Design Review on Self-Cleaning Basket Strainer (Formula Based)

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Abstract— The demand of continuous uninterrupted flow from the pipelines with debris removal is done efficiently by self-cleaning basket strainer. Clean water or other fluid is basic requirement of an individual. Now a days in thermal power plants, for condensing of steam sewage water is be supplied. This makes us think of designing self-cleaning basket strainer.

Key words: Self-Cleaning Basket Strainer, Strainer, Partition Plate

I. INTRODUCTION

A major issue of natural water resources is water pollution which further affects the health and environment of the people. Waste water from the industrial and domestic areas are the major source of water pollution .Waste water treatment is the best way to prevent the water pollution. Chemical treatment to adjust the level of oxygen and separation of solid impurities are the basic involvement in the waste water treatment. Scope of our research is mainly focused on filtration aspect of waste water treatment. For waste water treatment strainers, filters, purifiers can be used which depends on the type of impurities and debris to be removed. We have decided to design a basket strainer for treatment.

II. PARTS DESCRIPTION

The main feature of self-cleaning basket strainer is its self-cleaning mechanism of filtering element. The filtering element in this case is strainer basket.

The various elements of self-cleaning basket strainer are upper & lower shell, partition plate, top & bottom head, lower and upper supporting plate, strainer basket, nozzle's, backwash assembly, drain assembly, flanges etc. Openings have their functional requirements such as Inlet and Outlet connections; drain pipe connection pressure gauge connection etc.

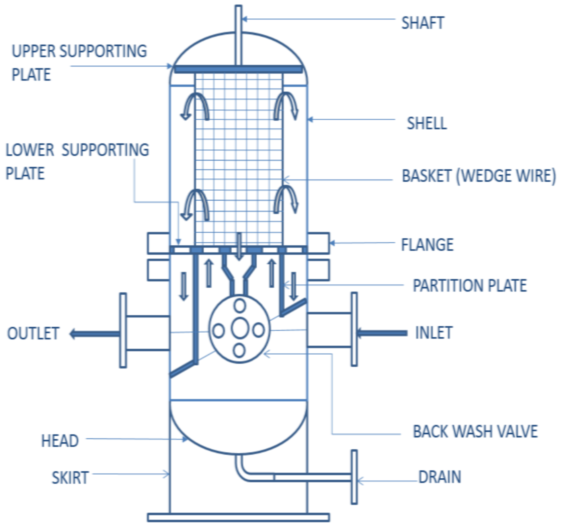

Fig. 1: Self-Cleaning Basket Strainer

III. MATERIAL REQUIREMENT AND METHODS

We have designed a strainer considering Design pressure up to 5 bars. Wherever in the design, t_x = minimum thickness of the x component [suppose] (mm), P_i = internal pressure ($\frac{\text{N}}{\text{mm}^2}$), d_i = internal diameter of shell (mm), σ_{all}=allowable stress ($\frac{\text{N}}{\text{mm}^2}$), η=weld joint efficiency, *c.a.* = corrosion allowance (mm)

A. Design Aspects for Shell:

Pressure to which strainer is going to subject is not very high, so we choose to manufacture thin pressure vessel.

$$t = \frac{P_i \times d_i}{2\sigma_{all} \times \eta - P_i} + c.a. \quad (1.1)$$

For shell, we can use SA 516 Gr. 70 sheets. This sheet is suitable for moderate and low temperature service.

We have divided the shell into two parts to facilitate easy removal of internal parts. We joined flanges to each shell and then bolted them together.

B. Design aspects for Head:

We have preferred semi elliptical head over others. Semi elliptical heads are stronger than Formed plain head and torispherical head. Also even though semi elliptical head is weak we than hemispherical head but forming required for hemispherical head is more and it is costly. Hence to optimize cost and strength, we have selected semi elliptical head.

$$t_h = \frac{K_f \times P_i \times d_i}{2 \times \sigma_{all} \times \eta - 0.2 P_i} + c.a \quad (1.2)$$

$$K_f = \text{Stress intensification factor} = \frac{2 + K_i^2}{6} \quad (1.3)$$

$K_i = a/b$ =2….. (Aspect ratio)

K_f=1.For better rigidity, thickness of head should be more than the calculated one.

Knuckle radius = 0.1 di(1.4)

Straight flange length =3 t_h or 20 mm (whichever is larger)

C. Nozzle and Openings:

Material – SA 106 Gr. B Nozzles used are seamless Carbon Steel pipes, hence Joint efficiency = 1σ_{all}= S_{ut}/ 4The speed of waste water into the vessel should be below 3m/s (IS 2825-1969), therefore consider the appropriate diameter of nozzle

$$t_n = \frac{\text{Pi} \times \text{dn}}{(2\,\sigma_{all} \times \eta - \text{Pi})} + \text{c.a} \quad \text{do} = \text{di} + 2t_n;$$

d_n= inner dia. of nozzle in uncorroded condition(mm); d_{nc}= inner dia. of nozzle in corroded condition(mm); t_s= actual thickness of the shell (mm); t_{rs}=minimum required thickness of the shell (mm); c=corrosion allowance (mm); t_n=thickness of nozzle (mm) H_1= height of effective compensation in nozzle wall outside the vessel (mm); H_2=height of effective compensation in nozzle wall inside the vessel(mm); t_{rn}=min. required thickness of the nozzle wall(mm);

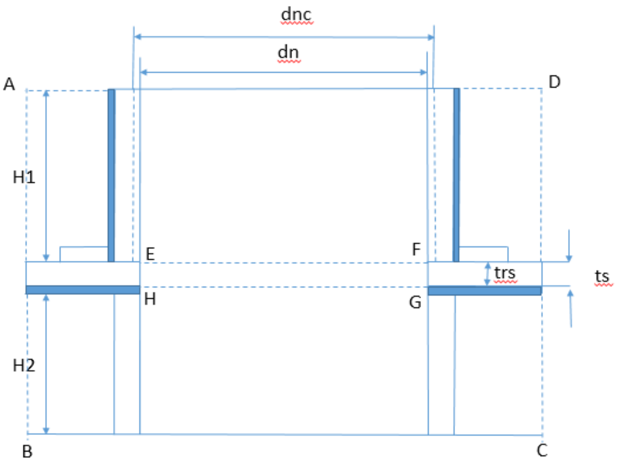

Fig. 2: Area Method of Compensation

Check for Compensation: $H_1 = \sqrt{(d_{nc}\ (t_n - c))}$

$H_2 = \sqrt{(d_{nc}\ (t_n - 2c))}$ or 15 mm ...whichever is smaller.

Area of opening in corroded condition for which compensation is required $A_r = d_{nc} \times t_{rs}$

Area available for compensation

1) Area of excess thickness in portion of vessel shell is
$A_1 = d_{nc}\ (t_s - t_{rs} - c)$
2) Area of excess thickness in portion of nozzle wall outside the vessel shell is ,
$$A_2 = 2H_1(t_n - t_{rn} - c)$$
3) Area of thickness of nozzle wall inside vessel shell is $A_3 = 2H_2(t_n - 2c)$

Total area available (A_a) for compensation is,

$$A_a = A_1 + A_2 + A_3$$

If $A_a > A_r$, compensation is adequate and no reinforcing pad is required otherwise padding required

D. Backwash Nozzle:

Backwash nozzle's diameter is 3" NPS Standard.

$$t_b = \frac{\mathrm{Pi} \times \mathrm{di}}{2\sigma_{\mathrm{all}} \times \eta - \mathrm{Pi}} + \mathrm{c.a.}$$

$$d_o = d_i + 2t_b$$

Also due to inertia, some of heavier particles will settle down at the bottom of the vessel. So to remove it, drain pipe is provided at the bottom.

E. Straining Element Selection:

Screen opening should be selected carefully for proper filtration. Screen opening should be selected on the basis of necessary protection required. If we select smaller opening than required then more debris will be retained and subsequently resulted into larger cleaning duration and increases backwash fluid loss. Also pressure loss is also increases.

We have selected strainer size as 500 OD X 1050 mm. For strainer of 20 (suppose) mesh size, open area is 40%.Net straining Area = 0.4×Gross straining Area

$$\text{Open Area Ratio} = \frac{\text{net straining area}}{\text{pipe inlet area}} \qquad (1.5)$$

Basket strainer should be corrosion resistant and it should be

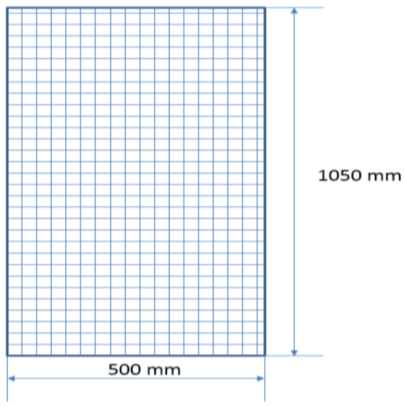

Fig. 3: Strainer

Able to withstand pressure also. So according to the requirements, SS316 is selected. SS316 is better corrosion resistant than SS304 which is widely used comparatively cheaper material.

F. Partition Plate:

$$Volume_A = ((\tfrac{\pi}{4} d_p{}^2) \times \mathrm{L}) + \text{volume of head} \qquad (1.6)$$

$$Volume_B = ((\tfrac{\pi}{4})(d_i{}^2 - d_p{}^2)) \times \mathrm{L} \qquad (1.7)$$

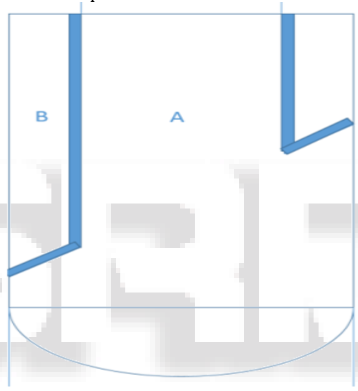

Fig. 4: Partition Plate

Hence we have seen that volume of lower portion of vessel inside of partition plate is equal to that outside of partition plate. Thickness of partition plate(t_p) is given by,

$$t_p = \frac{\mathrm{Pi} \times \mathrm{dp}}{(2\,\sigma_{\mathrm{all}} \times \eta) - \mathrm{Pi}} + \mathrm{c.a.}$$

To have necessary rigidity so that it can support other loads the thickness should be more than calculated one

G. Flanges:

For design of nonstandard flange considering design pressure is 5 bar (for ex.) i.e. 72.5 psi so flange of 125# rating is suitable for it.

$$\frac{G_o}{G_i} = \sqrt{(\sigma_g - \mathrm{m\ x\ Pi})/(\sigma_g - (\mathrm{m}+1)\mathrm{Pi})} \qquad (1.8)$$

Where σ_g = gasket seating stress; G_o=Outer dia. of gasket, (mm), G_i=inner dia. of gasket (mm);

$$\text{m= gasket factor} = \frac{\text{residual gasket stress}}{\text{internal pressure}} = \frac{\mathrm{Fgi} - \mathrm{Fp}}{\mathrm{Fp}} \qquad (1.9)$$

Gasket material – Stenotherm-SSTC

H. Design of Bolts:

$$\text{Preload on bolts:} W_b = \pi G b \sigma_g \qquad (1.10)$$

G= mean dia. of gasket, mm

Load under operating condition:

$$W_{b2} = (\pi/4)G^2 P_i + 2\pi b G m P_i \qquad (1.11)$$

Diameter of bolts& number of bolts:

The no. of bolts which should be multiple of four is given by an empirical relation,

$$N \cong \frac{G}{25}(mm) \quad (1.12)$$

$A_c = \frac{Wb_1}{\sigma_{b1} N}$ or $\frac{Wb_2}{\sigma_{b2} N}$Where σ_{b1}=allowable tensile stress for the bolt at atm. temperature (N/mm^2); σ_{b2} = allowable tensile stress for the bolt at operating temperature (N/mm^2)

Allowable Tensile stress of hot rolled carbon steel (at atmospheric condition) and allowable tensile stress under operating condition …….(100° C)

Bolting material selection: (Appendix 1, Joshi Mahajan)

$$Ac = \frac{\pi}{4} {d_c}^2$$

Nominal diameter (d_b)is given by, d_b = *1.19* d_cDiameter of bolt pitch circle: -*D* = *G* + *2*d_b+ *12*Outside diameter of flange; -D_o = *D* + *2*d_bDesign of flange: W_b = W_{b2}… (As it is maximum) Radial distance between the gasket load reaction and bolt pitch circle

$$(h_g) = \frac{D-G}{2}(mm) \quad (1.13)$$

$\therefore$ hydrostatic pressure force $(F_p) = \frac{\pi}{4} G^2 P_i$ (1.14) K = 0.3

$$+ \frac{1.5\, W_b h_g}{F_p\ G} \quad (1.15)$$

$\therefore$The thickness of flange (t_f) is given by,$t_f = G\sqrt{\frac{K \times P_i}{\sigma_{all}}} + C$

(1.16)

Material used for flange is SA105S_{ut} = 70 ksi and S_{yt} = 36 ksiσ_{all} =$\frac{S_{ut}}{4}$Design of flanged joint:Standard steel flanges (IS 4864-4867) for shell sections, heads & pipes are selected according to pressure or temperature requirements. So no design is required. We can select appropriate flanges according to the pressure ratings. But where standard flanges cannot be used due to unavailability. Non standard flange for shell is to be designed.

I. *Design of Skirt:*

Skirt is best suitable support for pressure vessel. Because out of all supports, skirt has largest sectional modulus& economical for manufacturing.

Skirt may be welded directly to the bottom head, flush with shell or to the outside of the shell. We are not going to weld it outside of shell, as weld joint will be in shear. Hence weld joint is welded flush to the shell.

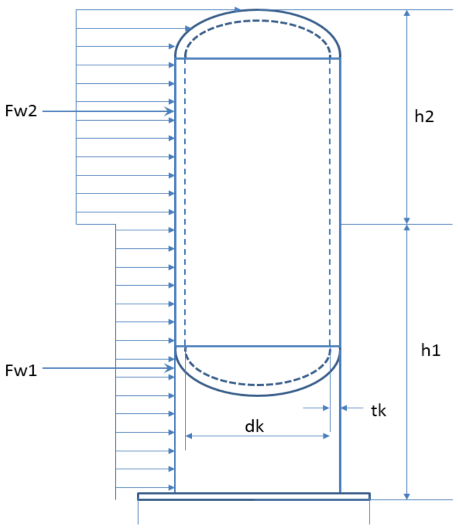

Fig. 5: Wind Load

Stresses induced in skirt support: 1.Direct compressive stress 2. Bending stress$\sigma_c = \frac{Wt}{\pi(dik+tk)tk}$\$\sigma_b$ = $\frac{M}{Z}$Whered_{ok} (outer diameter of skirt); t_k (Thickness of skirt)d_{ik} =d_{ok} -2t_k $\therefore Z = \frac{\pi \times (d_{ik}+2t_k)^2 \times t_k}{4}$;

Where Z=sectional modulus Maximum Bending Moment (M_w) Due to Wind Load:

$$M_w = F_{w1}\frac{h1}{2} + F_{w2}(h_1 + 0.5h_2) \quad (1.17)$$

Where F_{w1}=force due to wind load acting on the lower part of vessel shell, (N);F_{w2} =force due to wind load acting on the upper part of vessel shell (N);P_1=wind pressure for lower shell ($\frac{N}{mm^2}$)P_2=wind pressure for upper shell ($\frac{N}{mm^2}$)

F_{w1} =$P_1 h_1 d_0 F_{w2}$ = $P_2 h_2 d_0$According to I.S. 875-1987 Part 3-Wind loadsV_b is basic wind velocity from wind speed map of India Design wind speed is given by,V_w =$K_1 K_2 K_3 \times V_b$ (1.18)Where K_1= Risk factor; K_2= Terrain height & structure size factor;K_3 = Topography factor from basic wind speed map of India, wind speed in western Maharashtra is 39m/s. From table in IS-875,K_1 = 0.92(for life of 25 years);K_2 = 0.91(for class A & terrain category-3); K_3 = 1(as effect of topography is not significant.)

1) *Maximum Bending Moment Due to Seismic Load:*

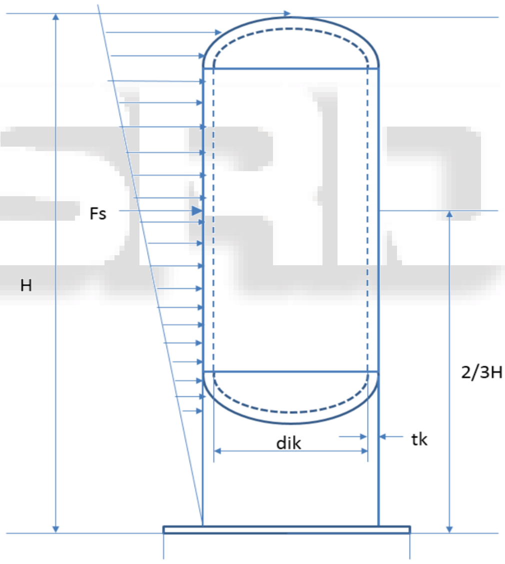

Fig. 6: Seismic Load

Seismic load varies linearly from maximum at top to zero at the bottom. Resulting seismic load is considered to acting at a distance $\frac{2}{3}$ H from the base and it is given by,

$$M_s = \frac{2}{3} C W_t H \quad (1.19)$$

Hence as M_s>M_w then maximum bending stress generateddue to seismic load is given by,

$$\therefore \sigma_b = \frac{M_s}{Z} \quad (1.20)$$

Now direct compressive stress is given by,

$$\sigma_c = \frac{W_t}{\pi(d_{ik}+t_k)t_k} \quad (1.21)$$

Maximum compressive stress generated in skirt is, $\sigma_{ck} = \sigma_b + \sigma_c \sigma_{ck}$value will state the permissible stress of material which will decide the design of skirt to be safe or not.

J. Skirt Bearing Plate:

Assume *bolt circle diameter* = *Skirt diameter* + *150*Compressive stress (σ_c) between the bearing plate and concrete foundation is given by,$\sigma_c = \frac{W}{A} + \frac{M}{Z}$Maximum compressive stress is less than the permissible compressive stress. Thickness of the bearing plate is determined by considering it as a uniformly loaded cantilever with $\sigma_c$as uniform stress.Maximum bending moment for cantilever occurs at the junction of skirt and bearing plate.$M_{max} = \frac{\sigma_c \times b \times l^2}{2}$Stress is given by, $\sigma = \frac{6 \times M_{max}}{tb^2}$so minimum thickness of bearing should be more from safety point of view.

K. Anchor Bolts:

Minimum Weight of empty vessel = W_{min}

$$\sigma_c = \frac{W_{min}}{A} - \frac{M}{Z} \qquad (1.22)$$

Since $\sigma_c$is negative, the vessel skirt must be anchored to the foundation by anchor bolts to prevent overturning. Hence load on each bolt = $\sigma_c \times \frac{A}{n}$ where n =no. of bolt For that we select a bolt of appropriate diameter and then find area and proceed for further calculations to determine safety.

IV. ACKNOWLEDGMENT

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V. CONCLUSION AND FUTURE SCOPE

Thus we have design a strainer and further it would be interesting to see the behavior of static structural component in fluctuating cases and conditions of various degree of filtration with different variety of debris and with what extent they harm the vessel

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