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Design of Air Storage Tank and Run Time Calculation for Supersonic Blow Down Wind Tunnel

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Abstract: ***The probed analysis report pertaining to the effectual design of the air storage tank, together with experimental findings incorporated into the piping element and pipe bends, brings to light concerning the efficient theory of thick –walled cylindrical vessels, with hemi-spherical heads, on meticulous consideration of parameters ranging from mechanical properties to economic feasibility. Mild steel was identified to be the apt material for the construction of the tank and together by aiding in product design tasks.***

Moreover, electroplating and passivation processes were carried out for providing strength to the selected material. Passing ahead of the limitations imposed by this design, calculated results end up by satisfying the boundary conditions. Finally, after estimating the suitable material for pipes and bends, the product design triumphantly surges ahead to be wholesome in efficiency.

Keywords: Supersonic wind tunnel, runtime, blow down, storage tank and piping

I. DELIBERATION OF AIR STORAGE

Tank Design

Owing to the observable credence of tank potency, on the evident parameters comprising the rudimentary shape of the storage tank ,the radical material properties and the ultimate optimization of product design, the volitional prototype of the proposed design has been presented.

Selection of Parameters for Storage tank

Basically, a pressure vessel can be constructed in any shape, but hazardous consequences occur during the process of manufacturing and testing unconventionally shaped cylinders .The typical morphology of pressure vessels inherently comprises of spherical, cylindrical or conical shapes. Theoretically, a spherical pressure vessel would be the most preferred type, but owing to manufacturing inconveniences and the requirement for larger diameters, it additionally becomes more impractical, thereby eliminating the construction of such a type. But, the counteraction is acquired with cylindrically shaped vessels being the common design, with end caps called heads which are either hemispherical or dished (torsispherical).

Predominantly, the wall thickness factor is estimated to bear values lesser than one-tenth of the inner radius $t_w=(1/10)(r_i)$ in case of thin walled cylinders, contrary to that of thick walled cylinders, wherein the output value is greater than or equal to one –tenth of the inner radius($t_w=1/10(ri)$. Additionally, the radial shear stress (σr)is essentially taken into account ,with regard to thick walled cylinders, but invariably neglected in the case of thin walled cylinders. Moreover σh is assumed to be uniformly distributed throughout the wall thickness in the case of thin walled cylinders, but parabolic variation of σh is vindicated for a thick walled cylinder. But the thin walled pressure vessels, inspite of being statically determinate together with the analytical treatment of stresses being simple, allegedly gives approximate result values. Interestingly, even though being statically indeterminate and the analytical treatment involving complex functionalities, thick walled cylinders yield accurate stress values. On intricate analysis of thick walled cylinders in forms of tyres and gas storage tanks together with thick walled ones such as gun barrels and high pressure vessels, the culmination of design output is rendered to be the selective choice of **thick – walled cylinders with hemi-spherical** ends**.**

Material Selection

As material properties amount to a great deal of vitality in driving the functional resoluteness of cylinders, diverse mechanical and chemical parameters including economical feasibility are cautiously accounted for, while identifying the better material. Property-wise classification of subjected materials i.e., **mild steel** and **stainless steel** has been carried out in order to staunchly assert the selection of the superlative material (**mild steel).**

Comparative Consolidation Of Material Properties: (Mild Steel Vs Stainless Steel)

The densities of mild steel and that of stainless steel are more –or less of the same intensity, but on intricate observation mild steel has greater density (i.e.,7.8-8g/cc).On the basis of ultimate strengths between the materials, stainless steel fares far more well with 655Mpa, while a meagre 420 Mpa is observed for mild steel. Unfortunately, stainless steel amounts to a value of 550 Mpa which is far more acceptable in terms of yield strength. But the elongation factor of the two materials remain the same i.e, 15%.

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On ubiquitous consideration of chemical properties, both the materials are basically non –corrosive in nature. But stainless steel is corrosion –resistant, as the formation of Cr_2O_3, as a result of the interaction of chromium present in the material with oxygen ,makes it corrosion resistant ,thus avoiding rust –formation.

On the other hand, mild steel is liable to be deteriorated by oxidation or chemical action. But finally, the turn is made for mild steel, because it is inexpensive, thus barring out the need for stainless steel to be used.

So, there arises a need for **zinc-plating or chromium plating** to improve its strength and **passivation methods** to protect the coatings. In this case passivation is carried out initially, followed by zinc-plating.

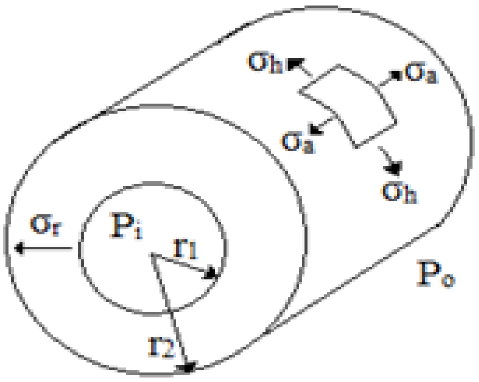

Fig. 1: Pictorial representation of thick –walled cylinder

The given illustrative figure denotes the **outward** and the **inward pressure forces** acting on the **thick-walled cylinder.**

Boundary Conditions

Yet, restrictions are prevalent on account of the inner diameter or the thickness of the wall being less than 15mm,and with the height being greater than the width, thus reducing the total spatial area inside the cylinder. The chosen boundary conditions are

1. Pressure = 3 MPa
2. Volume = $10m^3$
3. Ultimate strength of mild steel=420Mpa
4. Factor of safety of mild steel =5

A. Parameter design of Storage tank

Working stress was computed using material properties of mild steel and it is 20MPa.

By assuming thickness t_w is = 127mm[1],

Diameter of storage tank was found using the following equation.[4]

$$\sigma_{working} = Pi(2\, r_i^2+t^2+2tr_i)/(t^2+2tr_i) \qquad (1)$$

Now inner diameter becomes 1.557m.

Di/ t_w =12.26 and the boundary condition is also satisfied.

Next, height of the storage tank was finalized by fixing the value of Volume of the cylinder is$10m^3$ which results the height of the storage tank h is 5. 252m. Now diameter of container exceeds height of it.

Redesign of sizing was insisted on occupying less space for optimal design. It was met by fixing t_w = 17mm. The new Inner diameter value is 0.227m which satisfies the requirement economically.

Validation of Sizing

Subjection of a thick walled cylinder, to internal or external pressures produces a hoop and longitudinal stress in the wall.

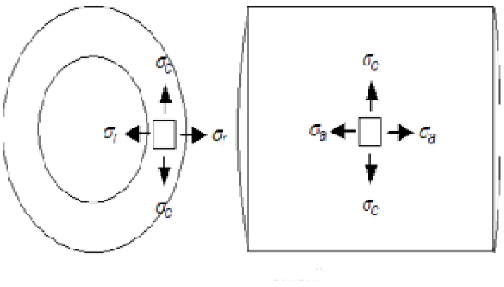

Fig. 2: Stresses in a thick cylinder

The stress acting in the **axial direction** at a point in the tube or the cylinder wall can be expressed as:

$$\sigma_a = (p_i\, r_i^2 - p_o\, r_o^2)/(r_o^2 - r_i^2) \qquad (2)$$

Where, σ_a = stress in axial direction (Mpa), p_i = internal pressure in the tube or cylinder =3MPa,p_o = external pressure in the tube or cylinder =0.101Mpa,r_i = internal radius of tube or cylinder =0. 7785m,r_o = external radius of tube or cylinder = 0. 9055m. The value of axial stress σ_a using the above equation is 8.1131Mpa

The stress in the circumferential direction at a point in the tube or the cylinder wall can be expressed as:

$$\sigma_c = [(p_i\, r_i^2 - p_o\, r_o^2) / (r_o^2 - r_i^2)] - [r_i^2\, r_o^2\, (p_o - p_i) / r^2\, (r_o^2 - r_i^2)] \qquad (3)$$

Where, σ_c is stress in circumferential direction (MPa) , r is radius to point in tube or cylinder wall maximum stress when $r = r_i$ (inside pipe or cylinder).

The circumferential stress value was σ_c=19.23Mpa.

The stress in tangential direction at a point in the tube or cylinder wall can be expressed as:

$$\sigma_r = [(p_i\, r_i^2 - p_o\, r_o^2) / (r_o^2 - r_i^2)] + [r_i^2\, r_o^2\, (p_o - p_i) / r^2\, (r_o^2 - r_i^2)] \qquad (4)$$

Radial stress value becomes -3Mpa

II. PIPING DESIGN

Pipes which are principally required for conducting fluid mediums such as liquids and gases, must be competent enough to ensure the successful operation of the storage tanks. Therefore, divergent aspects have been taken into notice, so as to provide the eventual pipe design.

On the basis of variant pipe –properties, a precise account of the product design process has been summarized below:

Process of Manufacturing

Steel tubes are commonly manufactured by four processes which include:

1. Hot-finished Seamless(HFS)

Hot finished pipes and seamless tubea are firstly produced by heating a solid bloom and then consequently it, in order to obtain a hollow from it. Finally after passing through multiple stands, the hollow is eventually dimensioned by a tool.

2. **Electric Resistance Welded (ERW)**

This process is based purely on the ohmic heating of different parts, in which the culmination of the heat produced results in the fusion of the metal.

3. **High Frequency Induction Welded (HFEW)**

Induction heating is the process of heating electrically conducting objects (usually a metals) by electromagnetic induction, through the heat generated in the object by eddy currents.

4. **Hot –finished Welded(HFW)**

Manufacturing Methods

Moreover the methods involved in the manufacture of pipes also plays a pivotal role in industrial design processes; and they are categorized into Seamless pipes and Welded Pipes. Seamlesss pipe, also known as CDS PIPES (COLD DRAWN SEAMLESS PIPES) finds an aggravated use in the design of parts such as **SHAFTS**, BEARINGS, CYLINDERS with 5175 Bar Tensile Strength. Elementally, being a solid carbon steel it is manufactured by pushing it into the mandrel for procuring different sizes, and it also withstands pressure better than the other types. Welded Steel, (also Electric Resistance Welded (ERW) or Electric Fusion Welded (EFW)) pipe, which is cheaper and possessing tighter dimensional tolerances than seamless pipes is formed by rolling the plate and welding the seam, with the weld flash being removable from the inner or the outer surfaces by means of a scarfing blade. Basically, this process is purely based on the ohmic heating of different parts,as a result of which heat is produced ,and which in turn causes the fusion of the metal.

Classification On The Basis Of Ends Of The Steel Pipe

Even though, there is a strong prevalence of a variety of pipes based on their steel ends, the major classification goes in this manner. They are the following:

1. Socket End Pipe
2. Screwed End Pipe
3. Plain Ended Pipe

Socket End Pipe consists of an expansion at one end of the pipe to receive the end of a connecting pipe.

Pipes with threaded connections are generally regarded as **screwed pipes.** Moreover the underlying advantage lies in the fact that screwed fittings without the need for weldior other permanent means of attachment. A coupling for **plain ended pipe** is particularly adapted as it provides a high ease of construction together with furnishing the ability to join and seal the pipe and the fittings of a given nominal diameter over the full range of their tolerances with excellent rigidity.

By careful consideration of all the above factors, **electric resistance welded steel tube with plain ends** was garniered to be the most appropriate design for the efficacious construction of the air storage tank.

Piping Design

The proposed piping design takes into account of the characteristics of the materials chosen and also on the final product design.[2]

Mild steel is invariably considered as the eligible material for the piping design since chromium or zinc plating followed by passivation improves the hardness characteristics of the mild steel.

Saliently, it is relevant that the quantity of flow of the medium conducted through the pipe, has to be essentially large instead of being limited. And this certainly is of paramount vitality attached to it. The pressure gradient generated by the reciprocating compressor should be uniformly distributed at all points of the tank, since this enormously increases the ease of proper storage. Thereafter the friction observed in the pipe should not in any case, be detrimental to the competence of the tank. Additionally, surface roughness, which in fact is a measure of finely spaced surface irregularities noticed in the pipe, is also a contributing factor. Cardinally, the limit for allowable velocity should also be agreed upon. Universally, the eddies produced at the pipe fittings, the pipe bends, and the pipe line branched have to be nullified, so as to head onto the contemplated working of the pipe.

Limitations

But unfortunately the presence of certain limitations is obvious. In order to counteract those inhibitions, the envisioned design can be deployed by intently taking into account of the following dual criteria ;

Firstly, as the diameter of the pipe is superlative, it unequivocally offers less pressure loss.

Secondly, when the size is tremendous, then it signifies that there is an increase in the capital outlay, in direct proportionality.

By regarding the criteria imposed and by doing the needful, the optimum design can be procured.

Steps Involved in Pipe Sizing

$$Q=A*V \tag{5}$$

Where, Q represents discharge which is equal to $48m^3/hr$(taken from compressor because of the continuity in the flow), A is pipe area in m^2 ,V is velocity of air through the pipe in m/s^2 . For reciprocating compressor, V=16 to 20m/s for suction line & V=25 to 30m/s for delivery line. By considering V is 25m/s, inside diameter of the tube d is 26.1m. [3]In the process to determine the pipe size, the following methodology is followed. The varying diameters of the pipe are compared with respect to the standardized design, thus signifying the need to take into deliberation, the fact that the proposed model is a thick –walled cylinder. And so in regard with the above proposition, the pressure losses for varying diameter values are estimated.

For example if the internal diameter is taken to 32 mm, and if the pressure losses are a way too high, then the progressive determination can be continued by selecting the succeeding higher standard diameter.

Pressure Loss Calculation

$$P_1-P_2 =\{(fLV^2\Upsilon)/(d*2g) \pm h(\Upsilon-\Upsilon_a)\}^{[3]} \tag{6}$$

Where,P_1&P_2 are absolute pressure at the beginning and the end of the Pipeline respectively, L is length of the pipeline in m, Υ is specific weight in N/m^3,f is pipe friction factor, Υ_a is specific weight of air in N/m^3 which is equal to 12.93 N/m^3,h is height in case of a non-horizontal pipeline.

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For a horizontal pipeline h=0 with boundary conditions, the value of Re is $5.6996*10^4$. Assumed value of surface roughness k is 0.5mm.

By drawing graph between Re and f with d/k is 100,the obtained value of f is 0.038 [3].

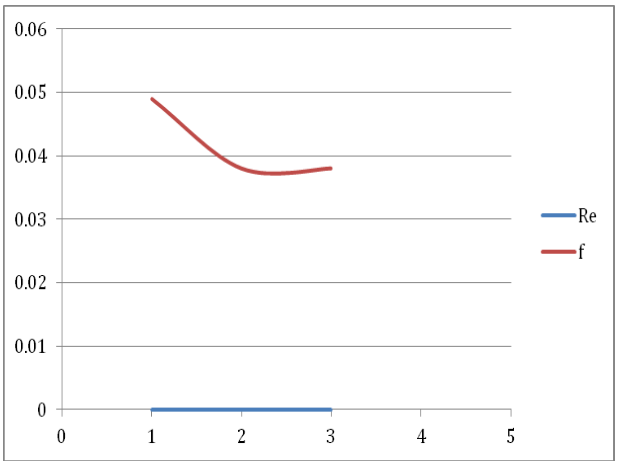

Fig. 3: Graph between Reynolds number and friction factor

The pressure loss (P_1-P_2) was calculated [2] when length of pipe is 4m. The computed value is $1956.47 N/m^2$.

Pipe bends

Bends are required for changing the flow direction of a medium conducted through a pipe. Bends may be connected to the pipeline by site welding or may be bolted to the pipe line if the bends are of flanged type**.**

Ultimately, the deliberated design is **straight pipe** which is **plain- ended**.

Calculation of Wall Thickness

Implementing the thin-cylinder approach for the calculation of wall-thickness is not desirable, due to the small thickness value obtained by this approach. Therefore, by using the thick-cylinder approach, for high-pressure vessels, the wall-thickness is calculated in the following manner as given below:

Using Lame's Theorem,

For high pressure vessels,

The formula for the calculation of the Internal Pressure is given as ,

$$P_r=(B/r^2)-A \qquad (7)$$

Where the pressure value is given as $3*10^6$ MPa

The formula for the calculation of the Maximum Stress is given as

$$f_r=(B/r^2)+A=10.468*10^6 N/m^2 \qquad (8)$$

The values of A and B were calculated to be

$$A=3.734*10^6$$
$$B=1.1468*10^3$$

By taking outside pressure as atmospheric pressure , outer radius R is found to be 0.0173 m.

The wall-thickness of the pipe can be calculated using the formula,

t= R-r =0.00425 m

III. RUN TIME CALCULATION

Run time calculation of supersonic blow down wind tunnel was performed with constant mass flow using the following equation[1] .

$$t=0.0862(V/A^*)(\sqrt{Tt}/Ti)(Pi/Pt)\{1-(Pf/Pi)^{1/n}\} \quad (7)$$

where, V-volume of storage tank in m^3,Pi-initial pressure in N/m^2 ,P_f-final pressure in N/m^2 and

Tt-Total temperature in K, n- polytropic exponent.

The Algorithm for run time calculation of supersonic blowdown wind tunnel is as below

1. Start
2. Get the input value for volume, area , total temperature in Kelvin, initial temperature in Kelvin, atmosphere pressure, Pt value, Pf value, n value.
3. Calculate the following

   n1=1/n;

   t1=v/area;

   t2=pow(Tt,0.5)/Ti;

   t3=Pi/Pt;

   t4=Pf/Pi;
4. Do the runtime calculation of the supersonic blowdown wind tunnel using the following formula $t=0.0862(V/A^*)(\sqrt{Tt}/Ti)(Pi/Pt)\{1-(Pf/Pi)^{1/n}\}$ by applying the n1,t1,t2,t3 and t4 values in the runtime formula t=0.0862*t1*t2*t3*(1-pow(t4,n1));
5. Print the value of t
6. Stop.

Similarly, the time to pump a tank from an initial pressure Pi to a final pressure Pf may be found from tp = Vp (Pf-Pi)/1.013Q (8)

Where, Q-compressor rating in cfm, $48 m^3/hr$

IV. PUMP TIME CALCULATION

The Algorithm of pump time calculation is shown below.

1. Start
2. Declare the following variable Vp as storage tank volume, Pf as final pressure, Pi as Initial pressure,Q as Discharge value,tp as time to pump a tank
3. Get the input value for Vp, Pf, Pi, Q.
4. Calculate the tp value using the following statement tp=(Vp*(Pf-Pi)/(1.013*Q));
5. Display the time to pump a tank (tp)
6. Stop

The following results were obtained for two different values of storage tank volume.

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Table 1: volume of storage tank, V =10m^3

S.No	Mach no	Runtime in seconds	Pump time in hour
1.	1.5	41.28	11.29
2.	2	46.97	11.14
3.	2.5	54.88	10.89
4.	3	52.38	10.26
5.	3.5	51.53	9.42
6.	4	33.68	7.24

Table 2: Volume of storage tank, V =15m3

S.No	Mach no	Runtime in seconds	Pump time in hour
1.	1.5	61.92	16.95
2.	2	70.46	16.72
3.	2.5	82.32	16.35
4.	3	78.57	15.41
5.	3.5	77.29	14.14
6.	4	50.52	10.86

V. CONCLUSION

In the proposition of the optimized design for the storage tank, parametrical guidelines were investigated so as to arrive at the final output. On keen analysis of inferences gleaned from studies pertaining to the shape of the vessel, material properties and the product design parameters, it was noted that thick –walled cylinders made of mild steel are the best. But, due the fact, that mild steel has a few inconsistencies, zinc plating /chromium plating preceeded by passivation techniques were administered to counteract its low-strength.

The problem in availability of less spatial area inside the cylinder was rectified. Moreover, the working stresses, the inner diameter and the height were meticulously calculated. Axial, Circumferential and Radial Stresses were also computed for this study. Furthermore, electric resistance welded steel tube with straight pipe, being plain –ended was chosen.

Moreover, the problem of much pressure loss due to the bigger diameter of the pipe and an increase in the capital outlay, as a result of its larger size were also attended. The internal diameter of the pipe and the pressure losses prevalent in gases were grafted and the friction factor value proved to be amicable too. On careful consideration and incorporation of the above standardizations, the product design of **thick –walled cylindrical vessels with hemispherical ends** emerged to be conclusively successful. Moreover the wall-thickness of the pipe was also estimated by the utilization of the thick cylinder approach. The runtime and time to pump storage tank calculations were done for range of mach numbers from 1.5 to 4.

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