Equatorial Journal of Engineering (2018) 9- 13
Journal Homepage: www.erjournals.com
ISSN: 0184-7937

EVALUATION OF THE BEHAVIOUR OF STEAM EXPANDED IN A SET OF NOZZLES, IN A GIVEN TEMPERATURE

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ABSTRACT
Nozzles are essentially adiabatic devices and are used to accelerate a fluid; therefore, isentropic process serves as a suitable model for nozzles. The paper discussed the characteristics of an expanding steam passing through a duct of varying cross-section area. Based on the converging-diverging nozzle, the research investigated the condition of steam at the throat and exit, stating it to be superheated. With a critical temperature and critical velocity of 304.35 ^{o}C and 819.11 m/s respectively, the minimum area of the nozzle *measured 0.35333 m^2. The expansion through the set of nozzles was largely isentropic, meaning that it is adiabatic and reversible. It was assumed that the fluid velocity and the fluid properties, change, only in the direction of flow. Nozzles are used, in practice, in steam and gas turbines, in jet engines, in rocket motors, in flow measurement, and in many other engineering applications.*

Keywords: Critical Conditions, Ambient Conditions, Expanded Steam, Converging-Diverging Nozzles

How to cite this article: Madu, K. E. (2018). Evaluation of the Behaviour of Steam Expanded in a Set of Nozzles, in a Given Temperature. *Equatorial Journal of Engineering* (2018) 9-13.

INTRODUCTION

A nozzle is a duct of smoothly varying cross-sectional area in which a steadily flowing fluid can be made to accelerate by a pressure drop along the duct. A steam nozzle is as passage of varying cross-section, which converts the heat energy of steam into kinetic energy (Ballaney, 2011). During the first part of the nozzle, the steam increases its velocity. But, in its later part, the steam gains more in volume than in velocity (Uppal and Rao, 2012). Since the mass of steam of steam passing through any section of the nozzle remains constant, the variation of steam pressure in the nozzle depends upon the velocity, specific volume, and dryness fraction of the steam. A well designed nozzle converts the heat energy of steam into kinetic energy, with a minimum loss. The main use of steam nozzle in steam turbines is to produce a jet of steam with a high velocity (Rajput, 2013). The smaller section of the nozzle is called throat. In practice, there are many applications which require a high-velocity stream of fluid; and the nozzle is the best means of obtaining this. For example, nozzles are used in steam and gas turbines, in jet engines, in rocket motors, in flow measurement, and in many other applications. When a fluid is decelerated in a duct, causing a rise in pressure along the stream, then the duct is called a diffuser; two applications, in practice, in which the diffuser is used, are the centrifugal compressor and the ramjet. The analysis presented in this paper, will be restricted to one-dimensional flow. In one-dimensional flow, it is assumed that the fluid velocity and the fluid properties change, only in the direction of flow. This means that the fluid velocity is assumed to remain constant at a mean value across the cross-section of the duct. The effects of friction will not be analysed fundamentally, suitable efficiencies and coefficients being adopted to allow for the departure from the ideal frictionless case. The analysis of fluid flow involving friction has become of increasing importance, due to the development of the turbojet, the ramjet, and the rocket, and the introduction of high-speed flight.

According to Khurmi, and Gupta, (2008), the following three types of nozzle are important, from the subject point of view:

Convergent Nozzle. When the cross-section of the nozzle decreases continuously from entrance to exit, it is called a convergent nozzle.

Divergent Nozzle. When the cross-section of the nozzle increases continuously from entrance to exit, it is called a convergent nozzle.

Convergent-Divergent Nozzle. When the cross section of the nozzle first decreases from its entrance to throat, and then increases from its throat to exit, it is called a convergent-divergent nozzle. This type of nozzle is widely used these days in various types of steam turbines.

In a convergent-divergent nozzle, steam enters the nozzle with a high pressure, but with a negligible velocity. In the converging portion (i.e. from the inlet to the throat), there is a drop in steam pressure with a rise in its velocity. There is also a drop in the enthalpy or total heat of the steam. This drop of heat is not utilized in doing some external work, but is converted into kinetic energy. In the divergent portion (i.e. from the throat to the outlet), there is a further drop of steam pressure, with a further rise in its velocity. Again, there is a drop in the enthalpy or total heat of steam, which is converted into kinetic energy. It will be interesting to know that steam enters the nozzle with a high pressure and negligible velocity, but leaves the nozzle with a high velocity and small pressure. The pressure at which the steam leaves the nozzle, is known as *back pressure* (Yunus, and Michael, 2011). Moreover, no heat is supplied or rejected by the steam during flow through a nozzle. Therefore, it is considered as isentropic flow, and the corresponding expansion (i.e. decrease in pressure), is considered as an isentropic expansion. In the above analysis, it is assumed that no friction exists between the nozzle surface and the flowing steam.

As regards nozzle shape, consider a stream of fluid at pressure, p_1; enthalpy, $h_;$ and with a low velocity, C_1. It is required to find the shape of a duct which will cause the fluid to accelerate to a high velocity as the pressure falls along the duct. It can be assumed that the heat loss from the duct is negligibly small (i.e. adiabatic flow, Q = 0), and it is clear that no work is done on or by the fluid (i.e. W = 0). The steady-flow energy equation is given as:

$$\dot{m}(h_1 + \frac{C_1^2}{2} + Z_1 g) + Q + W = \dot{m}(h_2 + \frac{C_2^2}{2} + Z_2 g) \quad (1)$$

In a steady flow, the rate of mass flow of fluid at any section is the same as at any other section. Applying the steady-flow energy equation, (Equation 1), between section 1 and any other section X – X where the pressure is p, the enthalpy is h, and the velocity is C, we have;

$$h_1 + \frac{C_1^2}{2} = h + \frac{C^2}{2}$$

i.e. $C^2 = 2(h_1 - h) + C_1^2$

or $C = \sqrt{\{2(h_1 - h) + C_1^2\}}$ (2)

If the area at the section X – X is A, and the specific volume is v, then, using equation for mass flow rate;

Mass flow, $\dot{m} = \frac{CA}{v}$

The area per unit mass flow is given as: $\frac{A}{\dot{m}} = \frac{v}{C}$ (3)

Then, substituting for the velocity C, from equation 2

$$\text{Area per unit mass flow} = \frac{v}{\sqrt{\{2(h1 - h) + C_1^2\}}} \quad (4)$$

It can be seen from Equation 4 that in order to find the way in which the area of the duct varies, it is necessary to be able to evaluate the specific volume, v, and the enthalpy, h, at any section X – X. To be able to do this, some information about the process undergone between section 1 and section X – X, must be known. For the ideal frictionless case, since the flow is adiabatic and reversible, the process undergone is an isentropic process, and hence;

s_1 = (entropy at any section X – X) = s

Now, using Equation 3, and the fact that $s_1 = s$, it is possible to plot the variation of the cross-sectional area of the duct against the pressure along the duct. For a vapour, this can be done using tables; for a perfect gas, the procedure is simpler, since we have

pv^y = constant, for an isentropic process. In either cases, choosing fixed inlet conditions, then the variation in the area, A, the specific volume, v, and the velocity, C, can be plotted against the pressure along the duct (Eastop, and McConkey, 1993). Typical curves are shown in Figure 1.

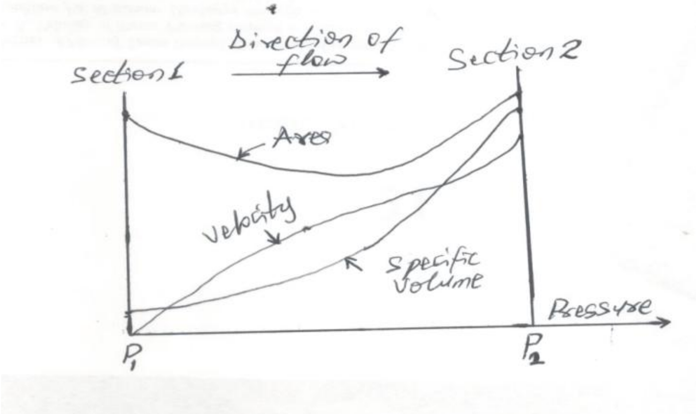

Figure 1: Cross- Sectional Area, Velocity, and Specific Volume Variations with Pressure through a Nozzle

It can be seen that the area decreases initially, reaches a minimum, and then increases again. This can, also, be seen from Equation 3. When v increases less rapidly than C, then the area decreases; when v increases more rapidly than C, then the area increases. A nozzle, whose area varies as in Figure 1, is called a convergent-divergent nozzle. The section of minimum area is called the throat of the nozzle. The velocity at the throat of a nozzle operating at its designed pressure ratio, is the velocity of sound at the throat conditions. The flow up to the throat is subsonic; the flow after the throat is supersonic. It should be noted that a sonic or a supersonic flow requires a diverging duct to accelerate it. Worthy of note is the fact that the specific volume of a liquid is constant over a wide pressure range. Therefore, nozzles for liquids are always convergent, even at very high exit velocities (e.g. a fire-hose uses a convergent nozzle).

MATERIAL AND METHOD

Steam is expanded in a set of nozzles from 10 bar, 350 ^oC to 3 bar as shown in Figure 2. Assuming an isentropic expansion, and a flow of 300 Kg/min, we are to determine the following;

- Condition of steam at the throat
- Condition of steam at the exit
- Critical temperature
- Critical velocity
- Critical Enthalpy
- Minimum area of nozzles
- Exit velocity
- Exit area
- Quality of steam at the exit plane
- Diameter of the nozzles at the exit, for a set of 5 nozzles
- Type of Nozzle

RESULTS AND DISCUSSION

At a pressure of 10 bar, T_s = 179.88 ^oC (cf. Saturated water – Pressure table)

10 bar = 1000 kPa; and $\frac{P^*}{P_0} = 0.546$

Thus, $P^* = P_0 \times 0.546 = 10 \times 0.546 = 5.46$ bar

From the problem statement,

$S_0 = S^* = S_1$ = 8.9155 kJ/KgK

h_0 = 4640.0 kJ/Kg

Task 1: Determination of Condition of Steam at the Throat

P^* = 5.46 bar

S^* = 8.9155 kJ/KgK

But S_g at 500 kPa (5 bar) = 6.8207 kJ/KgK

Also S_g at 600 kPa (6 bar) = 6.7593 kJ/KgK

Since S_g at 5 bar and 6 bar < S^*, then, the condition of steam at the throat is superheated.

Task 2: Determination of Condition of Steam at the Exit

P_1 = 3 bar

S_1 = 8.9155 kJ/Kgk

But S_g at 300 kPa (3 bar) = 6.9917 kJ/KgK

Since S_g at 3 bar < S^*, then, the condition of steam at the exit is superheated.

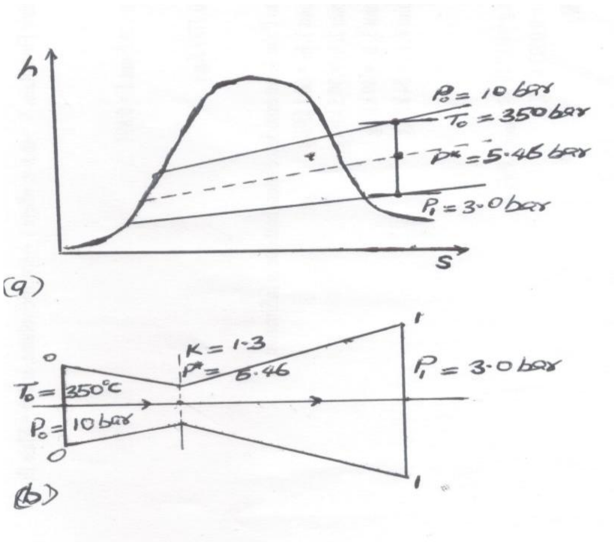

Figure 2: h-s Graph of Convergent-Divergent Nozzle

Task 3, and 4: Determination of Critical Temperature, and Critical Velocity

The condition of steam at the throat, it must be reiterated, is superheated.

At the throat,

P* = 5.46 bar

S* = 8.9155 kJ/KgK

For P_{500} kPA, by interpolation, we have (cf. Superheated Water Table);

$$\frac{S^*_{500}-S^{*I}_{500}}{S^{*II}_{500}-S^{*I}_{500}} = \frac{h^*_{500}-h^{*I}_{500}}{h^{*II}_{500}-h^{*I}_{500}} = \frac{V^*_{500}-V^{*I}_{500}}{V^{*II}_{500}-V^{*I}_{500}} = \frac{T^*_{500}-T^{*I}_{500}}{T^{*II}_{500}-T^{*I}_{500}}$$

$$\frac{8.9155-8.8240}{9.0362-8.8240} = \frac{h^*_{500}-4158.4}{4396.6-4158.4} = \frac{V^*_{500}-0.98966}{1.08227-0.98966} = \frac{T^*_{500}-800}{900-800}$$

Therefore,

h^*_{500} = 4261.11 kJ/Kg

V^*_{500} = 1.0296 m^3/Kg

T^*_{500} = 843.12 °C

For P_{600} kPA, by interpolation, we obtain

$$\frac{S^*_{600}-S^{*I}_{600}}{S^{*II}_{600}-S^{*I}_{600}} = \frac{h^*_{600}-h^{*I}_{600}}{h^{*II}_{600}-h^{*I}_{600}} = \frac{V^*_{600}-V^{*I}_{600}}{V^{*II}_{600}-V^{*I}_{600}} = \frac{T^*_{600}-T^{*I}_{600}}{T^{*II}_{600}-T^{*I}_{600}}$$

$$\frac{8.9155-8.7395}{8.9518-8.7395} = \frac{h^*_{600}-4157.9}{4396.2-4157.9} = \frac{V^*_{600}-0.82456}{0.90179-0.82456} = \frac{T^*_{600}-800}{900-800}$$

Therefore; h^*_{600} = 4355.45 kJ/Kg

V^*_{600} = 0.8886 m^3/Kg

T^*_{600} = 882.9 °C

For P_{546} kPA, by interpolation, we get;

$$\frac{P^*_{546}-P^*_{500}}{P^*_{600}-P^*_{500}} = \frac{h^*_{546}-h^*_{500}}{h_{600}-h^*_{500}} = \frac{V^*_{546}-V_{500}}{V_{600}-V_{500}} = \frac{T_{546}-T_{500}}{T_{600}-T_{500}}$$

$$\frac{546-500}{600-500} = \frac{h^*_{546}-4261.11}{4355.45-4261.11} = \frac{V_{546}-1.0296}{0.8886-1.0296} = \frac{T^*_{546}-843.12}{882.9-843.12}$$

Therefore, Specific Volume, V^*_{546} = 0.9647 m^3/Kg

Enthalpy, h^*_{546} = 4304.51 kJ/Kg

Temperature, T^*_{546} = 861.47 °C

But, $\frac{T^*}{T_0} = \frac{2}{r+1} = \frac{2}{1.3+1}$ = 0.8696

Critical Temperature, T* = 350 x 0.88696 = 304.35 °C

Critical Velocity, V* = 44.72(h_0 - h^*_{546})$^{1/2}$
= 44.72(4640-4304.51)$^{1/2}$
= 819.11 m/s

Task 5: Determination of the Minimum Area of Nozzle

A* = $\frac{\dot{m}v^*}{V^*}$; where v^* = Critical Specific Volume; and V^* = Critical velocity

A* = $\frac{300\ x\ 0.9647}{819.11}$ = 0.35333 m^2

Task 6: Determination of the Exit Velocity

Recall that the condition at the exit is superheated and that;

S_1 = 8.9155 kJ/KgK, and P_1 = 3 bar (300 kPa)

Interpolating, we obtain;

$$\frac{S_1-S^I_I}{S^{II}_I-S^I_I} = \frac{h_I-h^I_I}{h^{II}_I-h^{II}_I} = \frac{V_I-V^I_I}{V^{II}_I-V^I_I} = \frac{T_I-T^I_I}{T^{II}_I-T^I_I}$$

$$\frac{8.9155-8.8345}{9.0605-8.8345} = \frac{h_I-3928.2}{4159.3-3928.2} = \frac{V_I-1.49580}{1.65004-1.49580} = \frac{T_I-700}{800-700}$$

Therefore, h_I = 4011 kJ/KgK
v_I = 1.5511 m^3/Kg
T_I = 735.84 °C

Exit Velocity, V_1 = 44.72(h_0 – h_1)$^{1/2}$
= 44.72(4640-4011)$^{1/2}$
= 1121.57 m/s

Task 7: Determination of the Exit Area

$A_1 = \frac{\dot{m}v_1}{V_1} = \frac{300\ x\ 1.5511}{1121.57}$

$= 0.4149\ m^2$

Task 8: Determination of the Quality of Steam at the Exit Plane

$h_1 = h_{f_1} + x_1(h_{fg_1})$

$4011 = 561.43 + x_1(2163.5)$

$x_1 = 1.5944$

Task 9: Determination of the Diameter of the Nozzle at the Exit for a Set of 5 Nozzles

The exit area was determined in *Task 7.* A correlation could be established between this function and the diameter of the nozzle, thus;

$A_1 = \frac{\pi d_1^2 n}{4}$

$d_1^2 = \frac{4A_1}{\pi n}$

$d_1 = (\frac{4A_1}{\pi n})^{1/2}$

$d_1 = (\frac{4\ x\ 0.4149}{3.142\ x\ 5})^{1/2}$

$= 0.32511\ m^2$

Task 10: Determination of the Type of the Nozzle

The nozzle is typically convergent-divergent in nature; since there is a decrease in the value of pressure passing through the duct.

The condition or quality of wet vapour is most frequently defined by its dryness fraction; this is known as well as the pressure or temperature, then the state of the wet vapour is fully defined. Dryness fraction x, equals the mass of dry vapour in 1 Kg of the mixture. Sometimes, a wetness fraction is define as the mass of liquid in 1 Kg of the mixture, i.e. wetness fraction = 1 – x. Note that for a dry saturated vapour, x = 1, and that for a saturated liquid, x = 0. In our determination of quality of steam in *Task 8,* $x > 1$, indicating the superheated state of the fluid. Moreover, in the above analysis, the minimum area of the nozzle is taken to be equal to the area of the throat. Besides, it was assumed that the cross-section of the nozzle at the exit is circular or cylindrical.

CONCLUSION

A nozzle is a device that increases the velocity of a fluid at the expense of pressure. A diffuser is a device that increases the pressure of a fluid by slowing it down. That is, nozzles and diffusers perform opposite task; but a combination of the two as we have in a converging-diverging nozzle, give a different result as we have in the analysis above. The cross-sectional area of a nozzle decreases in the flow direction for subsonic flows, and increases, for supersonic flow. The amount of mass flowing through a cross section per unit time is called the *mass flow rate*, and is denoted by ṁ. The dot over the symbol is used to indicate *time rate of change.* The volume of the fluid flowing through a cross section per unit time is called the *volume flow rate*. The mass flow rate is a function of density and volume flow rate.

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