

Condensation in steam turbine flow has been tested all the time because it is an important factor in power production industry. For example, this phenomenon could be seen under different pressures of steam turbines in nuclear power plants and at the end stages of steam turbines in fossil power plants. The liquid phase in steam flow affects the pressure and temperature profiles and may reduce the efficiency of the turbine. On the other hand the existence of a drop as well as the kind of material used, in the environment causes corrosion. The above mentioned can show the importance of investigating the two-phase water-steam flow. In this paper, only the thermodynamic aspects of flow has been studied and now the corrosion phenomenon through drop-phase has been disregarded here because it should be demonstrated by materials science .The low-pressure section of the steam turbine attracts more attention in comparsion to the other sections because it has the most importance in power production. Moreover because of eliminating the enthalpy, it causes both more condensation and two-phase flow formation.

During the past decades, many different laboratory tests have been performed on simulation of two-phase flow behavior along with steam condensation. Before 2002, most studies have focused on laboratory and experimental tests. From 1967, Hill and Meyer published useful experimental data about nozzles [1]. Also Moore et al. continued the study of one-dimension flow in low-pressures on Laval nozzles [2]. Then Bakhtar et al. (1975) studied the flows in high-pressure nozzles [3]. Bakhtar and Zidi (1989-1990) used theoretical and experimental in the field of high pressure two-phase flows in one-dimension nozzles with different expansion rate [4]. According to these researches, the flow was inserted into the nozzle under superheat conditions and then it was changed into two-phase flow by decreasing the pressure and enthalpy, so the outlet of these flows was supersonic.

Modeling of steam-flow in steam turbines started by Young (1992) [5] through two-dimension calculation for nozzle and blade -tip with time-step method for temporary two-dimension flow. Bakhtar and Ebrahimi focused their experimental studies on geometry of rotor-tip section as static and reported the two-dimension condition results [6].

In 1995, Bakhtar and Mahpeykar studied about numerical solution of steam flow with time marching method and supposed the flow to be inviscid [7].

During the past years, two numerical solution have been suggested for the analysis of this method. Kermani and Gerber introduced and Eulerian - Lagrangian numerical method under verification of results by Moore et al. [8]. Also, Gerber and Kermani developed a non-equilibrium numerical method based on Eulerian-Eulerian method for two-phase transonic flow and properly validated their results at high and low pressure 2D flows through experimental data [9]. In 2006, Kermani and Zayernouri developed a modern numerical method by the use of equilibrium Thermodynamic Model. This methodology is based on changing the flow’s properties and thermodynamic relations in the equilibrium state; the quality parameter could be calculated in each point, so by using this method the condensation onset and the two-phase area completely can be recognized. Then, by using of equilibrium thermodynamic method under low-pressures Zayernouri and Kermani have developed an analytical one-dimensional method in steam flow through convergent-divergent nozzle [10].

In fact, from thermodynamic point of view condensation in rapid expanding flows is a non-equilibrium phenomenon. In the other word, condensation does not occur at the locus where the saturation line is crossed, i.e. the flow remains dry until the flow reaches the Wilson point, where condensation abruptly occurs. Following which the flow follows a pattern like that of the equilibrium case. Hence from non-equilibrium thermodynamics point of view condensation occurs slightly later than that of equilibrium thermodynamics model. However, beyond condensation, similar flow patterns are followed. The most important advantages of computations sing equilibrium model is its simplicity in computation algorithms.

According to the importance of the subject being studied, recognizing the parameters which have an effect on changing of two-phase flow will become more important. In this paper the factors affecting on changing of two-phase flow is reviewed:

1- Changes of geometry
- Investigation of vapor - Liquid flow by passing of divergent geometry.
- Investigation of vapor-liquid flow by passing of different rate of divergent geometry.

2- Changes of Upstream total flow
- Investigation of vapor-liquid flow based on different Total Temperatures.
- Investigation of vapor-liquid flow based on different Total Pressure.

3- Effects of Waves
- Investigation of vapor-liquid flow based on passing through compression waves.
- Investigate of vapor-liquid flow based on passing through expansion waves.

Firstly the most influencing parameters on the liquid formation are identified. Then a comprehensive parametric study is performed to obtain the trend of variations of the two-phase flow by changing the mentioned parameters. As stated in the previous section, the equilibrium thermodynamic method was used in order correctly predicts the trends of parametric changes and reduces the computation costs. In what follows, in each section, the changes of two-phase flow are shown. The above stated factors are considered as the most influencing factors on two-phase vapor-liquid flow.

This research will becomes more important, since the main flow features taking place in between turbines blades which are too similar to those in convergent-divergent nozzles. Therefore many scientists of this field such as Bakhtar have used this simplified geometry (nozzle) for modeling the flow among the turbine's blades. Hence, a correctly understanding of the effects in steam nozzles can give proper hints in designing the last stages of turbine blades, and provide a better control over condensation and its impact on the flow between the blades.

NOMENCLATURE

A	Area (m^2)
c_v	Vapor isochoric specific heat $\left(\frac{J}{kg.K}\right)$
E	Internal energy $\left(\frac{J}{kg}\right)$
F	Flux vector in x direction
F_1	Flux vector in ξ direction
G	Flux vector in y direction
G_1	Flux vector in η direction
h_{fg}	Enthalpy of evaporation $\left(\frac{J}{kg}\right)$

H	Total mixture enthalpy $\left(\frac{J}{kg}\right)$
J	Jacobian of transformation
M	Mach number
P	Pressure (kPa)
P_{back}	Back pressure (kPa)
Q	Conservative Vector
Q_1	Conservative Vector In General coordinates
R	Gas constant $\left(= 461.3999 \frac{J}{kg.K}\right)$
T	Temperature (K)
T	Time Coordinates (s)
U	Velocity in x direction $\left(\frac{m}{s}\right)$
V	Velocity in y direction $\left(\frac{m}{s}\right)$
X	Space Coordinate (m)
Y	Space Coordinate Perpendicular To x (m)
Greek symbols	
γ	Specific Heat Ratio (= 1.32 for vapor)
ξ	Space Coordinate In General coordinates (m)
η	Space Coordinate In General coordinates Perpendicular To ξ (m)
ρ	Mixture Density (vapor + liquid) $\left(\frac{kg}{m^3}\right)$
ρ_g	Gas (Vapor) Density $\left(\frac{kg}{m^3}\right)$
χ	The quality (mass fraction of vapor to that of the mixture)
Subscript	
f	Fluid (liquid)
fg	Interval of latent heat
g	Gas (vapor)
sat	Saturation
0	Stagnation (or total) condition

Governing Equations

The governing equations of fluid behavior for a two-dimensional, compressible, unsteady flow, inviscid with no body force in generals coordinate and full conservative form can be shown as [11, 12]:

$$\frac{\partial Q_1}{\partial t} + \frac{\partial F_1}{\partial \xi} + \frac{\partial G_1}{\partial \eta} = 0 \tag{1}$$

The quantities $Q_1$is the conservative vector and $F_1$is the flux vector in east and west direction and G_1 is the flux vector in north and south direction in general coordinates, and are given by:

$$Q = \begin{bmatrix} \rho \\ \rho u \\ \rho v \\ \rho e_t \end{bmatrix} \tag{2}$$

$$F = \begin{bmatrix} \rho u \\ P + \rho u^2 \\ \rho uv \\ \rho uH \end{bmatrix} \tag{3}$$

$$G = \begin{bmatrix} \rho u \\ \rho uv \\ P + \rho v^2 \\ \rho uH \end{bmatrix} \tag{4}$$

$$Q_1 = \frac{Q}{J} \tag{5}$$

$$F_1 = F\frac{\xi_x}{J} + G\frac{\xi_y}{J} \tag{6}$$

$$G_1 = F\frac{\eta_x}{J} + G\frac{\eta_y}{J} \tag{7}$$

$$J = \frac{1}{x_\xi y_\eta - x_\eta y_\xi} \tag{8}$$

For this study the pressure of the steam flow is lower than one atmosphere and the ideal gas equation of state is usage [13]:

$$P = \rho_g RT \tag{9}$$

In above formula ρ_g is the gas density and R is the gas constant. In dry regions, $\rho_g$is equal to the mixture density and in wet regions $\rho_g = \rho\chi$, where χ is the quality of the mixture. In this study, the volume of the liquid is ignored because it is supposed to be much less than that of the vapor.

The enthalpy of evaporation h_{fg} is obtained as follows:

$$h_{fg} = e_{fg} + RT \tag{10}$$

e_{fg} is the change in internal energy from the saturated vapor state to saturated liquid. A second-order polynomial can accurately represent the relationship between $e_{fg}$and T, and the coefficients are provided in [13]. The internal energy of the vapor, e_g, is determined by assuming a constant value for the specific heat at constant volume; that is,

$$e_g = c_v T \tag{11}$$

where $c_v = \frac{R}{\gamma - 1}$ with $\gamma = 1.32$ for vapor. Equation (11) indicates the saturated vapor internal energy with $e_g = c_v T_{sat}$, where can be used to determine the saturated liquid internal energy:

$$e_f = e_g(T_{sat}) + e_{fg} \tag{12}$$

The high resolution upwind scheme of Roe [14] is chosen for these analyses which is particularly useful for compressible flows; In this method determining "left" and "right" values at the cell faces, is to extrapolate the primitive variables (for example ,temperature T , pressure p , and velocities u, v) to cell faces using a high resolution MUSCL method [15]. These primitive variables are then used to determine the so-called Roe-averaged parameters corresponding to density, velocity in x, y direction and total enthalpy. Within wet regions, however, the primitive variables p and T are not independent and so a suitable substitute must be chosen to replace either p or T. In the present study, p is replaced by the mixture quality ρ, so that in wet regions T, u and ρ are used for the third order MUSCL extrapolation. The pressure within wet regions is simply evaluated using the saturated pressure at the local temperature.

The numerical method is spatially third-order accurate; with an explicit time-marching method that gives second-order accuracy in time. Any spurious numerical oscillations in the present high resolution computations have been damped using the van Albada flux limiter [16]. The expansion shocks have been avoided using the entropy correction formula given by Kermani and Plett [17].

NUMERICAL VALIDATION

To validate the results of numerical solution in this study, the pressure distribution on centerline of convergent-divergent nozzle is calculated and compared with the analytical results by Kermani and Zayernouri [10] and the experimental results by Moore et al. To improve the investigation, the convergent-divergent nozzles with different expansion rates are studied. So, nozzle A with maximum expansion rate for Moore nozzle Series and nozzle D with minimum expansion rate were examined. The inlet conditions for total pressure in both nozzles were 25 kPa and Total Temperature in nozzle A is 354.6 K and in nozzle D is 361.8 K. The outlet conditions were supersonic in both nozzles.

As shown by Figure 1, the conformity between the analytical and numerical results is apparent and there are a few differences when compared to experimental results. The existing difference comes from the difference between condensation onset location and the mass flow passing through the throat. In fact, under real conditions (experimental) the rapid expansion along the nozzle enters the flow in non-equilibrium thermodynamic condition, thus the condensation cannot start in the flow until it reaches proper super cooling degree. This matter causes a dried flow in the more length of the nozzle so it creates some differences between equilibrium and non-equilibrium conditions.

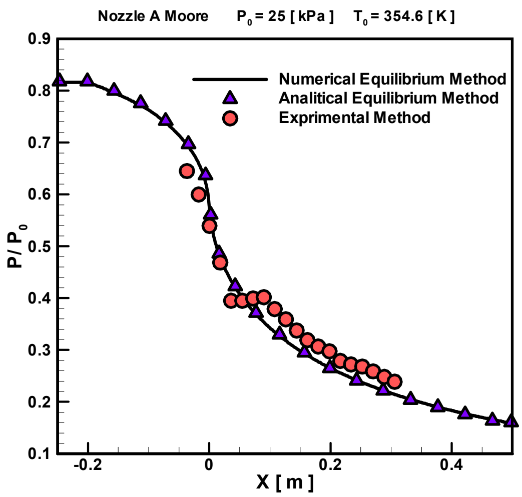

Figure 1: Validations of the present numerical algorithm vs. analytical solution and experimental data for A Moore nozzle.

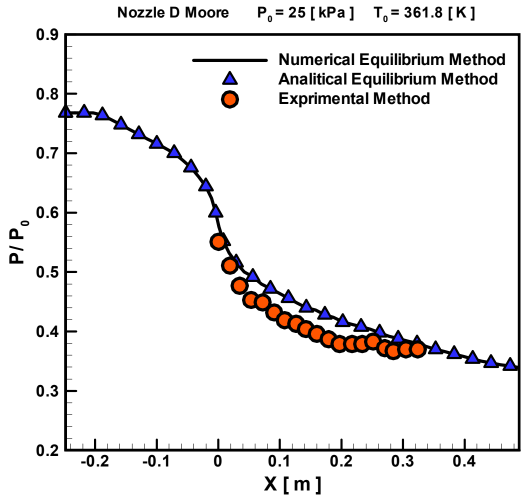

Figure 2: Validations of the present numerical algorithm vs. analytical solution and experimental data for D Moore nozzle.

EFFECT OF GEOMETRY

Type of Geometry is one of the parameters that affect the flow properties. In the other words, the convergence and divergence of geometry have direct effect on the changes of the flow parameters. Consider that the blade thickness in blade-to-blade domain is increased in a steam turbine cascade; by such a change the divergence rate of the blade is increased, in addition, by changing any geometrical parameter of the blade such as stager, maximum thickness, place of maximum thickness, etc., indeed the flow expansion or compression rates are changed. For this reason, now the effects of expansion and compression processes in an ideal state and then the effects of various rates of these changes, on two-phase flow will be studied.

By changing the geometrical conditions, the flow properties are changed. T-S thermodynamic diagram is used to show these changes. As shown by Figure 3, the geometrical changes could be simulated by considering the compression and expansion effects. It is necessary to say that this section is dealing with the geometry effects and the waves effect on the supersonic flow which will be described later. In the ideal expansion process, the flow will be accelerated. For Example, as the subsonic flow enters the convergent nozzle, its temperature falls and accelerates. Under these conditions two different states could be created: in the first state, the inflow (point 1) is a two-phase flow of liquid and vapor. Therefore, irrespective of the expansion process type and the rate of expansion, the quality of the flow will be decreased by passing through this process and finally it will be always in two- phase area (point 2).

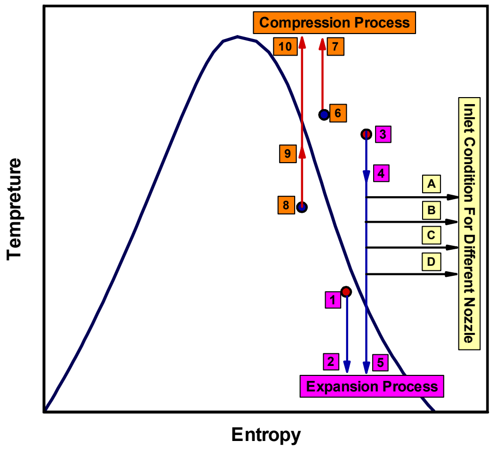

Figure 3: Illustration of isentropic compression/expansion processes of steam at various inflow conditions.

In the second state, the inflow is completely dry (point 3). So if the expansion process is continued enough by passing the saturation curve, the flow will be a complete two-phase flow (point 5). However, if the expansion process is not continued sufficiently or the rate of expansion is insufficient, it is possible for the flow by passing through expansion process that will be above the saturation curve and become completely dry (Point 4).

For an ideal compression process, as shown in Figure 3, it can be seen that if the process is continued enough or the rate of process-progress is increased even if the inflow is two-phase, in any case it is possible to create the dry flow. So by reviewing the steam turbine blade-to-blade domain in rotor tip often the flow is shown around the pressure surface as completely dry.

Since in most applications related to steam flow such as steam nozzle and steam turbine, the expansion of flow is considered important, in this section the expansion process and variation of the expansion rate will be studied. To study these changes, a flow with inlet total temperature of 361.8 K and total pressure of 25 kPa in nozzles of A, B, C, D which is shown in Figure 4 are examined.

Note that despite the dissimilarity of the inlet area and the throat in the convergent section, the convergence angles of these nozzles are the same, and in this section, the rate of flow expansion is equal in all nozzles. As shown in Figure 4, the rate of flow expansion is completely different in the divergent section, nozzle A has the maximum rate of expansion and nozzle D has the minimum.

When looking at Figure 5, as shown in T-S thermodynamic diagram in Figure 3, when passing through an expansion process the flow static temperature decreases until it reaches the saturation diagram and then the flow becomes a two-phase. 2D contours are shown in annex A.

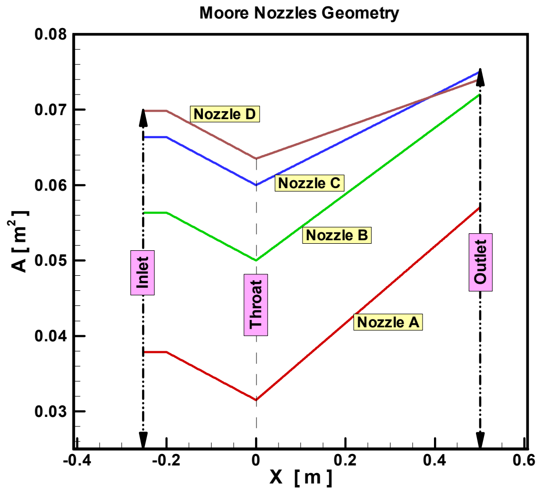

Figure 4: Geometry of four different nozzles A, B, C, D adopted from Moore et al. [2].

Since the total condition and the expansion rate in the convergent section of these nozzles are the same, the condensation onset will take place faster in a nozzle which has a lower static inlet temperature and is close to the saturation line. According to this hypothesis, the nozzle with a bigger Mach number shall have a lower static temperature and the condensation of flow will take place faster in that geometry. This matter is shown in the thermodynamic diagram, and among the examined nozzles the inlet conditions in nozzle D is closer to the saturation line. As shown in Figure 6, nozzle D shows the least inlet area to throat area ratio and it has the maximum Mach number at the inlet. On the contrary, nozzle A has the minimum Mach number. For this reason as shown in Figure 5, in nozzle A condensation starts faster than the other nozzles.

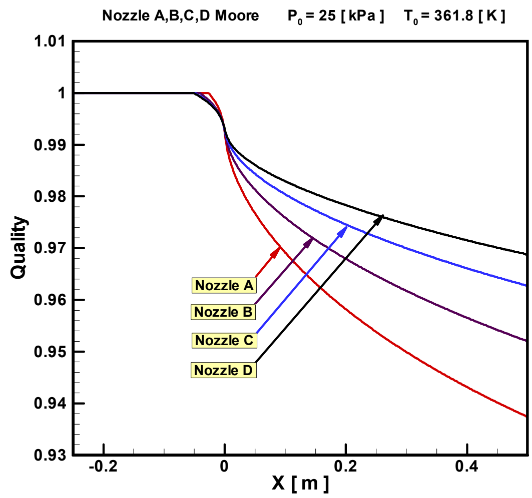

Figure 5: Quality distribution along the A to D Moore nozzles centerlines.

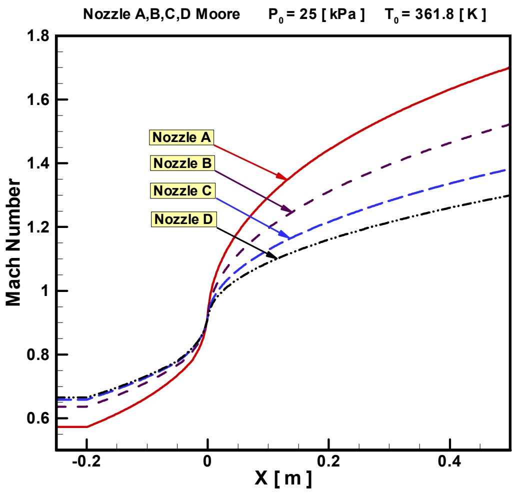

Figure 6: Mach number distribution along the A to D Moore nozzles centerlines.

After investigating the condensation onset, the effects of expansion rate in the divergent section will be studied. As mentioned before, nozzle A in the divergent section has the maximum expansion rate. As shown in Figure 5, in nozzle A in which the condensation onset was delayed in comparison with the other nozzles, the rate of second phase formation in the outlet is more. Therefore, as this diagram shows, it could be concluded that by increasing the rate of expansion, the rate of flow condensation is also increased.

EFFECTS OF INLET TOTAL CONDITION

One of the effective parameters on the two-phase flow is the effect of flow parameters on the total condition. So the effect of both total temperature and total pressure parameters are considerable. The total temperature shows the total energy of flow and in turbo-machine applications it always shows the internal and the kinetic energy of the flow. In addition, note that changing the value of T_0 in the length of steam turbine has a direct relation with output works; the next parameter in this section is the total pressure, P_0. As it will be shown, the effect of these two parameters on the two-phase flow is important.

EFFECT OF TOTAL TEMPERATURE

To study the effects of inlet total temperature, under constant inlet total pressure (P_0=25 kPa) into the nozzle A and under supersonic outlet conditions, the flow is studied for different inlet total temperature.

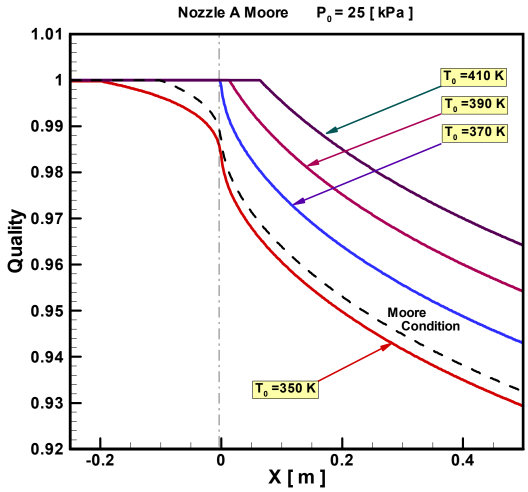

Figure 7: Quality distribution along the A Moore nozzle; a parametric study representing the influence of inflow total temperature.

As shown in Figure 7, by increasing the total temperature, condensation will be moved to the downstream. For further description, the T-S thermodynamic diagram is used. According to Figure 8, in a constant pressure curve which shows the inlet total pressure, increasing the total temperature in isentropic flow, will cause an increase in the temperature difference. This temperature difference shows the flow expansion from total condition to condensation onset, which is shown as ΔT in Figure 8 and by increasing the temperature from 1 to 3, this quantity will increase as well.

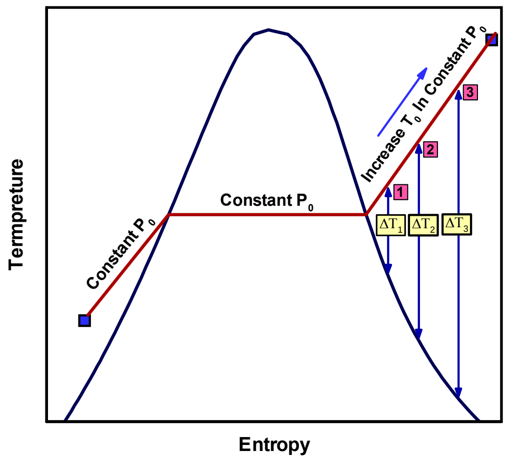

Figure 8: Schematic temperature-entropy diagram, representing the influence of inflow total temperature on the condensation onset point.

Another important note in thermodynamic diagram is that if total temperature increases, the static temperature of condensation onset will decrease. As shown in Figure 8 by increasing the total temperature, from isentropic state 1 to isentropic state 2 and 3, the static temperature of condensation onset will decrease significantly.

When reviewing Figure 8 it could be understood that by increasing the inlet total temperature the value of flow expansion will increase in the dried area and the start of condensation will take place under the lower temperature. For these reasons by increasing the total temperature the condensation point will move to downstream in the length of nozzle.

Note that there is more sensitivity in the condensation onset point in a supersonic region with regards to variation the inlet total temperature. In Figure 7, see the curve of temperatures 350 K and curve of total temperature 354.6 K (Moore condition) and note that the condensation onset point will be replaced even by this small change. However the change of condensation onset point in the supersonic area comparing with the increase of the total temperature is insignificant.

If a diagram of Mach number is drawn for different inlet total temperatures two points shall be determined by investigating the two-phase flow:

First, as shown in Figure 9, the value of Mach number after condensation onset decreases compared with that for dried flow. This difference will be developed by increasing the liquid-phase percentage, because by start of condensation and formation of droplets, the latent heat of condensation will be transferred to gas-phase that causes an increase in the static temperature. So the value of sonic velocity will increase and the Mach number of gas-phase will decrease.

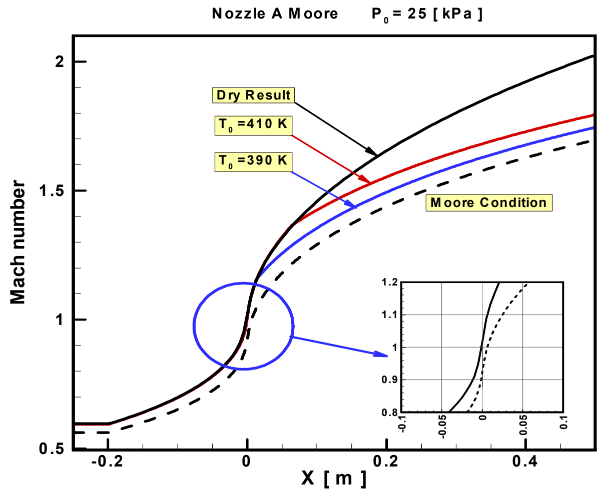

Figure 9: Mach number distribution along the A Moore nozzle; a parametric study representing the influence of inflow total temperature. As can be seen in the enlarged picture, in the case of condensation upstream of throat sonic condition occurs downstream of throat.

Second, for the case of a flow through a converging-diverging nozzle when heat along the nozzle is added, the following relation exists between the Mach number, A and T_0 [18]:

$$\frac{dA}{A} = \frac{dM}{M}\frac{(M^2-1)}{\left(1+\frac{(\gamma-1)}{2}M^2\right)} + \frac{(\gamma M^2-1)}{2}\frac{dT_0}{T_0} \tag{13}$$

According to Eqn. (13), in the case of adiabatic flow (T_0 =constant), sonic condition occurs right at the throat. However, in the case of heat addition ($dT_0 > 0$), sonic condition (M=1) occurs in the diverging portion of the nozzle ($dA > 0$). In the present study, with the flow under condensation process, the total temperature of the vapor phase (due to the latent heat removal toward the vapor phase) increases in condensing region ($dT_0 > 0$). Hence, the sonic condition occurs downstream of the nozzle throat and for condition that condensation onset occurred after throat the sonic line happen right at the throat. This phenomenon is clearly shown in the enlarged picture in Figure 9.

EFFECT OF INLET TOTAL PRESSURE

For studying the effect of changing the inlet total pressure under constant inlet total temperature 360 K and in A Moore nozzle, the outlet supersonic flow was studied for four different total pressures.

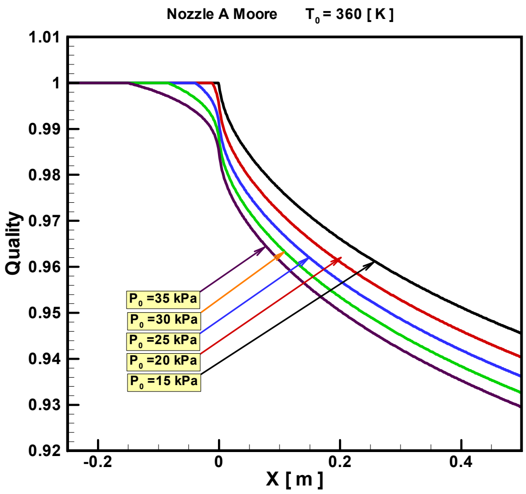

Figure 10: Quality distribution along the A Moore nozzle; a parametric study representing the influence of inflow total pressure.

By looking at Figure 10, one can understand that by increasing the inlet total pressure the condensation onset point will be moved to the upstream and the value of liquid-phase will be increased at the nozzle outlet.

To a better explanation of these changes, T-S thermodynamic diagram is investigated. As shown in Figure 11, under a constant total temperature condition, if the total pressure is decreased (from 1 to 3), the temperature difference which shows the value of the flow expansion from total point to the condensation onset, will increase. Its main cause is related to the decrease of temperature in condensation onset under increasing the inlet total pressure. So, as shown in Figure 11, condensation onset point occurs faster when inlet total pressure is decreased.

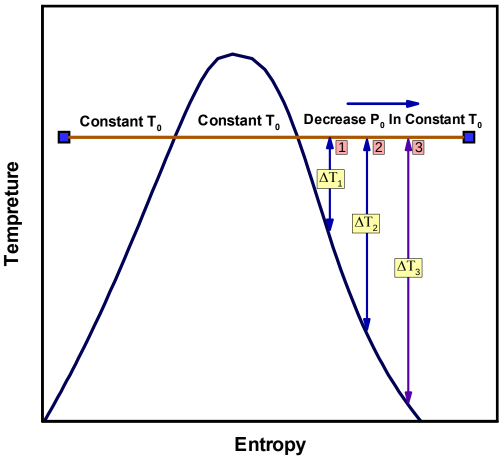

Figure 11: Schematic temperature-entropy diagram, representing the influence of inflow total pressure on the condensation onset point.

By comparing Figures 8 and 11, it could be understood that although increasing the inlet total temperature and decreasing inlet total pressure would move the condensation onset point to the downstream, but the effects of changing the total temperature are more significant than the changes of the condensation onset point.

EFFECTS OF WAVES

At the last rows of steam turbine where the blades height is maximum, it is possible to form the liquid-phase on the upper section of the blade. In this region, according to the high rotational speed of the shaft and the large radius of the blade it is possible to create a supersonic flow. However, with regards to the physics of the turbine blades and accelerating the flow, the flow is chocked in the throat and then the supersonic flow will be created.

By creating the supersonic flow after the blade throat, flow data couldn't be spread from downstream to upstream. So the flow tends to exit the blade without sensing the outlet pressure. Therefore, by spreading the waves into the downstream, the flow pressure will be adjusted by the outlet pressure. The pressure is adjusted as follows: if the flow pressure is less than the outlet pressure, for low pressure differences the oblique shock waves are spread to the downstream and for high pressure differences a normal shock wave is formed and is progressed into blade passage; the flow pressure will be adjusted by the outlet pressure. If the flow pressure is more than the outlet pressure, by forming expansion waves at the trailing edge and spreading them to the downstream, the flow pressure will be decreased by the outlet pressure.

In addition to that stated above, another practical sample, that significantly emphasizes the role of waves, is their spreading towards the upstream. If there is a supersonic inlet flow and the axial Mach number is less than unity, the compression and expansion waves that emanated from suction surface will be spread towards the upstream and influence the inlet flow properties.

EXPANSION WAVES

For investigating the effects of expansion waves on the two-phase flow the geometry of Figure 12 was used.

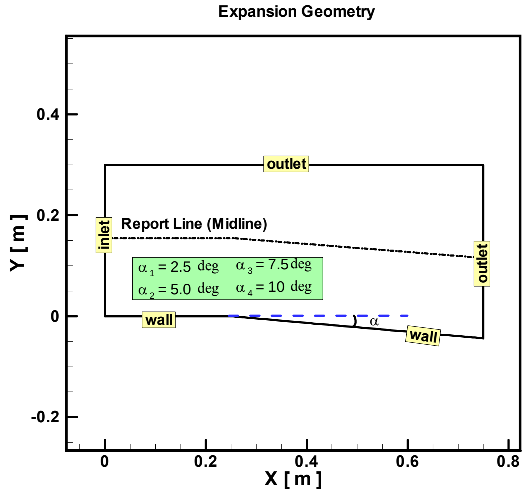

Figure 12: Schematic of the geometry used for condensing flows over expansion fans.

The supersonic flow enters the physical domain completely uniform and horizontally. When it faces the expanding corner, it changes its direction when passing through an expansion fan so that it is parallel to that surface. As shown in Figure 12, the left boundary is considered as the inlet, the right boundary as the outlet, the lower boundary as the wall and the upper boundary as the outlet. To study the effect of different expansion waves, under constant inlet condition in a static temperature 340 K, static pressure 30 kPa and Mach number 1.1, the different divergent angles were examined. To show and compare the results, the data of centerline was used and 2D contours were shown in annex A.

As shown in Figure 13, by increasing the divergent angle, the amount of flow condensation will also increase.

In supersonic flows, since the data of the downstream couldn't be spread to the upstream, the upstream cannot be influenced by the changes of the downstream conditions. Figure 13 clearly shows that when the flow faces a divergent surface it shall pass from a series of expansion waves. These waves will continue until the flow becomes parallel to the diverging surface. However the upstream flow is not aware of the value of divergent angle, for this reason the condensation will start from one point in all the four geometries and the angle of the first wave that goes to the downstream only relates to the upstream Mach number which is similar for all geometries. On the other hand, changing the size of the expansion fans did not change the condensation onset point.

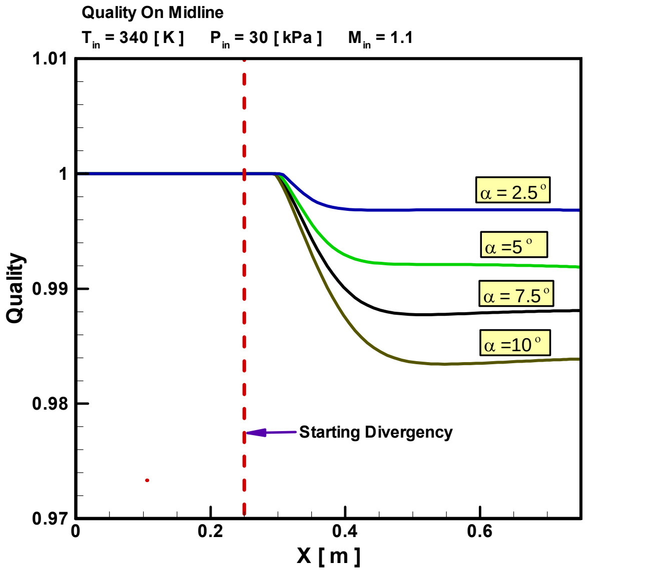

Figure 13: Quality distribution over an expansion fan; a parametric study representing the influence of expanding corner angle. As depicted in this figure, the larger the fans, the higher condensations occur.

If the passage of a supersonic flow through an expansion fan is drawn by a T-S thermodynamic diagram, the variations of two-phase flow can be well investigated.

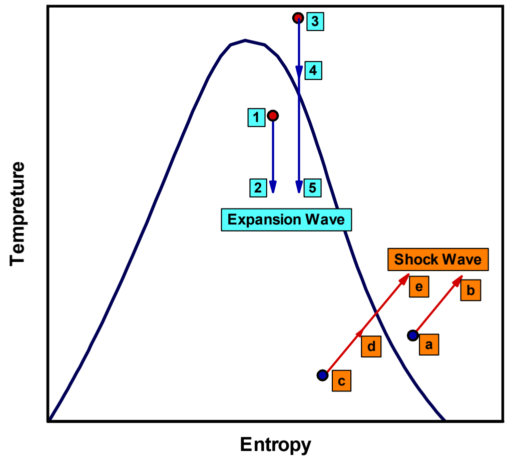

Figure 14: Schematic temperature-entropy diagram, representing a steam flow through either an isentropic expansion fan (processes 1 to 2, or 3 to 5), or shock waves (a to b, or c to d).

The process of passing the flow through an expansion fan is a reversible process; so since the equilibrium thermodynamic method is modeling the condensation phenomenon completely reversible; entropy parameter could be constant when passing through the expansion fan. As shown in Figure 14, by passing a flow through expansion waves, in an isentropic process, the flow temperature will decrease. So the creation of two different states is possible: First, the inflow (point 1) is a two-phase flow that is independent of the size of the expansion fan, the quality of the flow will decrease and it will remain in the two-phase state (point 2). Second: The inflow is completely dry (point 3) and if the expansion fan is wide enough it could pass through the saturation curve and liquid-phase will be created (point 5). But if the expansion fan is not wide enough, the flow after passing through the expansion waves may remain completely dry (point 4).

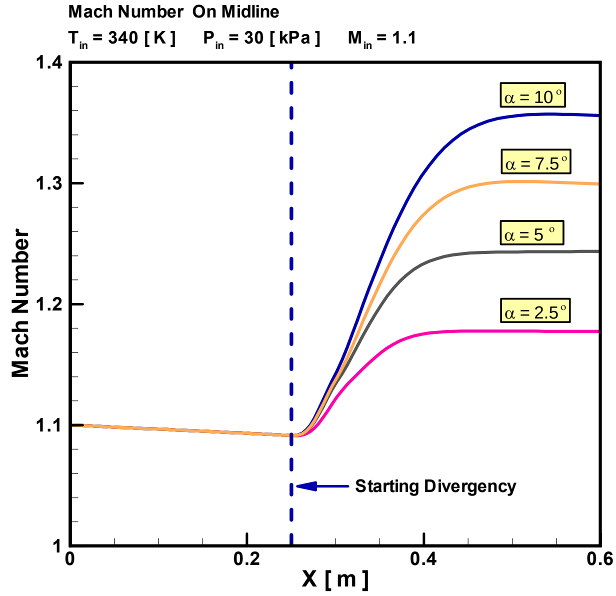

Figure 15: Mach number distribution over an expansion fan; a parametric study representing the influence of expanding corner angle.

In Figure 15, the changes of Mach number for flow during passage of the expansion fan are shown. Note that the size of the expansion fan in quality diagram, shown in Figure 13, is smaller than the Mach diagram, shown in Figure 15 (compare Figure 22 and 23 in annex A), i.e. at first the dry flow was accelerated by passing through the expansion fans and its static temperature is decreased until it reaches the saturation curve. At this point, although the flow is accelerated and the Mach number is increased but the flow is completely dry; as a result, the part of expansion fan that is located at the first section, could not be shown in quality diagram, shown in Figure 13.

SHOCK WAVES

To investigate the effects of shock waves, different back pressures under constant conditions of inlet total temperature of 354.6 K, inlet total pressure of 25 kPa and the geometry of A Moore nozzle, have been used. In general, in a supersonic flow both compression and expansion waves are seen in the following condition:

1) When the supersonic flow rotates to follow another surface.
2) When the pressure of the supersonic flow shall be adjusted by the outlet pressure.

In this section, by increasing the back pressure in convergent-divergent nozzles, the normal shock will be initially formed on the outlet. Then increasing the back pressure can move back the shock wave along the nozzle until the flow pressure becomes equal to the outlet pressure.

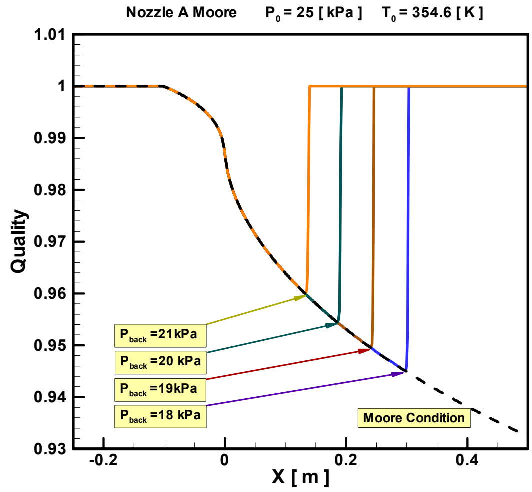

Figure 16: Quality distribution through shock waves; a parametric study representing the influence of back pressure for A Moore nozzle.

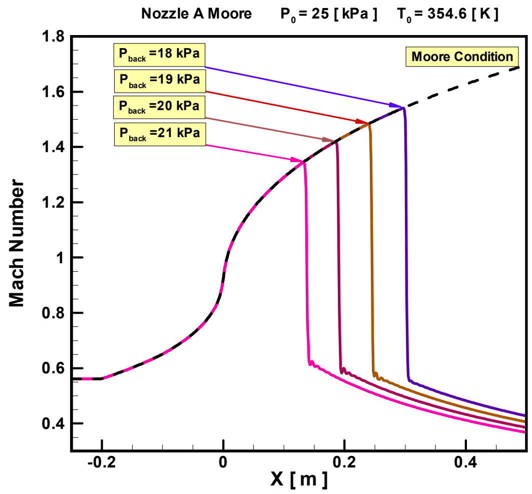

Figure 17: Mach number distribution through shock waves; a parametric study representing the influence of back pressure for A Moore nozzle.

As shown in Figure 16, the wet flow will be dried when waves come across the normal shock.

As shown in Figure 17, by increasing the outlet pressure, the shock wave will move to the upstream and the value of Mach number behind the normal shock and its strength will decrease. The relevant 2D contours are shown in annex A.

As shown in T-S thermodynamic diagram, by passing the flow through the normal shock, the temperature increases and the amount of entropy will be increased as well. If the dry flow (point a) is encountered a shock wave, surely it will remain dry after passing through the shock wave (point b); however if the inflow is wet (point c) the state of the outflow depends upon the temperature, quality and Mach number of the shock wave. So, by changing the above parameters, it is possible to have one of the completely dry flow (point f) or the two-phase flow (point e).

Figure 16 shows that the two-phase flow will be dried by passing through the shock waves. Another difference between the shock waves in Figure 13 and the expansion waves in Figure 16 is that the flow in the shock waves will reach the final state suddenly, but the flow in the expansion waves will pass through this path continuously; however when flow passes through the shock waves the entropy will increase since the passage of the flow through the expansion waves is completely isentropic.

CONCLUSION

In this paper, by passing the flow through compression and expansion processes it has been concluded that:

- When the liquid-phase flow is passing through an ideal compression process, by increasing the length of the process or the rate of compression, a completely dry flow can be made.
- If the inflow into an ideal expansion process is completely dry, by increasing the length of the process or the rate of expansion, the liquid droplets can be formed.
- By increasing the inlet total temperature and decreasing the inlet total pressure the condensation point will be delayed.
- If condensation takes place before the throat, the local total temperature at the throat will increase and the mass flow will decrease in proportion to the dried throat. Furthermore the physical throat, at which the Mach number is unity, is located ahead of the nozzle geometry throat.
- By creating a two-phase flow, the latent heat of liquid phase will be transferred to the steam phase, thereby increasing the static temperature and decreasing the Mach number of gas-phase in comparison with a completely dry sate.
- The waves which are spread to the downstream for the rotation of the supersonic flow cannot change the condensation onset point.
- By passing a dry flow through wide enough expansion fans a liquid phase can be formed.
- By passing a two-phase flow through shock waves, the flow after the wave is completely dry, provided that the shock wave is strong enough. Otherwise it is possible for the flow to be wet.

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ANNEX A

In this Annex the contour plots of Mach number and quality levels, obtained from the present 2D computations, are shown.

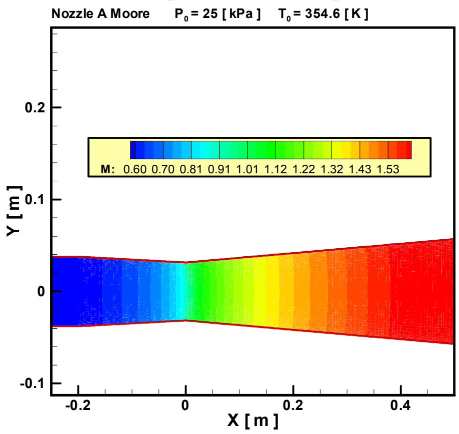

Figure 18: Mach number contours in A Moore nozzle computation.

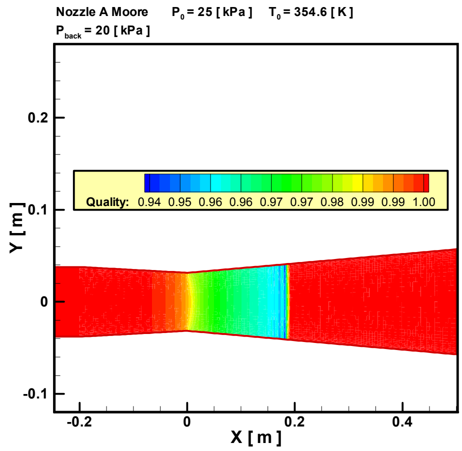

Figure 21: Quality contours in A Moore nozzle computation with normal shock in the diverging portion of the nozzle.

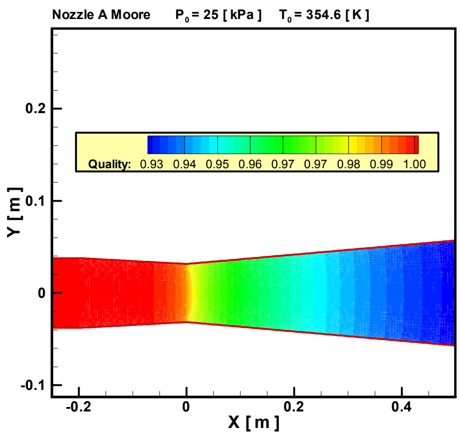

Figure 19: Quality contours in A Moore nozzle computation.

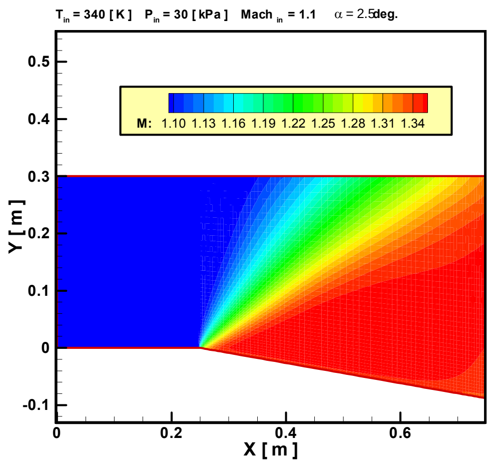

Figure 22: Mach number contours over a condensing expanding corner.

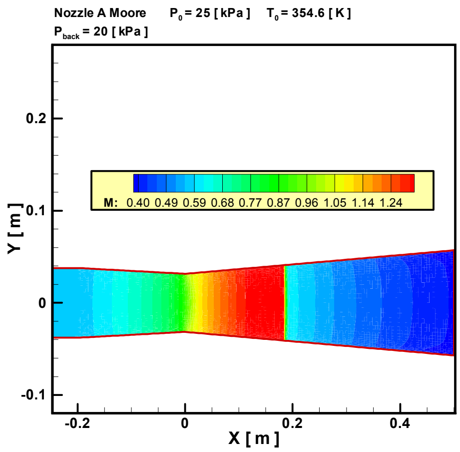

Figure 20: Mach number contours in A Moore nozzle computation with normal shock in the diverging portion of the nozzle.

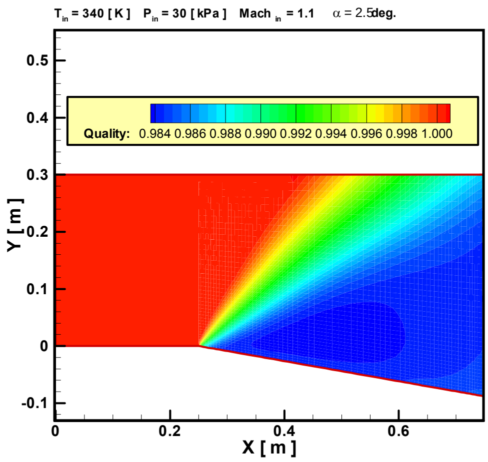

Figure 23: Quality contours over a condensing expanding corner.