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PARAMETRIC STUDY OF THE COUPLING FACTOR FOR THE STATISTICAL ENERGY ANALYSIS OF PIPING ACOUSTIC VIBRATION

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ABSTRACT

Energy-based finite element model is utilized for the evaluation of the Statistical Energy Analysis (SEA) coupling factor and the dependence of the coupling factor on the different system parameters is studied.

Previous research has shown that the coupling factor is largely dependent on the modal densities of the fluid and pipe subsystems, which depend on the pipe dimensional parameters. The coupling factor depends also on the spectrum of the acoustic power generated, which in turn depends on the mass flow rate, the pressure reduction ratio and the characteristics of the pressure-reducing device.

This study is concerned with the piping system parameters, downstream of the pressure-reducing valve. The system parameters selected for consideration are the pipe diameter to thickness ratio D/T and the pipe length to diameter ratio L/D. The study presents the effect of the variation in these two dimensionless parameters on the coupling factor.

The results of the analysis can be used directly in the formulation of SEA power flow equations for large piping systems with multiple sources of acoustic energy as part of the fatigue life evaluation in critical services.

INTRODUCTION

At points of large pressure reduction in piping systems, such as pressure safety valves or control valves, high level of acoustic energy is produced. This acoustic energy excites pipe wall vibration and can cause fatigue failure at points of localized high stresses such as branch connections.

Reference [11] provides empirical curves for the assessment of the fatigue failure risk due to acoustic induced vibration in industrial plant piping. This method is commonly used in the industry, and is based on evaluating the likelihood of failure as a function of the overall sound power level and the diameter to thickness ratio. The likelihood of failure is then used as a general criterion, based on which design solutions are recommended.

This method is efficient in preliminary screening of the systems for lines with potential problems. However, due to the lack of detailed analysis of the vibration, it is not accurate and can be overly conservative. While detailed finite element analysis can be applied to the study of the acoustic induced vibration, the application is expensive and impractical for large systems with many excitation sources. The Statistical Energy Analysis (SEA) provides an alternative method of quantitative assessment of the vibration, which can be easily applied to large systems.

Lyon et al [1] first developed the SEA method as a noise and vibration estimation technique for complex sound and vibration systems. Woodhouse, J. [2] provided theoretical basis for the application of SAE to structural vibration. Since then the method of SEA has been used for the study of acoustic and vibrational problems in many structural and aerospace applications and has been applied recently to the acoustic vibration in piping systems.

The application of the SEA method requires the determination of the energy coupling factors, which is generally done by means of experimental measurement or finite element analysis.

In a previous work [4] and [5], finite element model has been developed and used to obtain the total dynamic energy of the system and the coupling factors required for building a SEA model of a large piping system. The analysis has been performed for a basic system consisting of a straight pipe filled with acoustic fluid downstream of an acoustic source. In this paper, the model is further applied for the determination of the coupling loss factor variation with the different system parameters.

Note that the finite element analysis is applied here not for the study of the vibration response of a specific piping system, but for investigating the characteristics of the SEA parameters.

STATISTICAL ENERGY ANALYSIS MODEL

The major strength of the SEA method of analysis is in the ability to treat both sound and vibration in a unified manner by considering the energy of vibration and sound in the system. Energy quantities can be averaged more easily and energy relations are not sensitive to small parameter changes. The equations are setup to define the transmission of energy from one part of the system to the other. The result is a set of linear algebraic equations, which can be solved for large and complicated piping systems to determine the dynamic stresses in the pipe for the purpose of fatigue life evaluation.

The method is based on the division of the system into subsystems that can be different physical parts or different vibrational modes of the system. Then coupling factors are applied to define the energy exchange between the subsystems.

For piping systems, the model of the system is built by considering each pipe run to be made up of a pipe element and an acoustic fluid element with elements connected according to the piping system connectivity. Acoustic power sources are placed at the points of pressure-reducing devices. A sample of the SEA model of a part of a piping system is shown in Fig. (1).

Since power is injected into the system through the fluid subsystem by the pressure-reducing valve as the acoustic power source, the net power flow is always from the fluid subsystem to the pipe subsystem. This power flow is the source of the pipe wall vibration and its magnitude defines the dynamic stress level in the pipe.

EVALUATION OF COUPLING LOSS FACTOR

The power flow model for a basic pipe-fluid element is shown in Fig. (2). The power flow from the acoustic fluid element to the pipe element P_{12} is given by the following equation:

$$P_{12} = (E_1 N_2 - E_2 N_1)\, \eta_{12}\, \omega / N_2 \qquad (1)$$

Where:

E_1= total acoustic energy of fluid subsystem, joule
E_2= total dynamic energy of the pipe subsystem, joule
N_1= modal density of fluid subsystem, sec
N_2= modal density of pipe subsystem, sec
P_{12} = net power flow from subsystem 1 to 2, watt
ω = band center frequency, rad/sec
η_{12} = coupling loss factor

Equation (1) above is the definition of the Coupling Loss Factor CLF [3].

The power loss of the pipe element P_{L2} can be obtained from the damping ratio and the dynamic energy of the pipe element as follows:

$$P_{L2} = 2\, \pi\, \zeta\, E_2\, f = \omega\, \zeta\, E_2 \qquad (2)$$

Where:

ζ = damping ratio

The balance of power flow for the pipe element gives the following relationship:

$$P_{12} = P_{L2} \qquad (3)$$

From Eq. (1) to (3), the CLF can be obtained as follows:

$$\eta_{12} = \zeta\, E_2\, N_2 / (E_1 N_2 - E_2 N_1) \qquad (4)$$

FINITE ELEMENT ANALYSIS

Finite element analysis by use of ANSYS 11.0 software is performed for the evaluation of the coupling loss factor.

The finite element model consists of a run of straight pipe filled with acoustic fluid. The pipe is modeled by use of 3D solid element with eight nodes and three displacement degrees of freedom at each node. The acoustic fluid is modeled by means of 3D acoustic fluid element with pressure as the single degree of freedom. The layer of acoustic fluid elements in contact with the pipe solid elements has three displacement degrees of freedom, in addition to the pressure degree of freedom, to interact with the solid elements.

An acoustic power source is placed at the upstream end of the pipe and the downstream end is provided with special acoustic fluid boundary elements to simulate an infinite absorbing medium. The acoustic power source is modeled as a sinusoidal pressure wave with a frequency sweeping the frequency range of interest.

The finite element mesh size has been selected to provide accurate solution over all the frequency range. The convergence and accuracy of the finite element model has been already demonstrated in previous work [5].

Dynamic harmonic analysis is performed to obtain the system acoustic fluid energy and the pipe vibrational energy. The dynamic modal analysis is used to obtain the pipe and fluid modal densities as function of frequency, as described in the previous work [4].

ACOUSTIC POWER SOURCE

International Electrotechnical Commission IEC Standard 60534-8-3 [12] provides a method for obtaining the spectrum of the acoustic power produced in the system at the downstream of a pressure-reducing valve. The total acoustic power W_a is related to the flow mechanical power W_m by the following equation:

$$W_a = \eta_a W_m \quad (5)$$

Where η_a is a dimensionless acoustic efficiency factor, which is a function of the pressure-reducing valve parameters, flow rate, pressure ratio and gas properties.

The IEC standard provides the distribution of the acoustic power over a wide frequency range as discrete values of acoustic power for each of 1/3 octave frequency band. This spectrum has a haystack shape with a middle peak frequency f_p dependent on the Strouhal number S_t of the flow and a gradual drop on both sides around the peak frequency.

Since this study is concerned with the effect of the piping parameters, and in order to separate the effect of the piping system parameters from other effects such as the pressure-reducing valve and flow parameters, the spectrum has been taken as a uniform spectrum with acoustic pressure amplitude as required to give the same acoustic power within the frequency range of interest.

The range of frequency considered is 1000 Hz – 6300 Hz, which covers the most important range for the acoustic vibration induced in piping systems as has been demonstrated by previous research work [4] and [14]. This range covers the range from the cut-off frequency f_c to the ring frequency f_r. The last two characteristic frequencies are defined as follows:

The cut-off frequency f_c is the acoustic natural frequency corresponding to acoustic mode (p=1 & q=0), where p is the number of plane diametral nodal surfaces and q is the number of cylindrical nodal surfaces concentric with the cylinder axis. The cut-off frequency is given by the following equation [14]:

$$f_c = 1.84\, C_2 / (\pi\, D_i) \quad (6)$$

The ring frequency f_r is defined as the frequency of the stress wave in the pipe corresponding to wavelength equal to the pipe circumference. The ring frequency is given by the following equation:

$$f_r = C_s / (\pi\, D_i) \quad (7)$$

Where C_s is the speed of sound in the pipe material.

PARAMETRIC STUDY AND RESULTS

A detailed parametric study of the effect of variation of the following two dimensionless parameters on the CLF is conducted:

- Pipe diameter to thickness ratio D/T
- Pipe length to diameter ratio L/D

The detailed parameters of the system used in this study are shown in Table (1). The basic parameters in the table correspond to the blowdown valve of flash gas system in offshore gas plant. The pipe size, schedule and material are typical for such blowdown systems. A structural damping ratio of 0.01 is considered to apply to all the frequency range. This value is based on reported experimental results of pipe vibration under acoustic fatigue [15].

For the study of the effect of the D/T parameter, a constant L/D value of 9.3 is used and the D/T value is varied from 10 to 78. For each value of D/T, the total energy of the acoustic fluid subsystem and pipe subsystem E_1 and E_2 and the modal densities of the two subsystems N_1 and N_2, are obtained against the frequency f. The quantities E_1, E_2, N_1 and N_2 are used to calculate the CLF against the frequency. Then the CLF is averaged over all the frequency range for each value of the system dimensionless parameter D/T.

For the study of the effect of L/D parameter, similar procedure is applied with constant value of D/T of 63 and L/D varies from 3.7 to 18.6.

Figures (3) to (11) show the results of the system energy quantities E_1 and E_2, the modal densities N_1 and N_2 and the CLF against the frequency for selected values of D/T. Figure (12) shows the CLF averaged over the frequency range against D/T.

Figures (13) to (22) show the results of the system energy quantities E_1 and E_2, the modal densities N_1 and N_2 and the CLF against the frequency for selected values of L/D. Figure (23) shows the CLF averaged over the frequency range against L/D.

As shown in Fig. (12), the average CLF increases with the increase of D/T. This is essentially due to the fact that the increase of the pipe wall thickness results in a general decrease in the pipe subsystem modal density over the range of the frequency considered, which in turn results in decrease in the energy transmitted from the fluid subsystem to the pipe subsystem. This reflects the importance of the modal densities of the two subsystems in the evaluation of the power transmitted from the acoustic fluid to the pipe.

As shown in Fig. (23), the average CLF decreases with the increase of L/D, initially at high rate then the rate of decrease levels off. The modal density of the acoustic fluid subsystem has similar values for different values of L/D at the higher end of the frequency range. At the lower end of the frequency range, the modal density of the fluid subsystem is higher for larger L/D. This means that in general, and over the whole frequency range, the modal density of the fluid subsystem is higher for larger L/D values. Therefore, the average CLF over the whole frequency range is lower for larger L/D ratio.

SUMMARY AND CONCLUSIONS

The Coupling Loss Factor CLF, required for SEA of acoustic induced vibration in piping systems, has been obtained for a basic piping component consisting of a straight pipe filled with acoustic fluid by means of finite element analysis. A parametric study of the effect of the variation of the pipe dimensional parameters D/T and L/D on the CLF has been conducted for a wide frequency range.

The results of the analysis provide sound basis for the definition of the CLF as a function of the pipe parameters, which can be applied directly in the formulation of the energy flow equations required for the performance of SEA of acoustic induced vibration in piping systems.

NOMENCLATURE

C_2 = sound speed in fluid downstream of pressure-reducing device, m/sec
C_s = sound speed in pipe material, m/sec
D = pipe average diameter, m
D_i = pipe inside diameter, m
E_1= total acoustic energy of fluid subsystem, joule
E_2= total dynamic energy of the pipe subsystem, joule
f = frequency, Hz
f_c = cut-off frequency, Hz
f_r = ring frequency, Hz
f_p = peak frequency, Hz
L = pipe length, m
N_1= modal density of fluid subsystem, sec
N_2= modal density of pipe subsystem, sec
P_{12} = net power flow from subsystem 1 to subsystem 2, watt
P_i = power input from the pressure-reducing device, watt
P_o = power at the exit of the fluid subsystem, watt
P_{L1} = power loss of the fluid subsystem, watt
P_{L2} = power loss of pipe subsystem, watt
S_t = Strouhal number
T = pipe wall thickness, m
W_a = sound power at the exit of the pressure-reducing device, watt
W_m = mechanical power of the flow, watt
ζ = damping ratio
η_{12} = coupling loss factor
η_a = acoustic efficiency factor
ω = band center frequency, rad/sec

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Pipe Size	DN 250
Pipe Outside Diam.	273 mm
Pipe Material	Stainless Steel A 312
Pipe Wall Thickness	3.40 mm–25.40 mm
Pipe Length	500 mm – 2500 mm
Modulus of Elasticity	195 GPa
Poisson's Ratio	0.29
Upstream Pressure P1	6000 KPa
Downstream Pressure P2	2000 KPa
Upstream Temperature T1	80 C
Upstream Density	2.04 Kg/m^3
Downstream Density	0.76 Kg/m^3
Mass Flow Rate	10 Kg/sec
Molecular Weight	28.4 Kg/Kmol
Valve Pressure Recovery Factor	0.90
Pipe Damping Ratio	0.01

Table 1: System Parameters

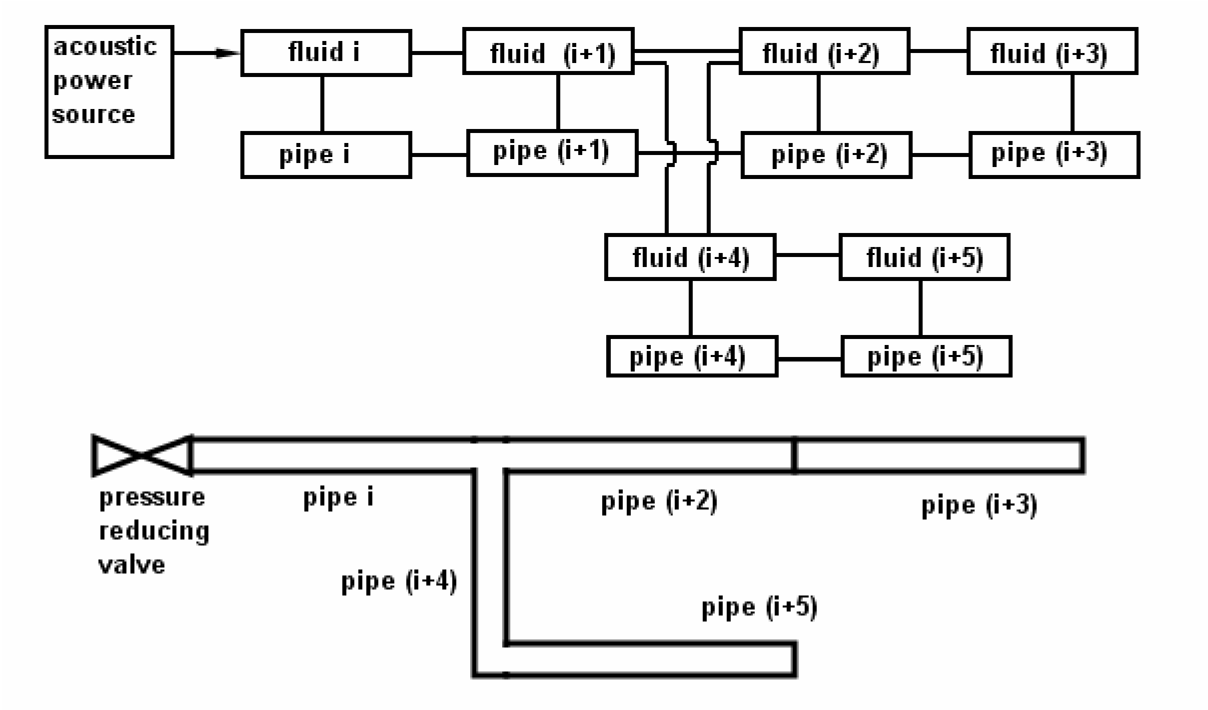

Figure 1: Statistical Energy Analysis Model of a Piping System

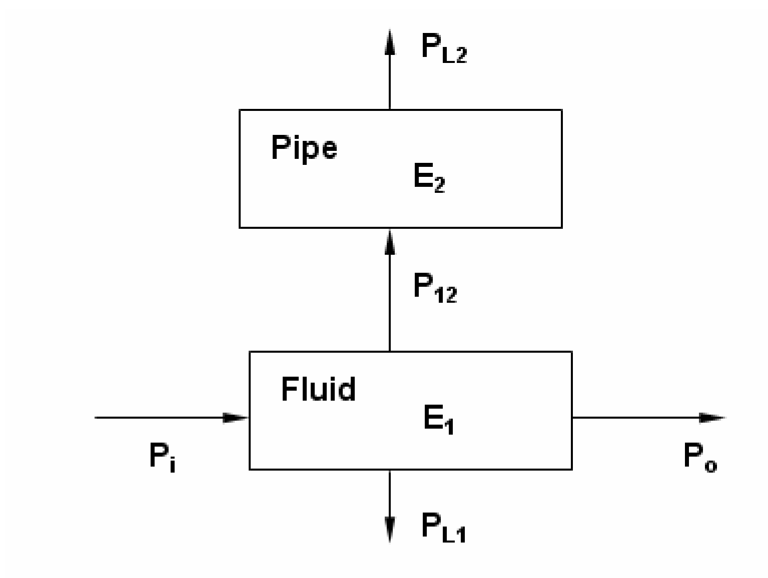

Figure 2: Power Flow Model of Pipe-Fluid System

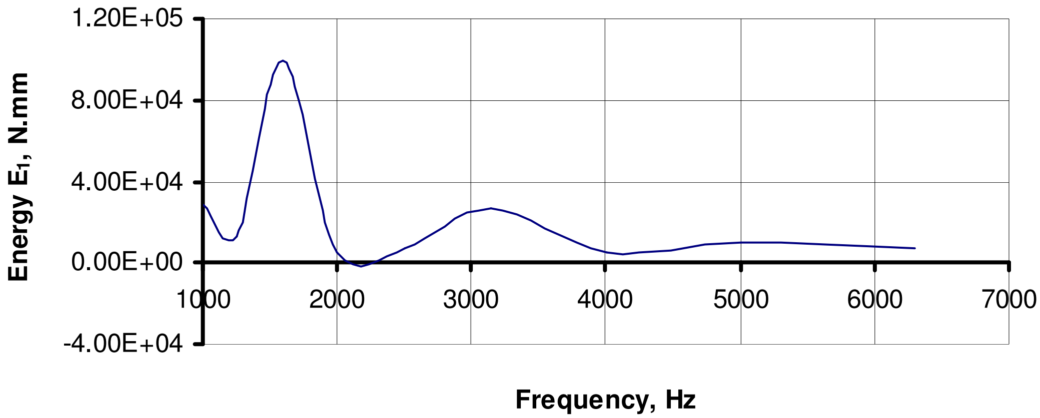

Figure 3: Energy of Acoustic Fluid Against Frequency – L/D =9.3

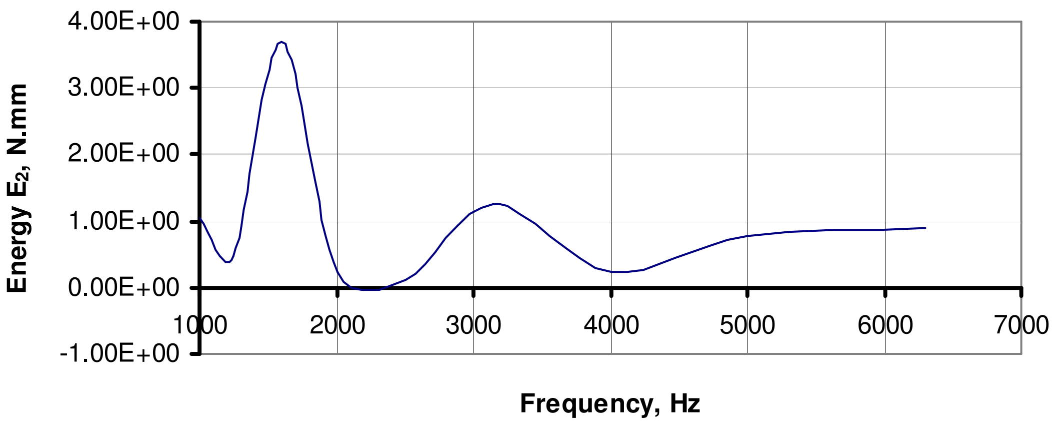

Figure 4: Energy of Pipe Against Frequency – D/T=63 & L/D =9.3

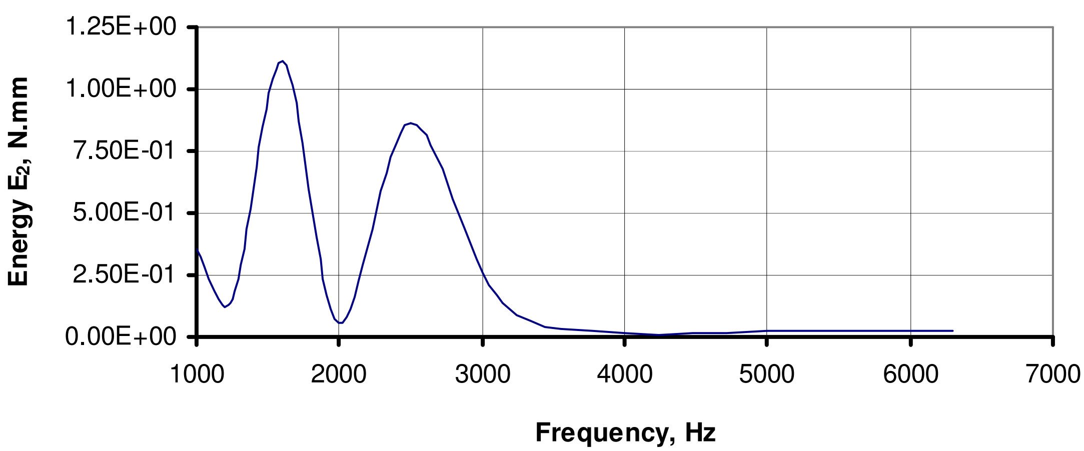

Figure 5: Energy of Pipe Against Frequency – D/T=15 & L/D =9.3

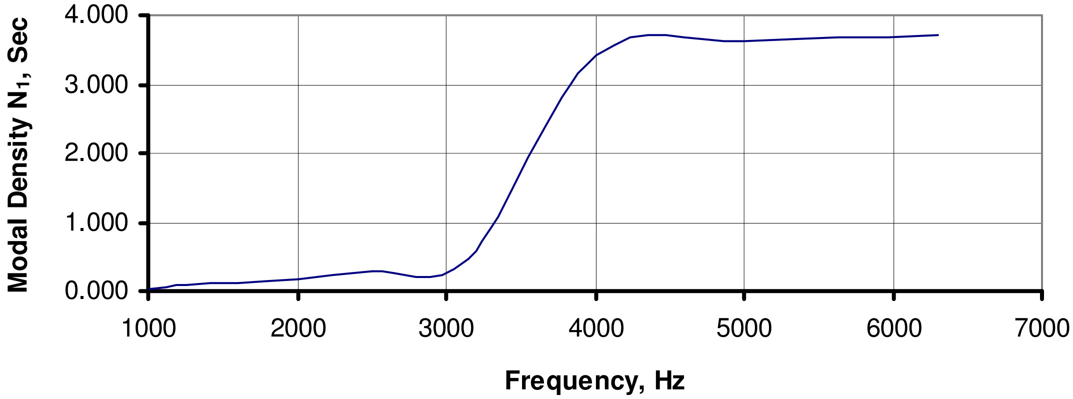

Figure 6: Modal Density of Acoustic Fluid Against Frequency – L/D =9.3

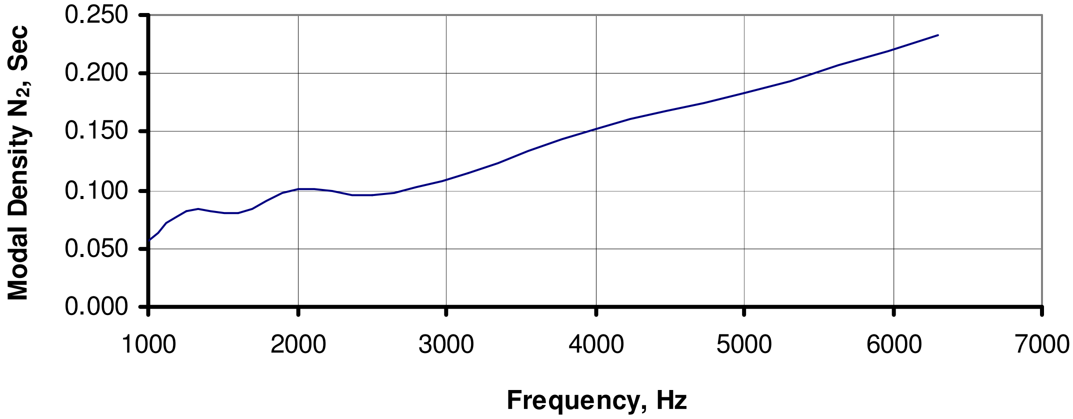

Figure 7: Modal Density of Pipe Against Frequency – D/T = 63 & L/D =9.33

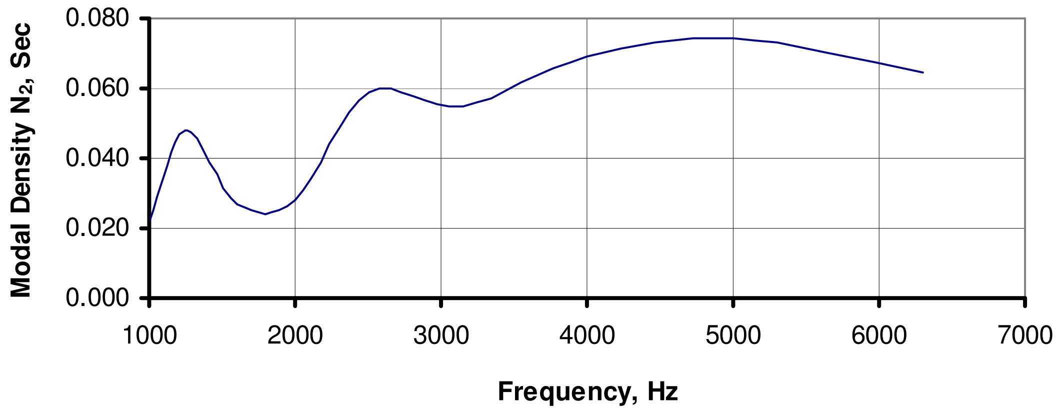

Figure 8: Modal Density of Pipe Against Frequency – D/T = 15 & L/D =9.3

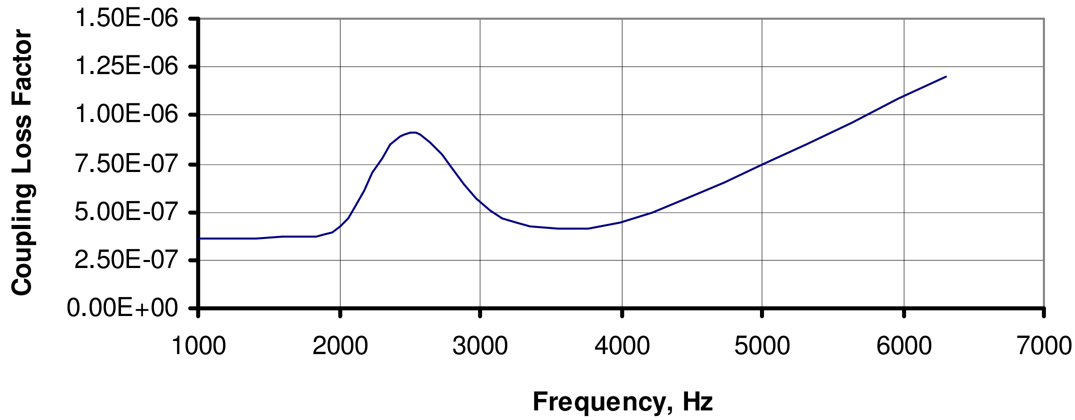

Figure 9: Coupling Loss Factor Against Frequency – D/T = 63 & L/D =9.3

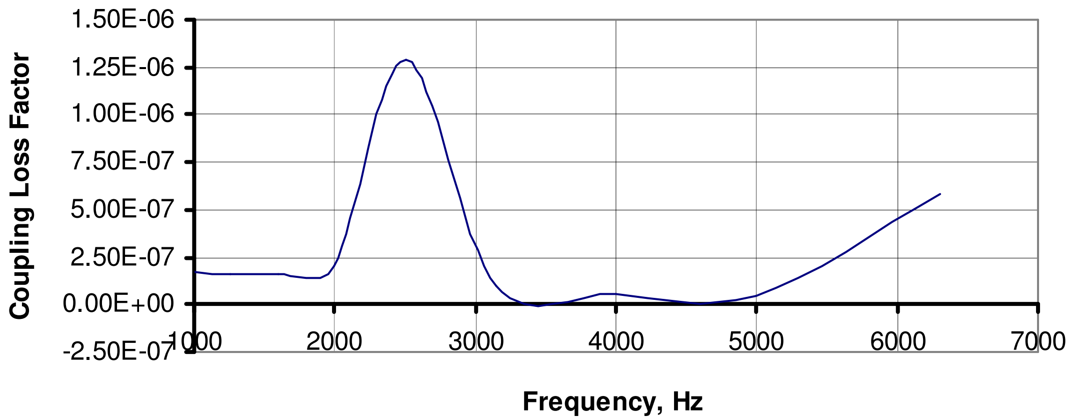

Figure 10: Coupling Loss Factor Against Frequency – D/T = 29 & L/D =9.3

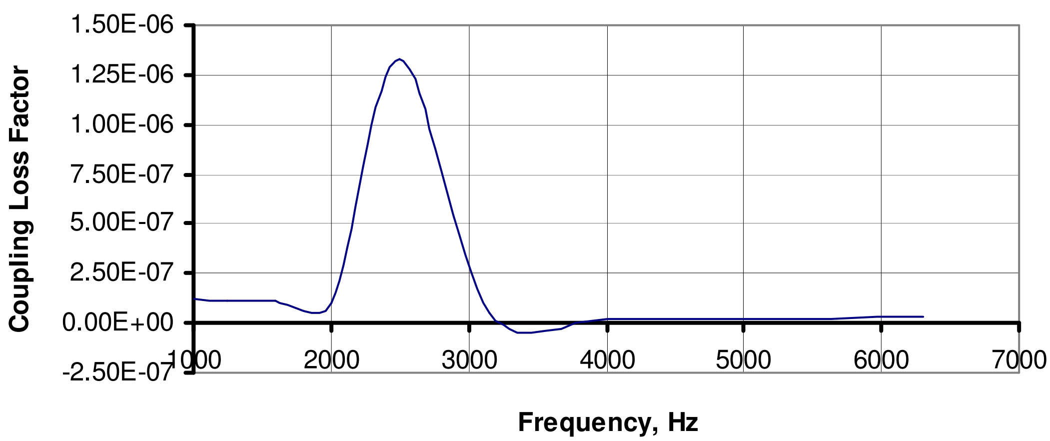

Figure 11: Coupling Loss Factor Against Frequency – D/T = 15 & L/D =9.3

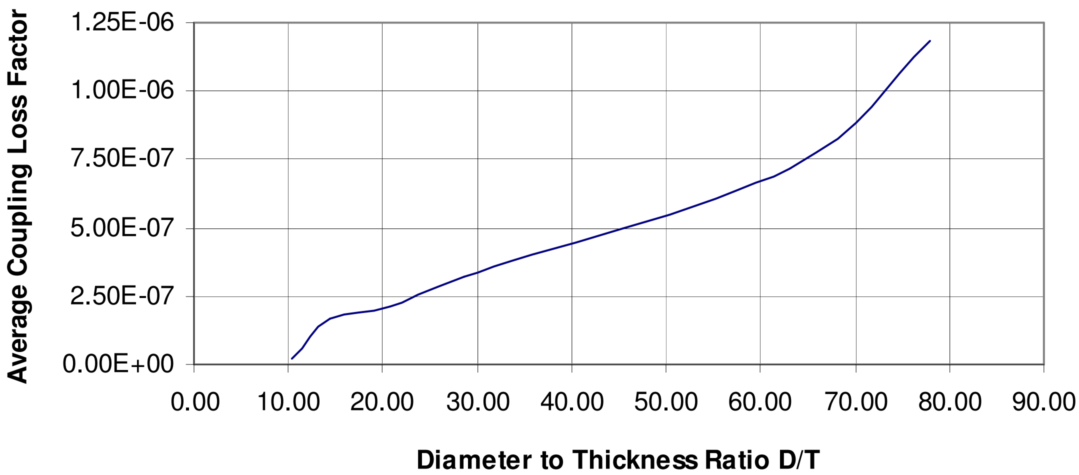

Figure 12: Average Coupling Loss Factor Against D/T – L/D =9.3

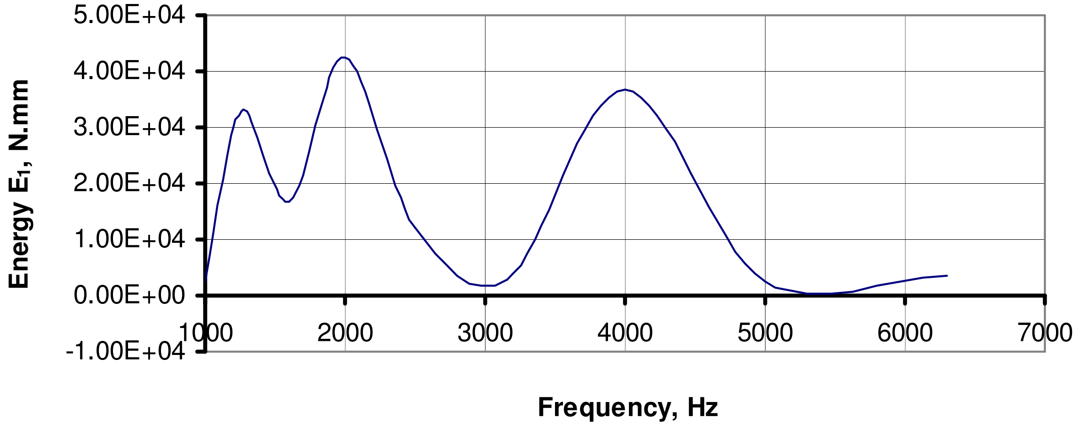

Figure 13: Energy of Acoustic Fluid Against Frequency – L/D =3.7 & D/T=63

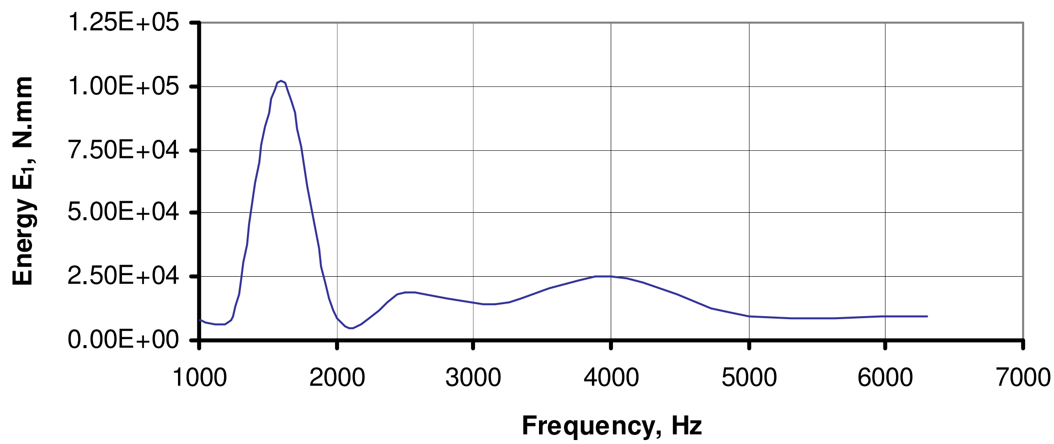

Figure 14: Energy of Acoustic Fluid Against Frequency – L/D =18.6 & D/T=63

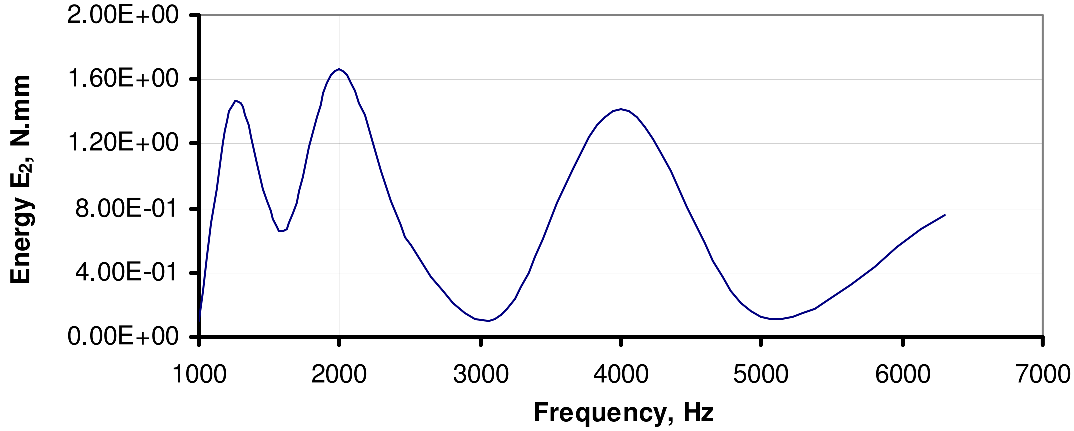

Figure 15: Energy of Pipe Against Frequency – L/D =3.7 & D/T=63

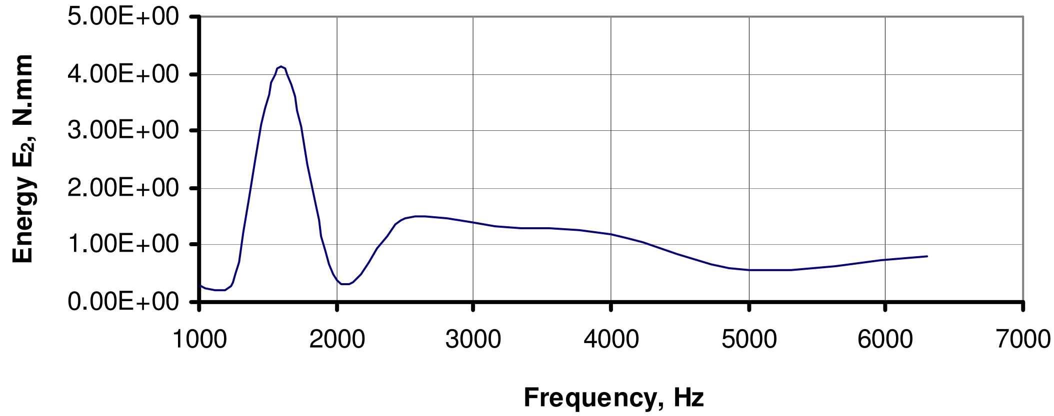

Figure 16: Energy of Pipe Against Frequency – L/D =18.6 & D/T=63

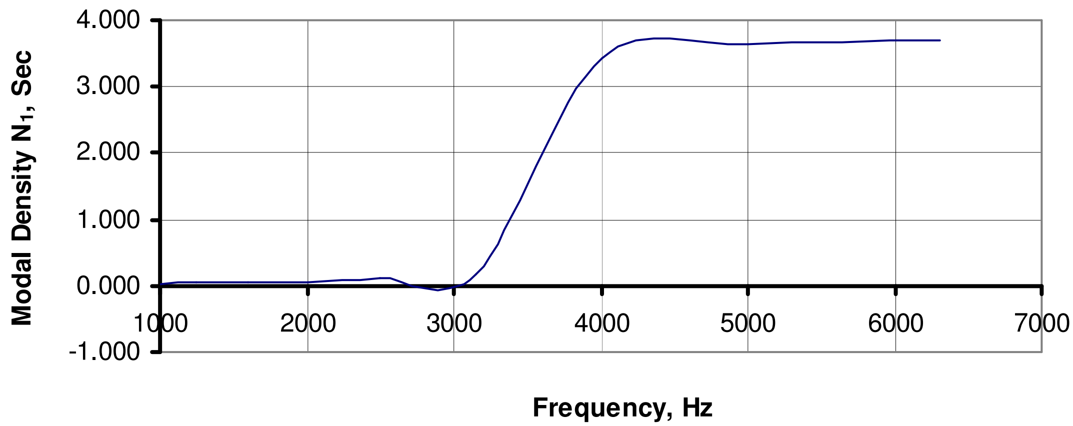

Figure 17: Modal Density of Acoustic Fluid Against Frequency – L/D =3.7 & D/T=63

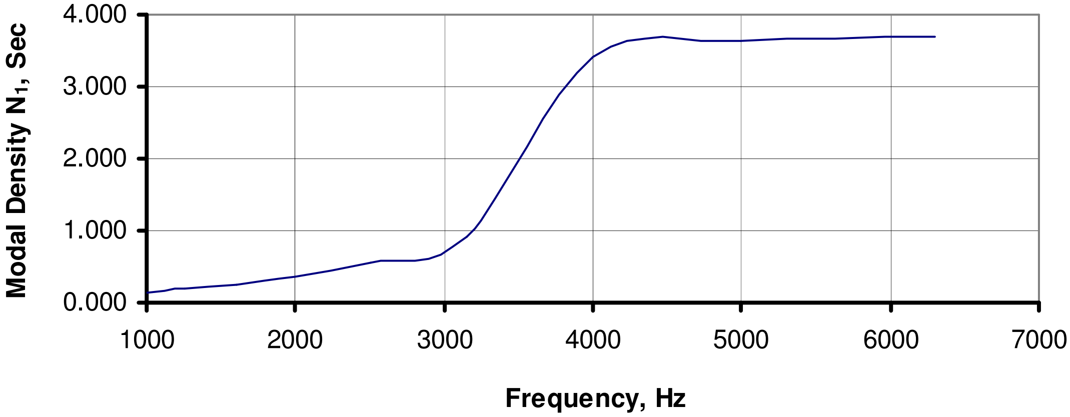

Figure 18: Modal Density of Acoustic Fluid Against Frequency – L/D =18.6 & D/T=63

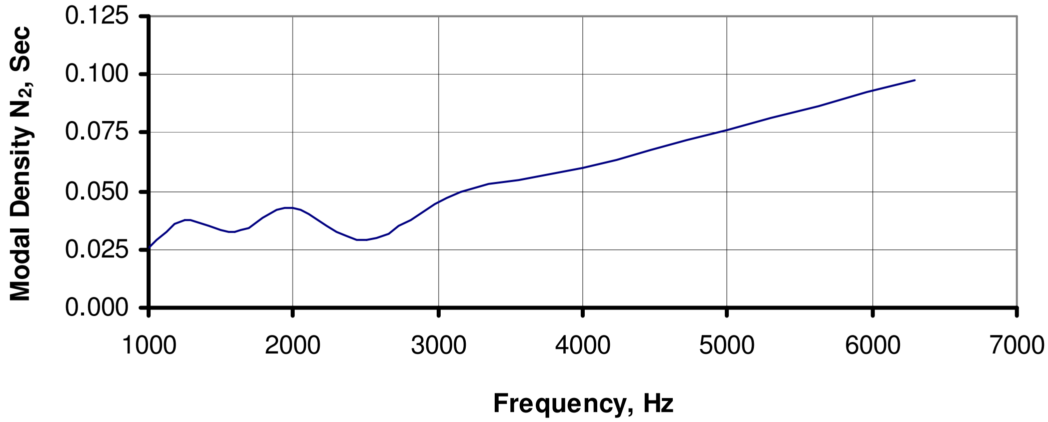

Figure 19: Modal Density of Pipe Against Frequency – L/D =3.7 & D/T=63

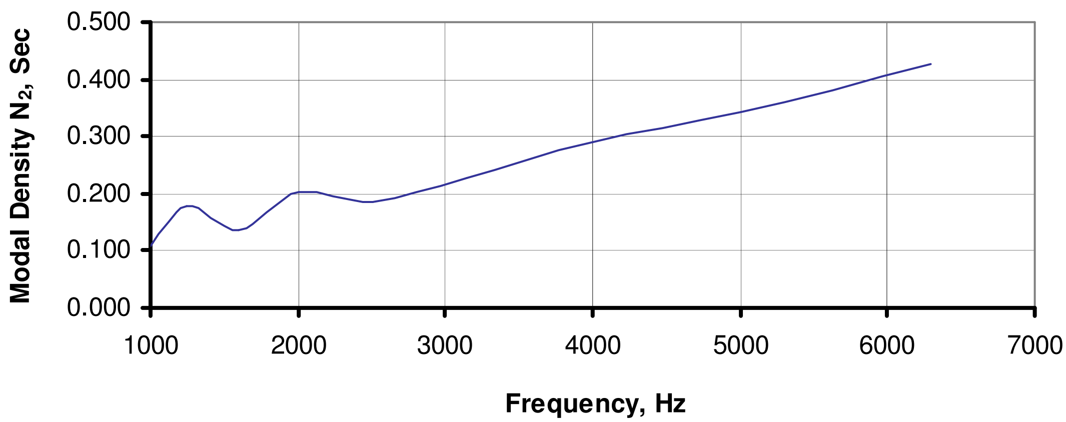

Figure 20: Modal Density of Pipe Against Frequency – L/D =18.6 & D/T=63

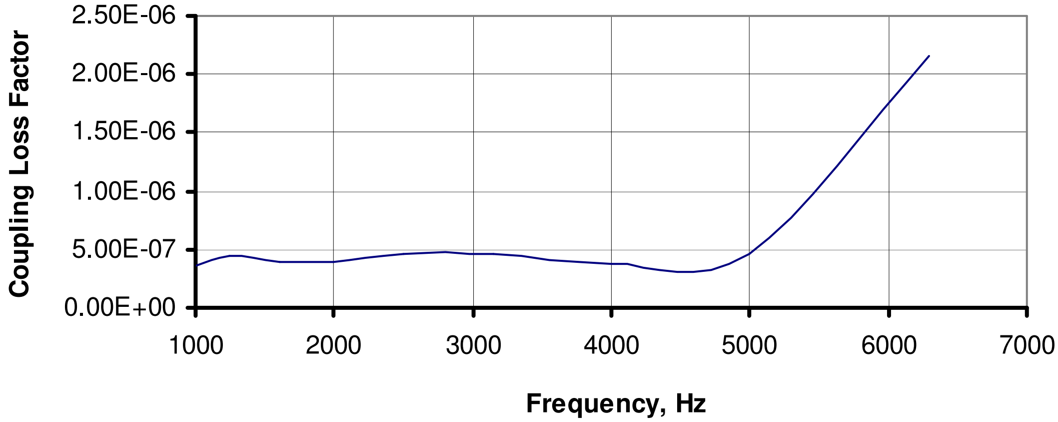

Figure 21: Coupling Loss Factor Against Frequency – L/D =3.7 & D/T=63

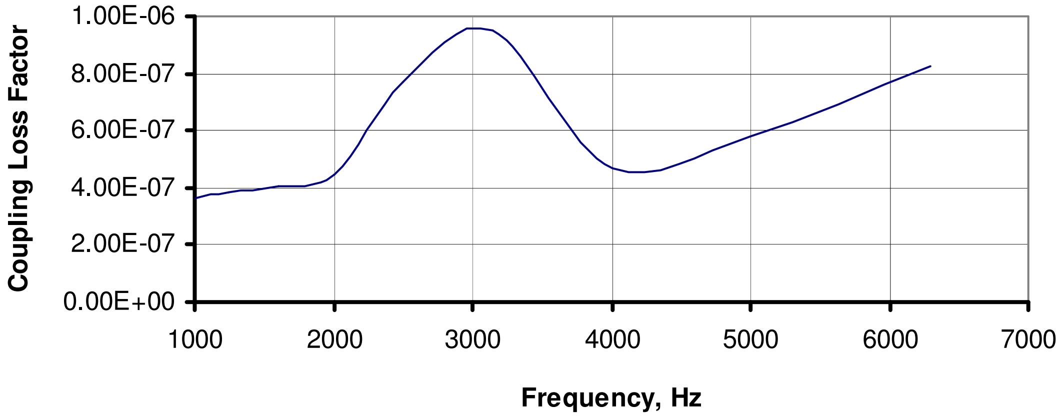

Figure 22: Coupling Loss Factor Against Frequency – L/D =18.6 & D/T=63

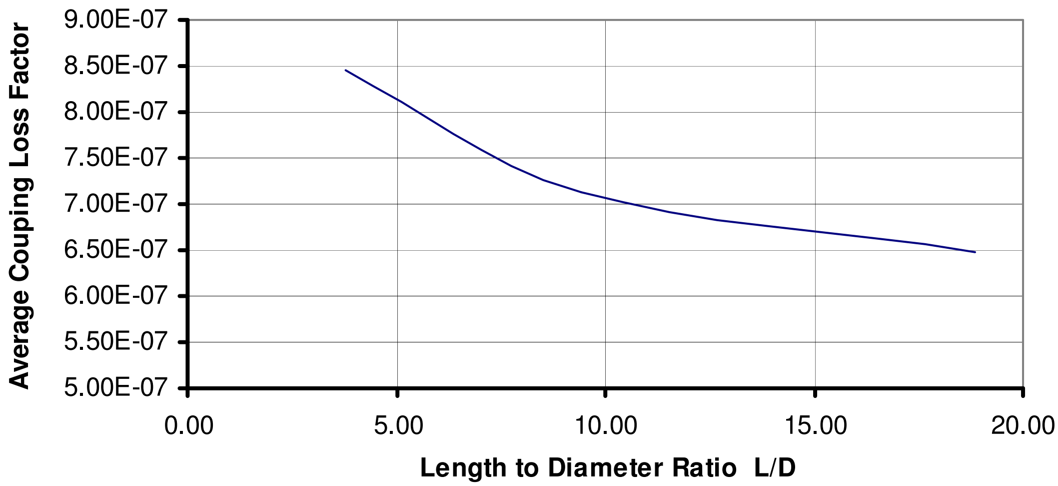

Figure 23: Average Coupling Loss Factor Against L/D – D/T=63