DESIGN OF MUFFLER FOR A LOW POWERED GAS TURBINE EXHAUST SYSTEM

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ABSTRACT

The report contains the design of the muffler to be installed in the stack of chimney for low powered 10 megawatt gas turbine. The parameters pressure and temperature of the exhaust gas are to be gradually reduced in three successive modules with the energy considerations. The design is accomplished for a range of possible magnitudes of friction coefficients for the gas expanding through perforations. Besides, based on the results of computations drawings are provided with the aid of auto cad for manufacturing and fabrication purposes.

1. INTRODUCTION

As combustion turbine power generation facilities are sited closer to residential areas and the market for power generation equipment becomes increasingly competitive, noise control and silencer manufacturing are continually challenged to develop more effective and less expensive silencing treatments. The design of the exhaust stack silencer is of particular significance since it is one of the most costly noise control components of the combustion turbine packaging. The exhaust stack silencer must be designed to attenuate a broad frequency range associated with the combustion turbines exhaust, including high frequency turbine tones as well as the low frequency combustion roar and flow related noise components' [**1**].In a gas turbine after expansion of the gases in the turbine, the gas passes through the transition and into the plenum of the exhaust stack. The exhaust gas exiting the stack generates pressure waves that significantly exceed atmospheric pressure. These pressure waves exit the exhaust stack at a very high velocity and hit atmospheric conditions, which produces an offensive exhaust boom. To offset this noise, resonance chambers (a.k.a. silencers) are used inside the exhaust stack. The chambers are relatively large areas that allow the exhaust gases to vibrate with the gases that are already present in the chamber. With this spring-mass vibration system, the gasses generate sound at the same frequency but in opposite phase to each other, which negates the noise. Since offsetting frequency is such a small range to hit, silencers also include other features to produce friction that also helps decrease noise. The size of the chambers is a function of the power developed by the turbine and its speed. The three main types of silencers are reactive (reflective), absorptive (dissipative), and combination reactive/absorptive. Combination silencers perform better over a wider range of frequency levels than a reactive or absorptive silencer alone. In dampers of the resistive type energy is dissipated by imposing additional resistance to the flow: an attraction of this type is its insensitivity to pulsation frequency, but the pressure loss may be excessive. Reactive dampers can achieve satisfactory attenuation at particular frequencies and the design of these dampers (resonators) is usually based on acoustic theory or accomplished by a trial and error procedure. The non-reactive type does not have the resonant characteristics of the reactive type and hence its efficiency is not a function of frequency. In the absorption version of the nonreactive type energy is absorbed in a porous lining fitted to a length of the pipe wall. In the snubber version the high static pressure generated by the pulsations drives gas through perforations in the pipe wall energy is thereby dissipated and the amplitude of pressure pulsations in the pipe is reduced.

In the present work a three stage multiple tube silencer is designed for gas turbine exhaust system for the operating conditions like pressure 3-4 bar, temperature 300^0C noise level in the range of 85db.The silencer consists of 3 modules fitted with pipes of different diameters .The top end of each pipe is closed while the bottom end is opened for the gases to diffuse through the pipe from the perforations provided along the length of the pipe. Sound attenuation is achieved by providing number of perforations along the pipe .The mechanics of fluid flow in the perforated tube is predicted by using energy equation at inlet and outlet of the gas flow .Pressure loss inside the pipe is assumed to be smaller when compared to the perforations along the pipe. Loss of pressure in the entry is expressed as a fraction 'α' of inlet pressure P_1 such that $P_L = \alpha P_1$ .The major parameters in the design of the silencer are the global porosity 'φ' which is defined as the ratio of the total area of the holes in the pipe wall to the pipe cross sectional area and the non dimensional loss coefficient 'P_L'. The pressure drop in each stage is expressed a function of number of pipes 'Z', number of perforations 'ZP', diameter of the pipe 'D', diameter of each perforation 'D_1' and length of each pipe 'Z_1' .In this model the mean flow velocity is assumed to be uniform in all the pipes of each stage .The Mach number of the gases increases in the first module when compared at inlet indicating considerable major transmission loss in the first stage of the silencer. The pressure loss in first stage is restricted to 2.5bar from 3.5bar at inlet by carefully controlling the above parameters. After number of iterations the solution converges which gives the optimized design parameters as number of perforations, diameter of perforations, diameter of pipe etc.Low frequency sound attenuation is achieved by providing mineral wool insulation at the base of each pipe up to a height of 0.5mts along the length of each pipe. In the second module the Mach number of gases is reduced by decreasing the number of pipes, length of the pipe and increasing the diameter of pipe, number of perforations and diameter of the perforations. The pressure in the second module is restricted to 1.4bar from 2.5 bar in the first module and the Mach number of the gases in the second module is reduced to one half of the first module .After number of iterations the solution converges which gives the optimized design parameters of the silencer in the second module similar to the first module. Insulation is also provided for the pipes in the second module up to a height of 0.25mt similar to the first module for low frequency sound attenuation. The gases come out of the second module and enter into the third module which consists of single large pipe of diameter 2.0mts opened at both the ends without any perforations. This pipe acts as annulus pipe for the chimney stack and there is no pressure drop in the annulus pipe however the Mach number of the gases coming out of the annulus pipe is drastically reduced to the acceptable levels of sound emissions as per ISO standards to a tune of 7mts/sec.

2. REVIEW OF LITERATURE

Perforated pipes for gases flow in the exhaust silencer are used in the gas chimney to reduce noise by providing uniform flow. The amount of fluid discharging from any opening in a pipe is dependent on the discharge coefficient and cross sectional area of the pipe. However smaller pipes must be used so that the static pressure may vary. Keller **[2]** predicted the variation of velocity along uniform perforated pipes assuming constant friction factor. The variation of cross sectional area of a pipe necessary to provide uniform outflow. Variation of friction factor for both laminar and turbulent flow in pipes was studied by Dow [**3**].Theoretical approach commonly used involves differential equations whose coefficients include the friction factor, diffusion and inertia. The behavior of friction factor in pipe flow is well established. B.J.Bailey **[4]** calculated discharge coefficient for perforated pipes for gas flow assuming uniform gap between the perforations with a maximum length –to-diameter ratio of 250 containing pairs of diametrically opposite holes. Multiple perforations in the pipe were considered by Rudinger [**5**] who derived the equations for unsteady gas flow in a pipe with mass removal. G.H.Trenghouse **[6]** measured discharge coefficient values for air flow through a single row radial holes in the wall of a pipe line .He also measured the values of pipe Mach numbers in the immediate vicinity of the holes. A wide range of pressure and area ratios were considered for the flow in both the directions of the holes. He found that the discharge coefficients are not being affected by the temperatures. J. Wang studied high energy fluids in perforated pipe distributors. An analytical study is made of the perforated pipe distributor for the admission of high-energy fluids to a surface steam condenser. He showed that for all perforated pipes there is a general characteristic parameter $M(kD/Lf)$, which depends on the pipe geometry and flow properties. Four cases are considered based on the value of the characteristic parameter M. (1) when M1/4, momentum controls and the main channel static pressure will increase in the direction of the streamline. (2) When 1/6M<1/4, the momentum effect balances friction losses and the pressure will decrease to a minimum, and then increase in the direction of flow to a positive value. (3) When 0<M<1/6, friction controls and the pressure will decrease to a minimum, then increase slowly, but the total pipe static pressure difference will always be negative. (4) When $M = 0$, a limiting case when the ratio of the length to the diameter is infinite. This analysis is useful not only for the design of perforated pipe distributors over a wide range of dimensions, fluid properties, and side hole pressure but also for many other technical systems requiring branching flow distribution.

3. FORMULATION OF THE MODEL

The thermodynamic conditions of the gas at inlet and outlet locations of perforations of the tube are as follows

P_1, P_2 - pressure of the gas at the inlet to the tube and exit of perforations N/m^2

ρ_1 ρ_2 Density of the gas at the inlet and exit

V_1, V_2 Velocity of the gas at the inlet and outlet to the tube

T_1, T_2 Temperature of the gas at locations 1 and 2

Applying energy equation between points 1 and 2 (see fig. 1) it follows

$$P_1 + \frac{1}{2}\rho_1 V_1^2 = P_2 + \frac{1}{2}[1+K]\rho_2 V_2^2 \quad (1)$$

Where G is the total discharge rate of the gas

$$V_1 = \frac{G}{\rho_1\, Z\, A} = \frac{G\, R\, T_1}{P_1\, Z\, A} \quad (2)$$

$$V_2 = \frac{G\, R\, T_2}{Z\, P_2\, A_P\, Z_1} \quad (3)$$

and

$$\frac{\rho_1}{\rho_2} = \left[\frac{P_1}{P_2}\right]^{\frac{1}{n}}$$

K is the friction coefficient and it depends on the nature of the orifice or perforation created on the tube. It is shown ( ) that the value K varies between 0.5 - 1.2.

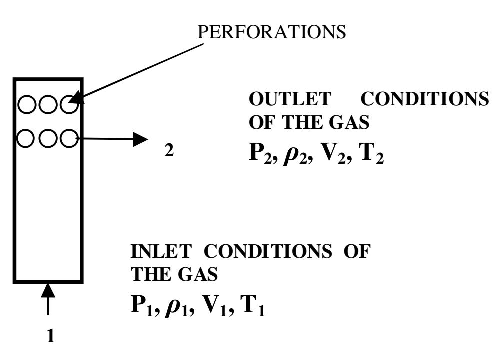

Figure 1 : Schematic of the perforated tube

Further assuming the gas follows the general law of expansion through the perforations: $\frac{P}{\rho^n} = Constant$, the exponent for the gases can be between 1.2 – 1.3.

Equation (1) can be manipulated to the form shown as follows

$$\frac{P_2}{P_1} = 1 + \frac{1}{2}\frac{\rho_1 V_1^2}{P_1}\left[1 - (1+K)\frac{\rho_2}{\rho_1}\left(\frac{V_2}{V_1}\right)^2\right] \quad (3a)$$

Substituting

$\left[\frac{V_2}{V_1}\right]^2 = \left(\frac{T_2\, P_1\, A}{Z_1\, A_P\, P_2\, T_1}\right)^2$ in Eq.(3) yields

$$\frac{P_2}{P_1} = 1 + \frac{1}{2}\left(\frac{G}{Z\,A}\right)^2\left(\frac{R\,T_1}{P_1^2}\right)\left[1 - (1+K)\left(\frac{P_2}{P_1}\right)^{\left(-\frac{1}{n}\right)}\left(\frac{A}{A_P}\right)^2\left(\frac{1}{Z_1}\right)^2\right] \quad (4)$$

Or

$$\frac{P_2}{P_1} = 1 + \frac{1}{2}\left(\frac{G}{Z\,A}\right)^2\left(\frac{R\,T_1}{P_1^2}\right)\left[1 - (1+K)\left(\frac{P_2}{P_1}\right)^{\left(-\frac{1}{n}\right)}\left(\frac{D}{D_P}\right)^4\left(\frac{1}{Z_1}\right)^2\right] \quad (5)$$

where

A-Area of cross section of the primary tube,m^2

G- Discharge rate [kg/s]

R-gas constant is 287 J/kg/K

$$\frac{D}{D_P} = \frac{Diameter\ of\ the\ tube}{Diameter\ of\ the\ perforations}$$

Z- Number of Primary tubes inter connecting the modules

Z_P - Number of perforations on each tube

Equation (3a) is the primary tool in the design of the muffler to harness and control the noise through the thermodynamic parameters. Primarily the design criteria should be so chosen that the exit pressure and temperature should be in the limits of prescribed norms for the given height of the chimney stack. The conceptual design of the muffler for further evaluation of the salient parameters is provided schematically in Fig.2.

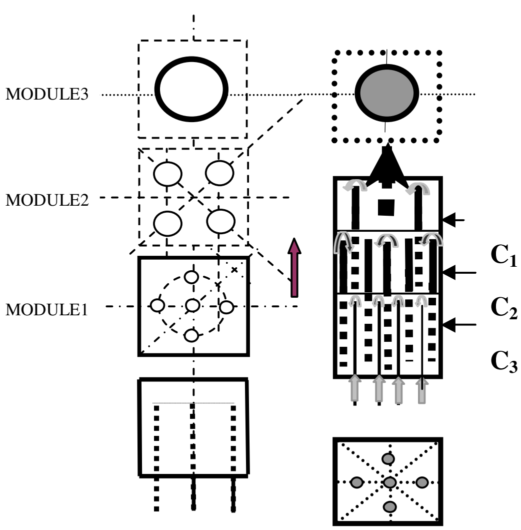

Figure 2: Conceptual positioning of the tubes in each module

(not to scale)

Thus the design of the muffler is undertaken as per the initial input data given below.

The prescribed power rating of the gas turbine is 10 megawatts.

The gas turbine exhausts at a pressure of 3.5 bar and at a temperature of 300°C to the first stage Module 1 through five stainless steel tubes each of 65 cm I.D. The gas further expands into the first module through perforations provided towards the end of the tube according to the law $\left(\frac{P}{\rho^n}\right) = constant$. Subsequently, the gas finds its path into the second module through four stainless tubes each of 80 cm I.D. These, tubes interconnect the first module with the second providing free passage through the perforations provided at the end of tubes of module1. Similarly the second module communicates to the third module. The flow passage of the gas from the second into the third is through the perforations made at the at the end of the tubes. running into the second module. .The third module is a single large tube of 180 cm I.D directly letting out the gas through the exit of the stack. The length of the tubes inter-connecting the modules is limited to 1.25 m.

Evaluation of Discharge rate of exhaust from the turbine:

The gas turbine under consideration is a low power 10 mega watt prime mover.

Amount of fuel to be burnt per second:

$$= \left[\frac{Power}{Efficiency\ \times Calorific\ value\ of\ the\ fuel}\right]$$

$$= \left[\frac{10\times 10^4}{0.65\times 44\times 10^4}\right]\left[\frac{kg}{s}\right]$$

As per design considerations of efficient combustion the amount of air required for complete combustion would be around 60 times the quantity of fuel burnt per second. .

Thus, the quantity of exhaust gases passing through the chimney stack would be = 20.97 $\frac{kg}{s}$

4. NUMERICAL METHOD

Equation (3a) contains the pressure ratio term $\left(\frac{P_2}{P_1}\right)$ both in LHS and RHS. For given geometry of the module with the number of tubes and perforations assigned, the ratio $\left(\frac{P_2}{P_1}\right)$ cannot be explicitly determined. Hence Eq.(3a) is numerically solved by an iterative technique.

The program for the three modules is herewith shown in appendix 1. The results for the range of $0.5<K<1.2$ are shown in table1.

5. ACOUSTIC CONTROL

The acoustics created due to high pressure and high velocity can be considerably controlled by providing equal number of perforations from the base along the flow direction of the gas in the tube. However, these perforations are to be wrapped up by glass wool packing of medium density as per commercial specifications. The number of perforations to be wrapped by the glass wool is by the rule of thumb is same as the perforations determined from energy considerations.(i.e equation 3a). The packing around the tube covering the perforations undoubtedly abates the severity of the two parameters that are responsible for noise and resonance effects.

6. RESULTS OF COMPUTATION

Typical computer results are provided in Table1.for $0.5<K<1.1$. These results are converted into drawings with the aid of auto cad for fabrication purposes of the modules to be assembled in the proximity of the exit (approximately 2 m below) of the chimney.

Nomenclature to the results shown in Table1:

T_1 – Temperature at the entry.

T_2 – Temperature after expansion

P_1 – Pressure at entry into the module

P_2 – Pressure in the module

G – Total discharge rate of the gas

D – Diameter of the pipe leading the gas into the module

D_P – Diameter of perforations in each module

V_1 – Velocity of the gas at the entry into the tube

V_2 – Velocity of the gas from the perforations

Z_P – Number of perforations provided

7. CONCLUSIONS

Application of energy equation in the design of the several modules in the muffler proved feasible in reducing the thermodynamic parameters with in permissible limits of environmental norms of operation and functioning of chimney stack.

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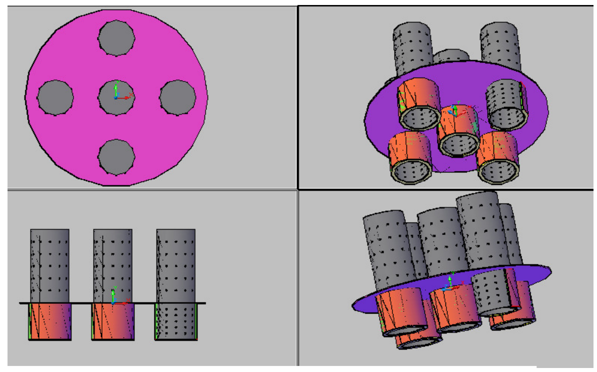
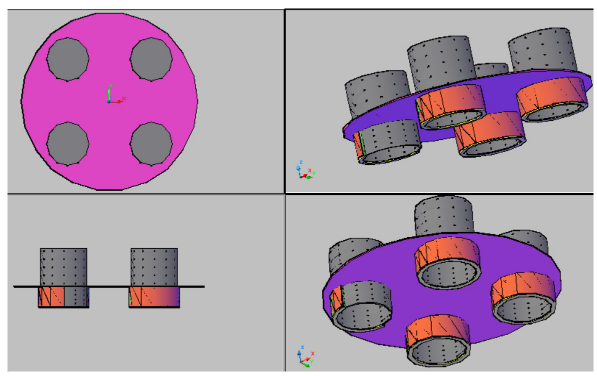
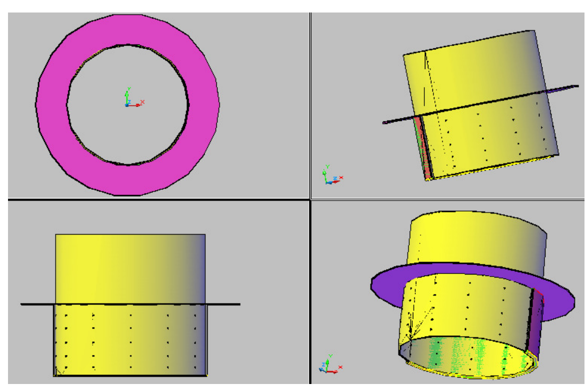
APPENDIX - 1



Chimney – 1st Module (All views)



Chimney – 2nd Module (All views)



Chimney – 3rd Module (All views)

TABLE – 1

RESULTS FROM EQUATION 3a											
RESULTS OF FIRST MODULE IZ=1											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
300	272.8	3.5	2.61	20.91	0.6	0.01	6.95	266.05	0.5	15.34	120
RESULTS OF SECOND MODULE IZ=1											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
272	265.3	2.614	2.41	20.91	0.65	0.02	9.45	133.61	0.5	15.34	80
RESULTS AT THE EXIT OF THE STACK IZ=1											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
265	265.3	2.407	2.41	20.91	1.90	-	4.738	4.74	0.5	15	34
RESULTS OF FIRST MODULE IZ=2											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
300	268.1	3.5	2.48	20.91	0.6	0.01	6.95	277.74	0.65	15.34	120
RESULTS OF SECOND MODULE IZ=2											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
268	258.9	2.483	2.24	20.91	0.65	0.02	9.86	141.83	0.65	15.34	80
RESULTS AT THE EXIT OF THE STACK IZ=2											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
258	258.9	2.241	2.24	20.91	1.90	-	5.029	5.03	0.65	15	34
RESULTS OFFIRST MODULE IZ=3											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
300	262.4	3.5	2.33	20.91	0.6	0.01	6.95	292.84	0.8	15.34	120
RESULTS OF SECOND MODULE IZ=3											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
262	250.9	2.33	2.04	20.91	0.65	0.02	10.4	153.07	0.8	15.34	80
RESULTS AT THE EXIT OF THE STACK IZ=3											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
250	250.9	2.045	2.04	20.91	1.90	-	5.427	5.43	0.8	15	34
RESULTS OF FIRST MODULE IZ=4											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
300	254.8	3.5	2.14	20.91	0.6	0.01	6.95	314.51	0.95	15.34	120
RESULTS OF SECOND MODULE IZ=4											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
254	239.6	2.139	1.79	20.91	0.65	0.02	11.17	170.73	0.95	15.34	80
RESULTS AT THE EXIT OF THE STACK IZ=4											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
239	239.6	1.794	1.79	20.91	1.90	-	6.053	6.05	0.95	15	34
RESULTS OF FIRST MODULE IZ=5											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
300	241.5	3.5	1.83	20.91	0.6	0.01	6.95	357.55	1.1	15.34	120
RESULTS OF SECOND MODULE IZ=5											
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
241	216.9	1.834	1.37	20.91	0.65	0.02	12.69	214.04	1.1	15.34	80
T1	T2	P1	P2	G	D	DP	V1	V2	K	POWER	ZP
216	216.9	1.367	1.37	20.91	1.90	-	7.589	7.59	1.1	15	34