Noise Attenuation by Helmholtz Resonators Attached to a Duct.

A thesis submitted to the department of Mechanical and Chemical Engineering (MCE), Islamic University of Technology (IUT), in the partial fulfillment of the Requirement for the degree of Bachelor of Science in Mechanical Engineering.

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DECLARATION

This is to certify that the work presented in this thesis is an outcome of the research carried out by the authors under the supervision of Prof. Dr. Md. Zahid Hossain and Lecturer Md. Shahriar Islam.

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Although we tried our best to complete this thesis flawlessly, we seek apology if there is any mistake found in this report.

ABSTRACT

The resonating frequency and the transmission loss due to two 1-DOF cylindrical Helmholtz resonator attached to a duct is thoroughly investigated in this literature to attenuate low frequency noise using finite element method. Primarily, the relative spacing between the two identical resonators attached to a duct is investigated in this study to find the optimum positions between the resonators. The investigation is further carried out by changing the dimensions of the cavity and neck for the optimum relative position .Finally, based on these several investigations, a conclusion is drawn for an optimized system of two identical 1-DOF Helmholtz resonators in terms of position, orientation and geometry. The results shows a significant improvement in noise reduction compared to a single 1-DOF Helmholtz resonator .The results of a single 1-DOF Helmholtz resonator of a published literature were also validated both analytically and numerically in this study.

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CHAPTER 1

Introduction and Background

1.1Introduction

The problem of the low frequency noise is inherent to road vehicles, IC engines, aircrafts, pipelines, power transformers, compressors etc. This humming nature not only creates disturbance to the personnel working at various industries but also causes serious health problems like hearing impairment, hypertension, heart disease, birth defects and changes in immune system. There are various techniques for noise attenuation like using reactive silencers, barriers, acoustically absorptive materials like fiberglass, mineral wool, Rockwell, porous foams .But one of the most effective ways of noise reduction is using a dynamic damper **HELMHOLTZ RESONATOR** which can attenuate significantly according to most recent research. So, Helmholtz resonator has been studied and applied for many years in various fields of Engineering to attenuate noise level signifiantly like Currently Dresser Rand uses Duct resonator arrays which has reduced up to 12 db at multistage centrifugal compressor on a platform in the North Sea . Subwoofers in producing low frequencies in surround sound systems. Architectural acoustics to reduce undesirable sounds in bulidings, Acoustic liners in present aircraft engines,In motorcycle and car exhausts as replacement of reed valve in 2 stoke engines. More recently, efforts have been made to change the acoustic response of the intake manifold system on-line of the car engines by tuning at two speeds and at a range of frequencies. The results are higher fuel economy and improved performance over a range of engine speeds.

1.2 Background

Neglecting the spatial distribution, the classical theory of a Helmholtz Resonator as an equivalent spring–mass system was developed by Rayleigh, J. W. S (1945). The effect of different aperture geometries on the resonance frequency of resonators was discussed by Ingard, U. (1953). The effect of the shape of the vessel of the resonator on the resonance frequency was investigated by Alster, M.(1972). Tang and Sirignano[4] developed a generalized theory of the Helmholtz resonator and investigated the performance of the conventional Helmholtz resonator and the quarter-wave tube. An experimental study was done by Panton, R. L., & Miller, J. M. (1975) on the end correction theory of Ingard. To account for the non-planar wave propagation in both the neck and the cavity, multi-dimensional analytical approaches have also been employed to predict the sound attenuation in Helmholtz resonators with circular concentric cavity [10], circular asymmetric cavity [11], and extended neck [12]..Extensive study on theoretical, computational and experimental investigation of Helmholtz resonators with fixed volume: lumped versus distributed analysis performed by Selamet, A., 1995). An analytical model of mechanically-coupled mounted 1-DOF Helmholtz resonator for a wide bandwidth was developed by Griffin, S. (2001).An experimental investigation showed that when the center distance of two identical resonators was greater than a quarter wavelength apart, the sound transmission loss was larger than that of a single resonator, however, when two resonators at same resonant frequency were in close proximity, the two resonators interacted and lead to a decrease in the overall performance compared to that of a single resonator (Soh 2001). Based on the experimental observation of (Soh 2001), the effect of the resonator position on the noise reduction and the relationship between two or more closely spaced identical 1-dof resonators close to the interior wall of a chamber core cylindrical fairing was experimentally investigated by D. Li, (2003) .A new design methodology for one and two degrees of freedom Helmholtz resonators attached to pipelines leading to an optimized transmission loss proposed by Mekid, S., &Farooqui, M. (2012. If the array of 1-DOF Helmholtz resonators in same plane are placed where the crest of the wave falls, the performance of noise attenuation increases significantly. (Farooqui, M. 2012).

But there hasn't been any significant study on the spacing between the two identical Helmholtz resonators attached to a duct or pipelines.

CHAPTER 2

LITERATURE REVIEW

2.1 Basics of Acoustics

Sounds consist of pressure waves. Sound pressure level (SPL) or sound level is a logarithmic measure of the effective sound pressure of a sound relative to a reference value. It is measured in decibels (dB) above a standard reference level. The standard reference sound pressure in air or other gases is 20 µPa, which is usually considered the threshold of human hearing (at 1 kHz).

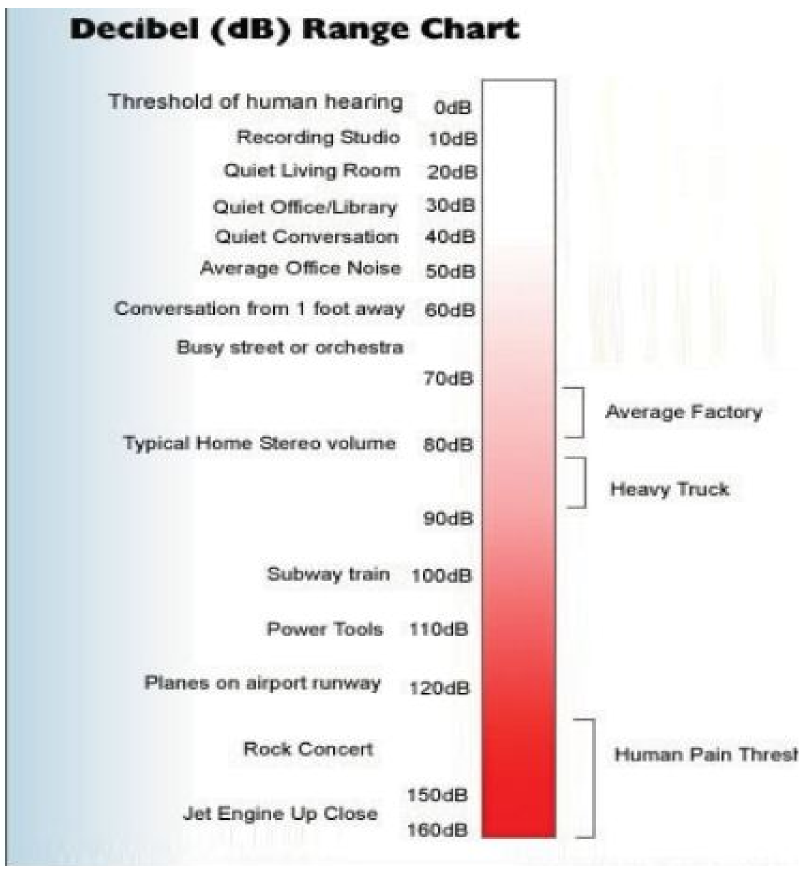


Figure 2.1.1: Chart displaying a comparison of Noise levels [18]

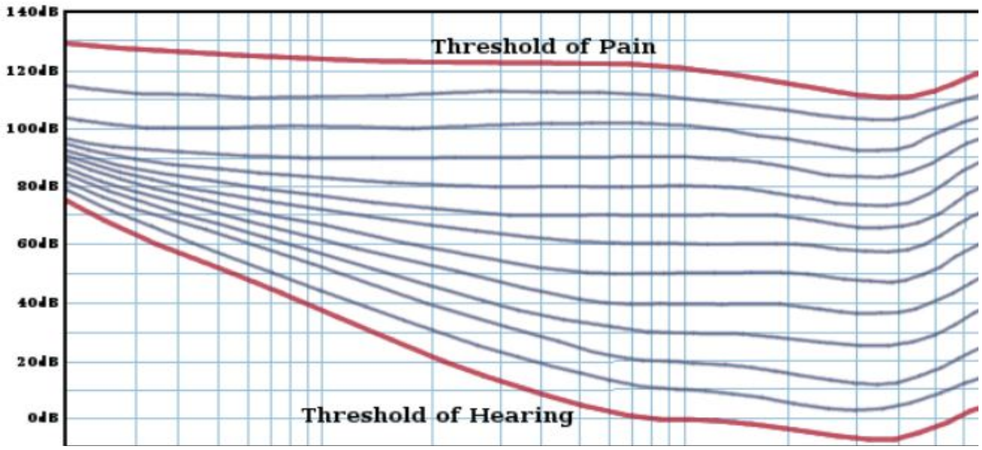


Figure 2.1.2 Chart displaying a comparison of Noise levels with frequency

SOUND FIELDS

In order to predict or model noise from equipment we need to understand, or better, define the sound fields and the predicted sound level associated with those fields. The near field, far field, free field and reverberant field are the most common. The regions that describe the sound fields and sound propagation are illustrated in Figure 2.3

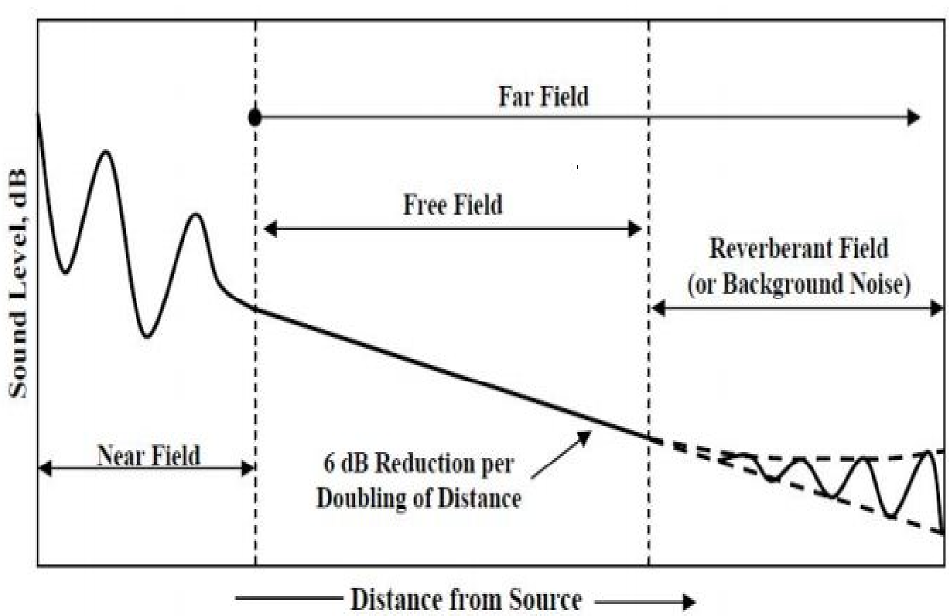


Figure 2.1.3 Definitions of Sound Fields

The near field region is probably the most difficult to predict as this describes the region where noise propagation is not well developed and construction techniques and equipment installation details that are generally unknown this affect the amount of noise around the equipment or structure. The far field starts where the sound field becomes more stable and propagation is fairly uniform. This location is frequency (wavelength) dependent and is usually two to four major source dimensions (width and height as you look at the source) away from the noise source. The free field describes where sound freely propagates and spreads uniformly. The sound level decreases approximately six decibels for every doubling of distance. As you get farther away from the source the decay rate starts to flatten out once the sound from the source approaches the ambient or background sound level as illustrated in the right section of the Fig 1.2. The reverberant field occurs where freely propagating sound waves are reflected back from a wall, a ceiling or other surfaces again causing variation in sound levels as illustrated.

Transmission Loss*(TL):* The accumulated decrease in acoustic intensity as an acoustic pressure wave propagates outwards from a source. TL was calculated from the sound pressure value at the proximal node of the outlet by the following equation

$$Tl = 20\log_{10}\left(\frac{Sound\ Pressure\ at\ Outlet\ port,\ Pa}{Reference\ pressurel\ evel(20\times10^{-6}Pa)}\right)$$

Resonance and Resonance Frequency *(f):*

In physics, resonance is the tendency of a system to oscillate with greater amplitude at some frequencies than at others. Frequencies at which the response amplitude is a relative maximum are known as the system's resonant frequencies, or resonance frequencies.

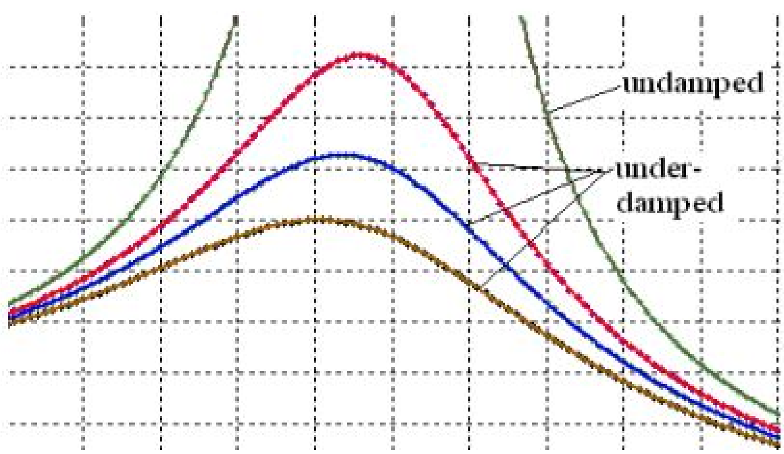


Figure 2.1.4: Resonance Curve

Sound Pressure Level:

Sound pressure level (SPL) or sound level is a logarithmic measure of the effective sound pressure of a sound relative to a reference value. It is measured in decibels (dB) above a standard reference level. The standard reference sound pressure in air or other gases is 20 µPa, which is usually considered the threshold of human hearing (at 1 kHz).

$$L_p = 10\log_{10}\left(\frac{{p_{\rm rms}}^2}{{p_{\rm ref}}^2}\right) = 20\log_{10}\left(\frac{p_{\rm rms}}{p_{\rm ref}}\right)\ \text{dB (SPL)},$$

NOISE REDUCTION TECHNIQUES

Noise reduction techniques fall under three broad categories: Primary noise reduction, Secondary noise reduction and Tertiary noise reduction. Primary noise reduction involves reducing noise at the source; this usually means a complete redesign of the component or an add-on solution within the compressor itself. Secondary solutions include externally blocking noise; acoustic lagging, sound insulation, sound enclosures and sound reduction in transmission are some examples. Tertiary solutions are solutions based on blocking noise on the receiving side; these can be in the form of ear plugs or some kind of ear protection for the personnel working around the facility. Normally in most industrial environments either tertiary or secondary noise reduction solutions are employed [2].

2.2 Helmholtz principle

Helmholtz resonance is the phenomenon of air resonance in a cavity, such as when one blows across the top of an empty bottle. A Helmholtz resonator consists of two parts, a rigid-walled cavity of volume *V*, and a neck or an opening with area *S* and length *L*. When excitation is applied on the neck, the fluid particle in the neck moves as a unit to store kinetic energy and provides the mass element .These mass enters in the cavity increasing the cavity pressure .The opening radiates sound, thus provides the resistance element, and the viscous losses of the moving air in the neck provides additional resistance. The compressible air inside the cavity stores potential energy, and is modeled as stiffness element. So it works like an equivalent mass-spring system. The classical formula for calculation of the resonance frequency of the Helmholtz resonator is $f = \frac{c}{2\pi}\sqrt{\frac{S}{L_{eff}V}}$ where c is the speed of the sound, L_{eff} is the effective neck length including end correction, s is the spring stiffness and V is the volume of the resonator.

2.3 Lumped element model of the Helmholtz resonator

The Helmholtz resonator acts as an acoustic filter element. The dynamic behavior of the Helmholtz resonator can be modelled as a lumped system if the dimensions of the Helmholtz resonator are smaller than the acoustic wavelength. The air in the neck is considered as an oscillating mass and the large volume of air is taken as a spring element [4]. Damping appears in the form of radiation losses at the neck ends and viscous losses due to friction of the oscillating air in the neck. Figure 2 shows this analogy between the Helmholtz resonator and a vibration absorber with defined parameters [8]. In Fig. 2, Ma is the acoustic mass of the resonator and Mm is the mass of the mass-spring-damper system. F is the force applied at the resonator neck entrance and P is the pressure at the neck entrance. V and Ra are respectively the cavity volume and acoustic damping capacity of the Helmholtz resonator. K and Rm are respectively the stiffness and damping capacity of the mass-spring-damper system. ω is the excitation frequency.

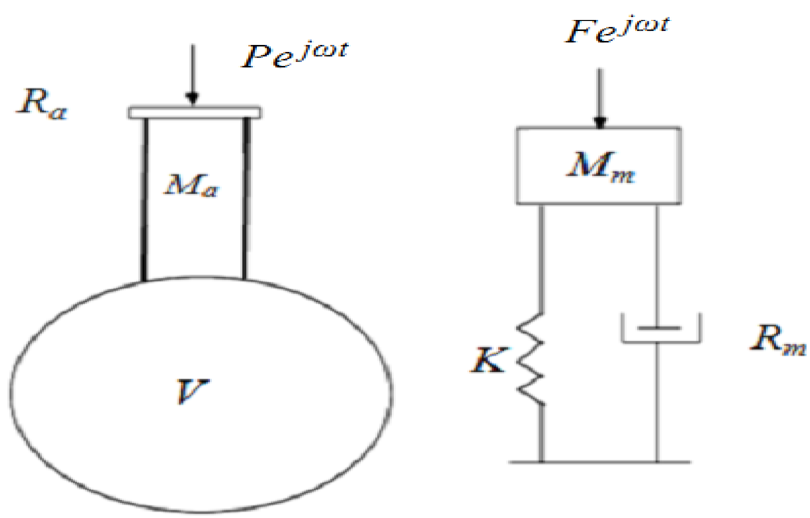


Figure 2.3: Helmholtz resonator and vibration absorber

2.4 Adaptation of a Single 1-DOF Helmholtz Resonator

There are various types of Helmholtz resonators like conical, spherical and cylindrical type. Among conical, spherical and cylindrical Helmholtz resonators, the overall noise attenuation performance of cylindrical resonator is much better (Farooqui, M. 2012). So, the cylindrical Helmholtz resonator has been chosen for the present study.

The resonating frequency (*f*) and the transmission loss (TL) due 1-DOF Helmholtz resonator attached to duct are represented by the equation 3[16] and the equation 4 [6].

$$f \;=\; \frac{c}{2\pi}\sqrt{-\frac{3.L_n+L_CA}{2{L_n}^3}+\sqrt{\left(-\frac{3.L_n+L_CA}{2{L_n}^3}\right)^2+\frac{3A}{{L_n}^3L_C}}} \tag{3}$$

$$TL \;=\; 10\log_{10}\left[1+\left(\frac{a_n\left(\frac{1}{A}\right)\tan(kL_C)+\tan\,(kL_n)}{2a_d\left(\frac{1}{A}\right)\tan(kL_C)+\tan(kL_c)-1}\right)^2\right] \tag{4}$$

Where A = $\frac{a_c}{a_n}$ is the area ratio, c is the speed of the sound in the medium, k is the wave number, L_c , $L_n$representthe length; d_c, d_n the diameter ; a_c, $a_n$the area cross sections of the cavity and neck respectively and d_d, a_d is the diameter and the cross-sectional area of the duct respectively. The dimensions for modeling a single 1-DOF resonator in this literature was adapted from (Resonator A, Table 1 of [6] shown in Figure 2.4.

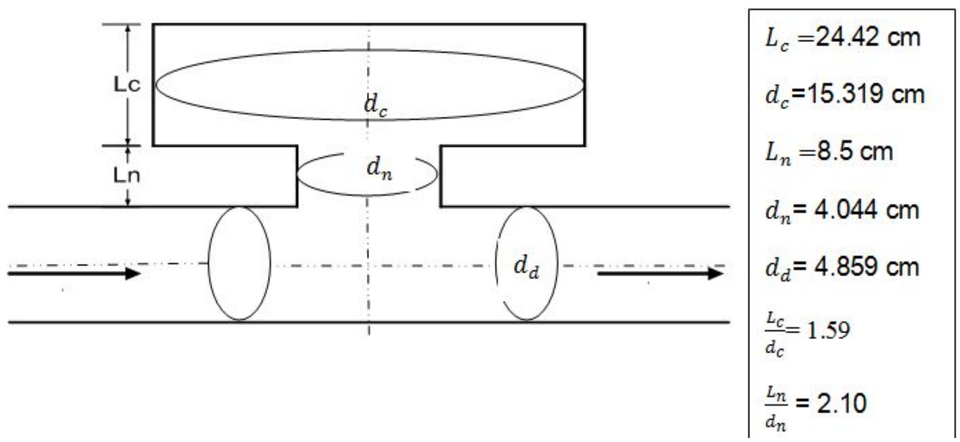


Figure 2.4: Single 1-DOF Helmholtz Resonator

2.5 Adaptation of a Single 2-DOF Helmholtz Resonator

The resonance frequency for a single 2-DOF Helmholtz resonator is represented by Equations. (1) [17] as shown in figure 1.

$$f_{1.2} \;=\; \frac{c}{2\sqrt{\pi}}\sqrt{\left(\frac{\alpha}{V_1}+\frac{\beta}{V_1}+\frac{\beta}{V_2}\right)\pm\sqrt{\left(\frac{\alpha}{V_1}+\frac{\beta}{V_1}+\frac{\beta}{V_2}\right)^2-4\frac{\alpha}{V_1}\frac{\beta}{V_2}}} \tag{1}$$

Now the transmission loss can be calculated using Eq.(2) from [9] with respect to α and β for getting the optimum $\frac{\alpha}{\beta}$or maximum transmission loss.

$$Tl = 20\log_{10}\left|1 + \frac{\alpha}{\left[2a_d\left(ik+\frac{\alpha}{ikV_1}\left(1-\frac{V_2}{V_2+V_1-\frac{V_2V_1k^2}{\beta}}\right)\right)\right]}\right| \quad (2)$$

Where $\alpha = \frac{a_{n1}}{l_{n1}}$, $\beta = \frac{a_{n2}}{ln_2}$, $V_1 = a_{c1}.l_{c1}$, $V_2 = a_{c2}.l_{c2}$ are the volume of the first and the second resonator, a_{n1}, a_{n2} are the area of cross sections of the first and the second neck and l_{n1}, l_{n2} are their respective lengths, where a_d is the cross section area and k is the wave number.

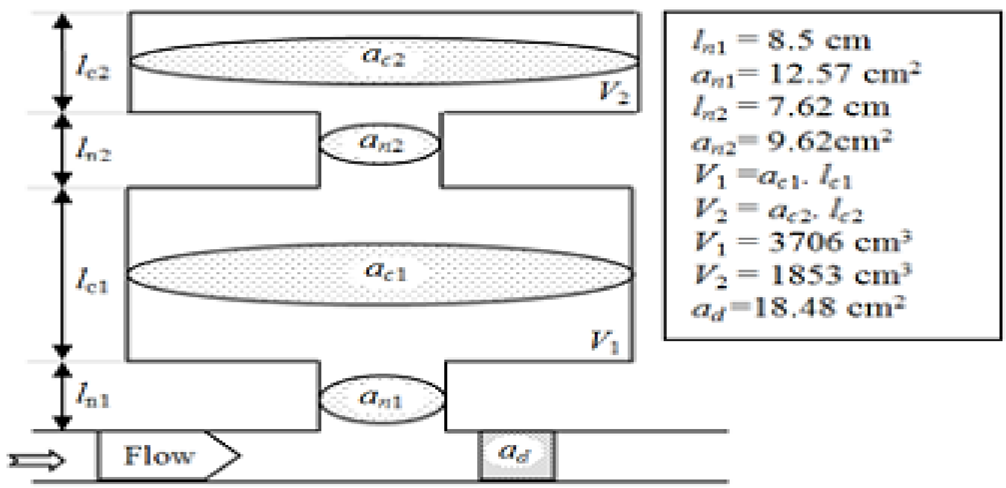


Figure 2.5: Single 2-DOF Helmholtz Resonator

End Correction

The length of the neck is corrected by adding end correction factors considering higher wave propagation effects between the neck and with both the cavity and duct. The end correction of a single 2-DOF Helmholtz resonator is calculated by following steps.

$$l'_{c1} = l_{c1} + \delta_v + \delta_p \tag{3}$$

$$l'_{c2} = l_{c2} + 2\delta_v \tag{4}$$

$$\delta_v = 0.85(1 - 1.25\frac{\mathrm{R_{neck}}}{\mathrm{R_{volume}}}) \tag{5}$$

$$\delta_p = 0.82\frac{R_{neck}}{2} \tag{6}$$

Objective of the Thesis

The primary objective of this thesis was to investigate the spacing between the two identical and different frequency 1-DOF and 2-DOF resonators attached to a duct in 3-dimensional plane by placing them at various relative positions like (closely spaced ($< \frac{\lambda}{4}$); 1st antinode ($\frac{\lambda}{4}$) ; 1st node ($\frac{\lambda}{2}$) where λ is the wavelength of the maximum working frequency to obtain maximum TL at the optimum relative position.

Secondary objective was to investigate the effect of changing both the geometry (length/diameter ratio) of the cavity and neck (keeping the volume of the resonators constant) on the resonating frequency and transmission loss. Finally, an improved system of both 1-DOF and 2-DOF Helmholtz Resonator system in terms of position, geometry and orientation has been proposed and demonstrated in this thesis.

CHAPTER 3

Methodology

3.1 Investigation Criteria

Case I (Relative Positions between the Two Resonators)

The two 1-DOF Helmholtz resonators attached to duct were modeled by placing them at various relative spacing Z (closely spaced; ($<\frac{\lambda}{4}$), 1st antinode ; ($\frac{\lambda}{4}$), 1st node; ($\frac{\lambda}{2}$) distance apart of the wavelength of the maximum working frequency 300 Hz .The dimensions of two resonators were considered exactly same as mentioned in the Figure 1 for a single 1-DOF resonator and the main duct was considered as a square cross section of 4.3*4.3 cm and 1.5 meter long.

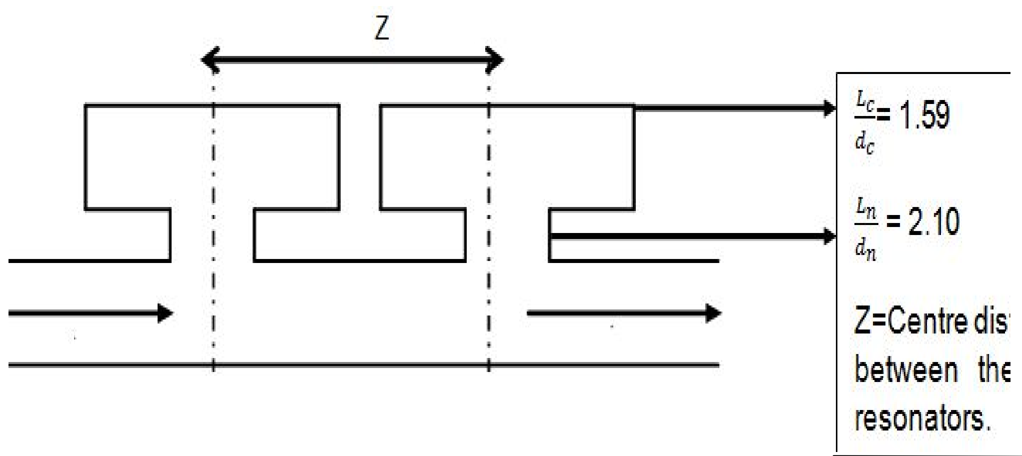


Figure 1.1: Position of the Two Identical Resonators

Case II (Varying the geometry of the cavity keeping the volume constant)

In Case I, the cavity ratio ($\frac{L_c}{d_c}$) used for both the resonators was 1.59. Then it was further changed to 2 and 2.77 keeping the volume of the cavity and dimensions of the cavity constant as mentioned in Table 1 to investigate the resonance frequency and maximum TL. The two resonators in this case were placed at optimum spacing found from Case I.

Cases No. II	L_c	d_c	$\frac{L_c}{d_c}$
a	**17.89**	**17.89**	**1**
b	**20.32**	**15.24**	**1.59**
c	**28.40**	**14.20**	**2**
d	**35.281**	**12.743**	**2.77**

Table 1: Dimensions of the Varying Geometry of the Cavity

Case III (Varying the Geometry of the Neck of the Two Resonators)

In Case I, the neck ratio ($\frac{L_n}{d_n}$) of both the resonators was 2.10. Then, it was further changed to 2.25, 2.5 and 3 as mentioned in Table 2 keeping the volume of the neck constant and using optimum cavity ratio found from Case II to investigate the resonance frequency and maximum TL. The two resonators in this case were also placed at optimum spacing found from Case I.

Cases No III	L_n **(cm)**	d_n **(cm)**	L_n/d_n
a	**8.5**	**4.044**	**2.10**
b	**9.88**	**3.95**	**2.25**
c	**9.54**	**3.80**	**2.5**
d	**10.74**	**3.58**	**3**

Table 2: Dimensions of the Varying Geometry of the Neck

3.2 MATLAB Codes for Analytical Validation

Single 1-DOF Helmholtz Resonator

```
format long;
l=24.42;
lc=8.5;
Av=184.31;
Ac=12.84;
Ap=18.543;
A=Av/Ac;
c=343.7*100;
f=[0:1:300];
TF=zeros (150, 2);
n=1;
for f=0:.5:300
lam= c/f;
k=(2*pi)/lam;
e11=Ac*(tan(k*lc)+(Av/Ac)*tan(k*l));
e22=(2*Ap)*(1-((Av/Ac)*tan(k*l)*tan(k*lc)));
e1=(e11/e22)^2;
TL=10*log10(1+e1);
TF(n,1)=TL;
TF(n,2)=f;
n=n+1;
end
plot(TF(:,2),TF(:,1),'-r');
```

Single 2-DOF Helmholtz Resonator

```
ln1=8.5;
An1=12.57;
ln2=7.62;
An2=9.62;
V1=3706;
V2=1853;
Ad=18.48;
alpha=An1/ln1;
beta=An2/ln2;
c=343.2*100;
f=[0:1:300];
lam=c./f;
k=2*pi./lam
D1=(V2+V1-((V2*V1*k.^2)/beta));
D2=V2/D1;
D3=1-D2;
D4=(alpha/(1i*k*V1))*D3;
D5=1i*k+D4;
D6=2*Ad*D5;
TL=20*log10 (abs(1+alpha/D6));
plot (f, TL,'-r');
```

Chapter 4

Numerical Analysis

4.1Numerical Analysis in ASYS by FEM

ANALYSIS USING FEM

Finite Element Method (FEM) is a numerical approach to obtain approximate solutions to partial differential equations and their system [20]. A complicated problem can be divided into finite number of small and simple elements; these are then solved in relations to each other, hence the overall solution for the whole problem can be evaluated. Thus this method provides a very good means to approximate the modal shape and frequency of the system. The same technique can be applied for harmonic analysis and amplitude variation due to external actuation can be approximated and it can be obtained whether or not the maximum amplitude is concurrent with the modal frequency**.**

ANSYS

ANSYS is engineering simulation software uses FEM to predict system response.

Structural mechanics solution from ANSYS provide the ability to simulate every aspect of a product, including acoustic Harmonic Analysis in ANSYS FLUID The ANSYS Mechanical (APDL) software suite is used for this acoustic harmonic analysis.

HARMONIC ANALYSIS

Any sustained cyclic load will produce a sustained cyclic response (a harmonic Response) in a structural system. Harmonic response analysis gives the ability to predict the sustained dynamic behavior of your structures, thus enabling you to verify whether or not your designs will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations.

Harmonic response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary sinusoidally (harmonically) with time. The idea is to calculate the structure's response at several frequencies and obtain a graph of some response quantity (usually displacements) versus frequency. "Peak" responses are then identified on the graph and stresses Reviewed at those peak frequencies.

This analysis technique calculates only the steady-state, forced vibrations of a Structure. The transient vibrations, which occur at the beginning of the Excitation, are not accounted for in a harmonic response analysis.

ACOUSTIC HARMONIC ANALYSIS

The analysis calculates the pressure distribution in the fluid due to a harmonic (sinusoidally varying) load at the fluid-structure interface. By specifying a frequency range for the load, you can observe the pressure distribution at various frequencies. You can also perform modal and transient acoustic analyses. (See the Structural Analysis Guide for more information on these types of analyses.)

4.2 PROCESSES INVOLVED IN ACOUSTIC HARMONIC ANALYSIS IN ANSYS

The procedure for a harmonic acoustic analysis consists of three main steps:

- Build the model.
- Apply boundary conditions and loads and obtain the solution.
- Review the results.

4.3 DETERMINATION OF TRANSMISSION LOSS OF A MUFFLER BY HARMONIC ACOUSTIC ANALYSIS WITHOUT COUPLING:

Test study:

Problem specification

Radius of chamber 0.0766445 m

Length of the chamber 0.2032 m

Radius of the inlet and outlet pipes 0.0174625m

Length 0.104775 m

Frequency range [0 to 3000 Hz]

FLUID221, a higher order 3-D 10-node solid element that exhibits quadratic pressure behavior used.

BUILDING THE MODEL IN MECHANICAL APDL:

FLUID221

FLUID221 is a higher order 3-D 10-node solid element that exhibits quadratic pressure behavior, and is used for modeling the fluid medium and the interface in fluid-structure interaction problems. Typical applications include sound wave propagation and submerged structure dynamics. The governing equation for acoustics, namely the 3-D wave equation, has been discredited, taking into account the coupling of acoustic pressure and structural motion at the interface. The element has four degrees of freedom per node: translations in the nodal x, y and z directions, and pressure. The translations are applicable only at nodes that are on the interface. Acceleration effects such as in sloshing problems may be included.

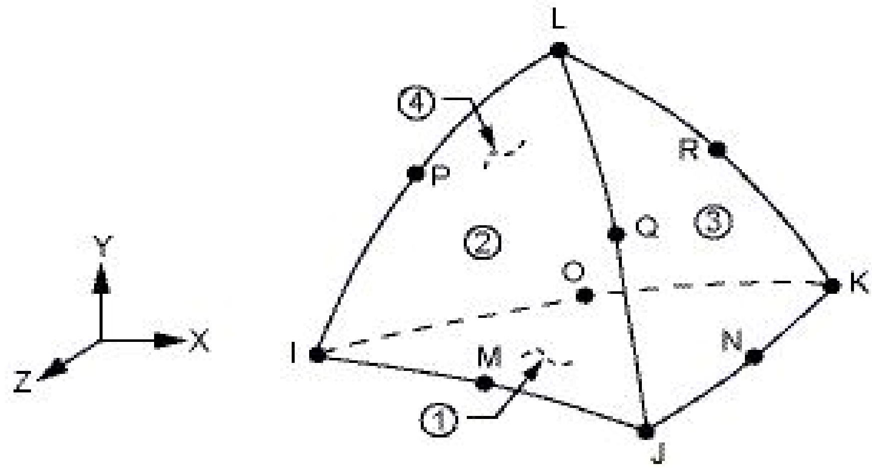


Figure 4.3.1: Fluid 221 Geometry

The element has the capability to include damping of sound absorbing material at the interface as well as damping within the fluid. The element can be used with other 3-D structural elements to perform unsymmetric or damped modal, full harmonic , and full transient method analyses (see the description of the TRNOPT command). When there is no structural motion, the element is also applicable to modal, and reduced harmonic analyses. See Acoustics in the Mechanical APDL Theory Reference for more details about this element. See FLUID220 for a hexahedral option and FLUID30 for a lower order option.

This example problem demonstrates the use of FLUID221 to predict the acoustic transmission loss of a muffler.

```
/batch,list

/title, Transmission Loss of Muffle

/show,png

/nopr

/PREP7

rho=1.2041            ! air mass density

c0=343.24             ! air sound speed
```

z0=rho*c0

freqE=3000 ! highest working frequency

wave=c0/freqE ! wavelength at the highest frequency

p=1

vn=-p/(rho*c0) ! normal velocity excitation

Define element and materials

et,1,221,,1 ! tet uncoupled element

mp,dens,1,rho ! material

mp,sonc,1,c0

Create the model

rapipe=0.0174625

lpipe=0.104775

rchamb=0.0766445

lchamb=0.2032

cylind,0,rapipe,0,lpipe,0,180

cylind,0,rchamb,lpipe,lpipe+lchamb,0,180

cylind,0,rapipe,lpipe+lchamb,2*lpipe+lchamb,0,180

vsel,all

vglue,all

Mesh the geometry

h=wave/10 (10 elements/per wavelength)

esize,h

type,1

mat,1

vmesh,all

nummrg,all (Merging all the meshed elements)

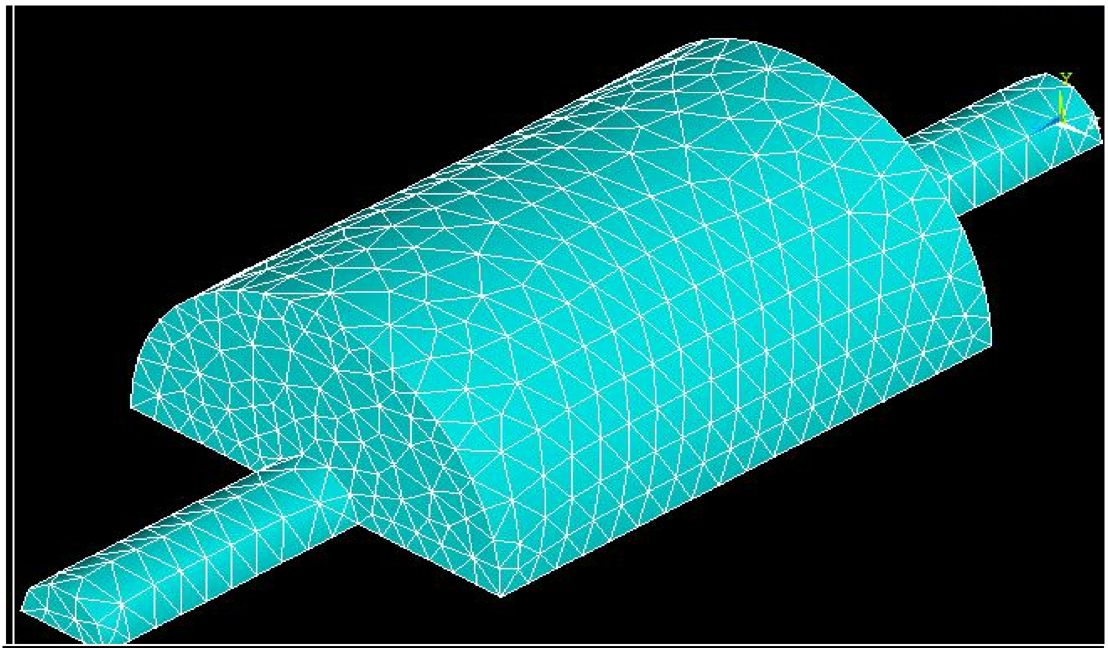


Figure 4.3.2: Acoustic Model of the muffler in ANSYS

Define excitation and boundary conditions on inlet and outlet port

nsel,s,loc,z,0 (nodes on inlet)

Define loads at inlet

Preprocessors > loads > Define loads > apply > Fluid/ANSYS > field surface > on nodes > pick all

sf,all,shld,vn (normal velocity)

sf,all,impd,z0 (impedance boundary on inlet)

nsel,s,loc,z,2*lpipe+lchamb (nodes on outlet)

Define loads at outlet:

Preprocessors > loads > Define loads > apply > Fluid/ANSYS > field surface > on nodes > pick all like inlet

sf,all,inf ! radiation boundary on outlet

alls

fini

Perform solutions

/solu

antype,harmic

hropt,auto

kbc,1

harf,0,freqE

nsub,60 (50 Hz interval with 60 steps)

solve

finish

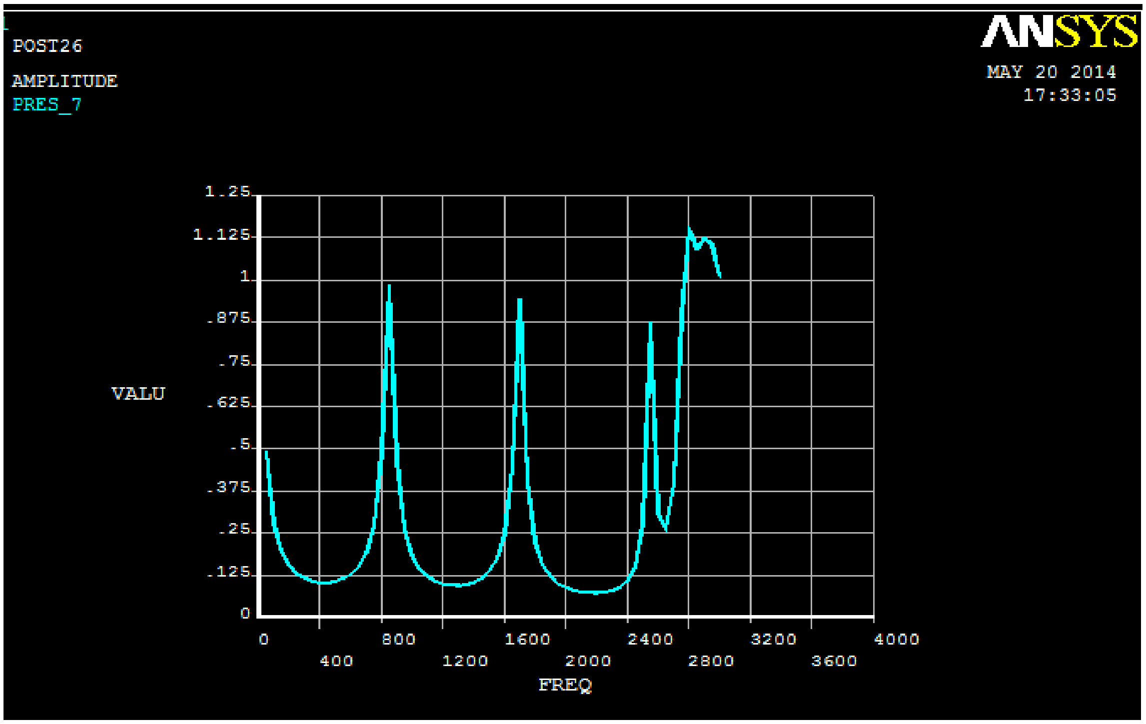


Figure 4.3.3: Sound Pressure drop

Processing the Results

The sound pressure value found at each frequency is converted to dB by following code in MATLAB

pressure(:,3)=20*log10(pressure(:,2))-20*log10(20*(10^-6));

TL(:,4)=94-pressure(:,3);

plot(freq(:,1),**TL**(:,4),'+');

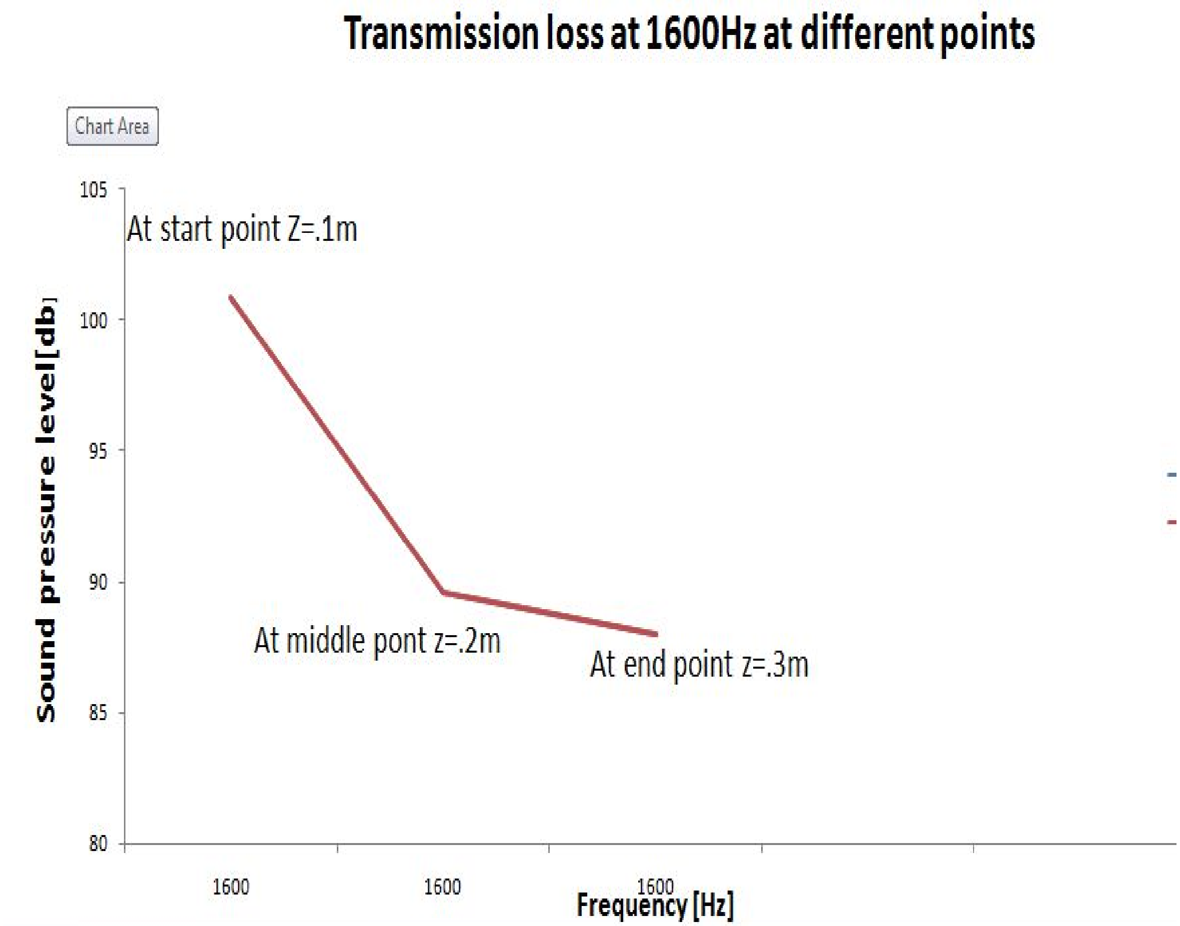


Figure 4.3.4: Transmission Loss of the Muffler at 1600HZ

Results:

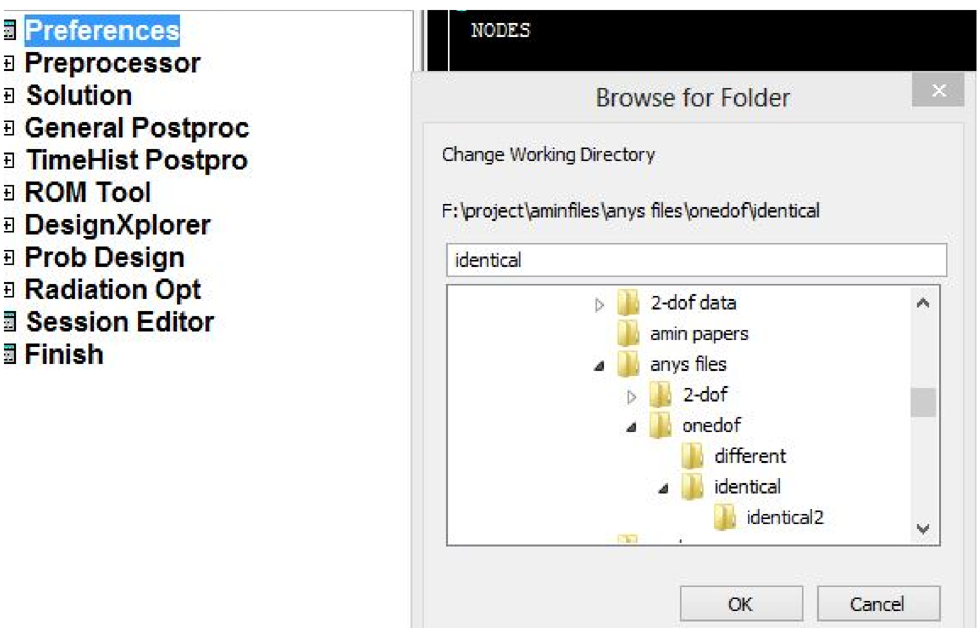
Transmission loss of (100.827-87.85)= approx.13 dB at 1600 Hz at the inlet & outlet node of the muffler without the Helmholtz resonator which can be increased further by using Helmholtz resonator in the muffler.

4.4 NUMERICAL ANALYSIS FOR DETERMINING RESONANCE FREQUENCIES AND TRANSMISSION LOSS OF HELMHOLTZ RESONATOR ATTACHED TO A DUCT

4.4.1 PROCEDURES FOR TWO IDENTICAL 1-DOF HELMHOLTZ RESONATORS IN MECHANICAL APDL:

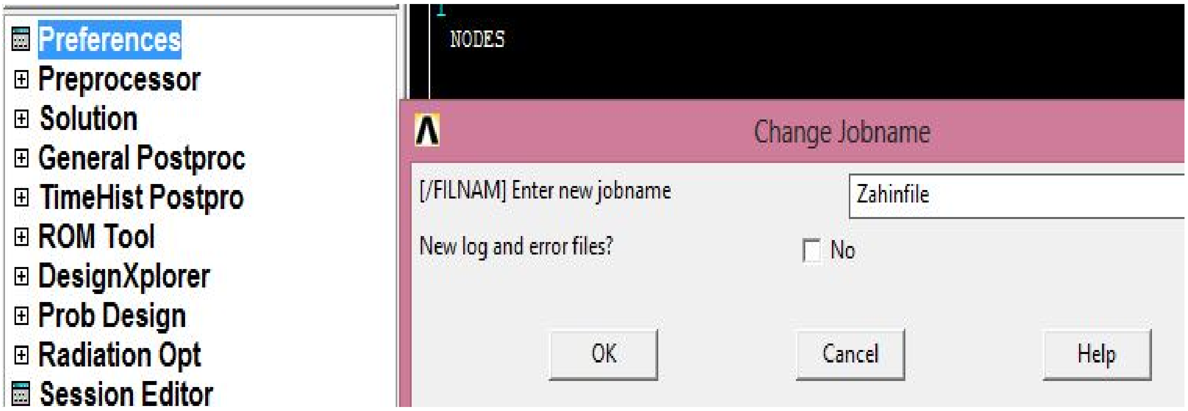
1. Select Directory:

Select directory f>project>amin files>ansys files>1-dof



2. Select Jobname

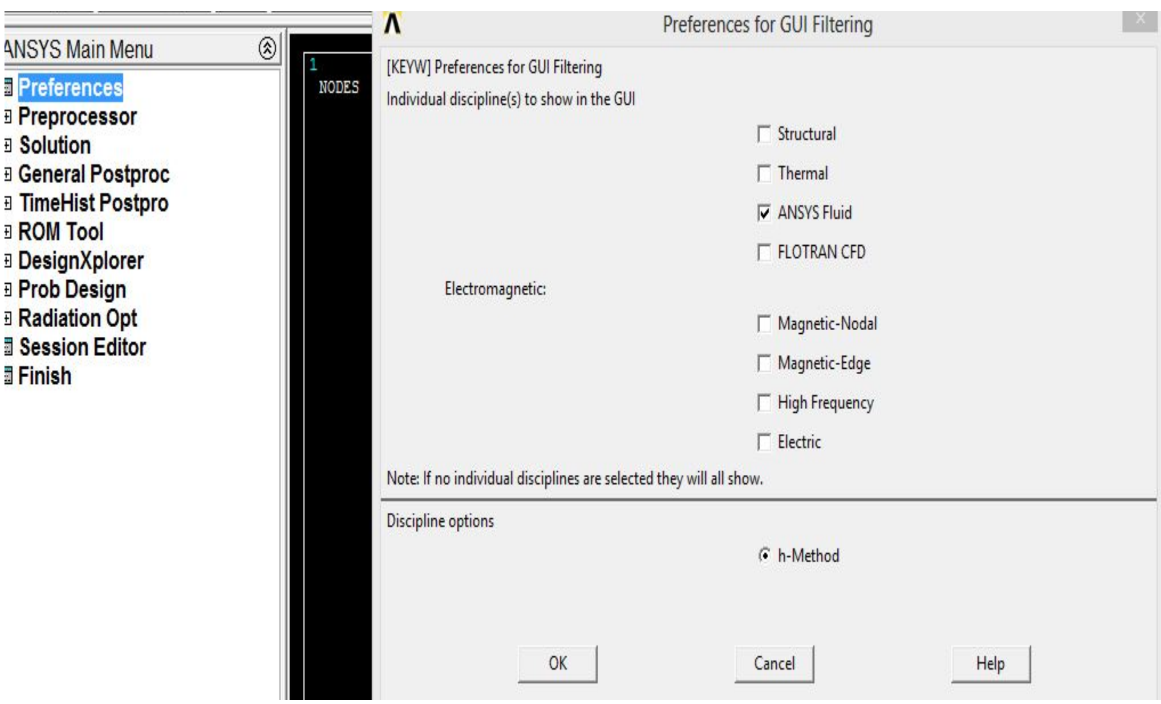
Select Jobname:2-doflessthanlamda>click ok>save.db



3. Selecting Preferences

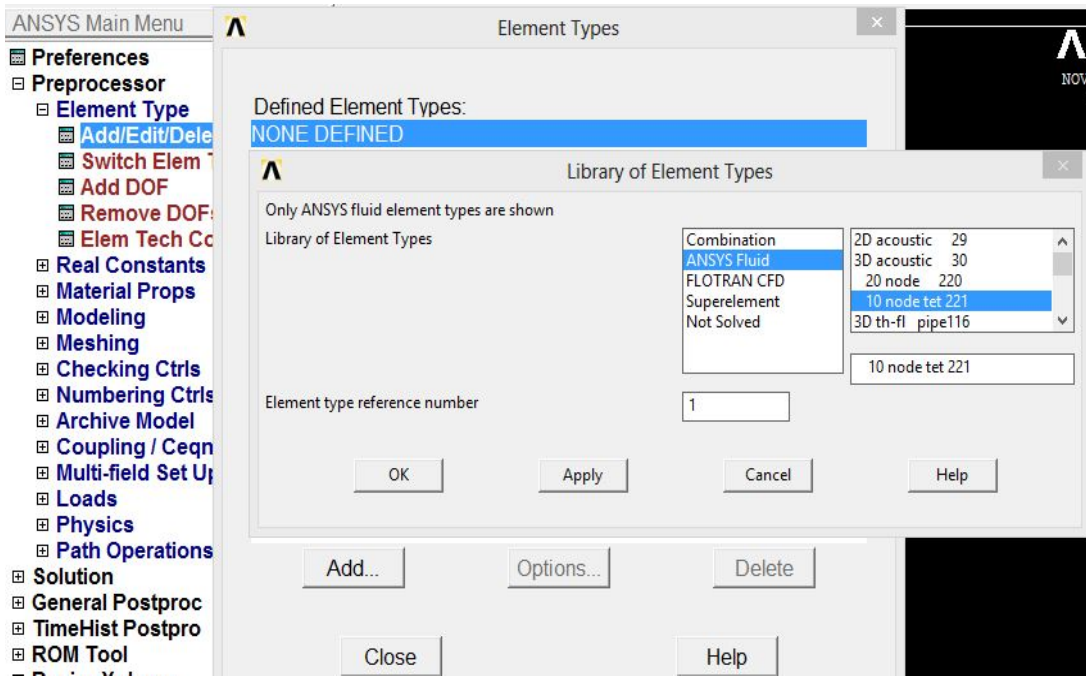
Firstly, the type of simulation to be done is selected. Steps involved are:

Opening the GUI>Preferences>ANSYS Fluid

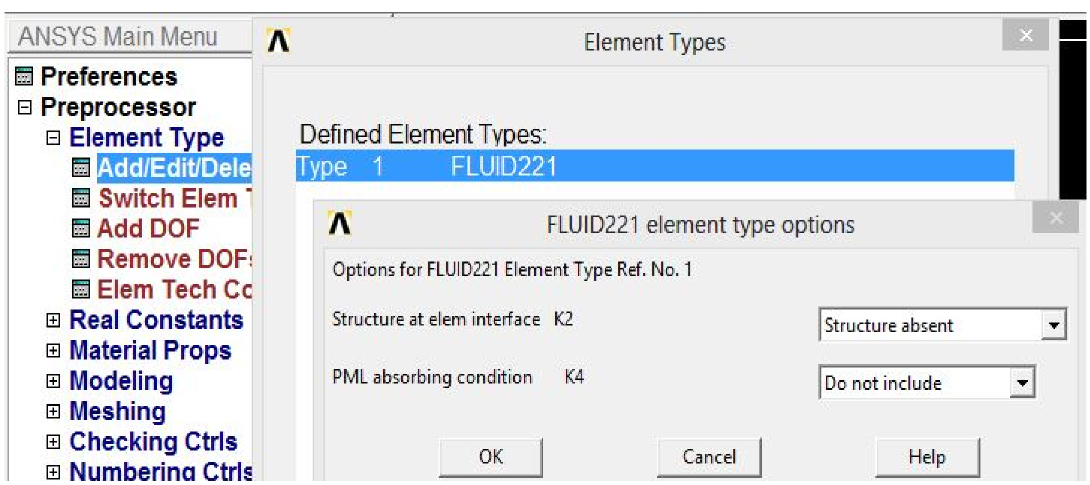


4. Select element types

Secondly, Select element types > add > ANSYS Fluid > select 10 node tet fluid 221



And then Go to Options > select structure absent



5. Define material properties

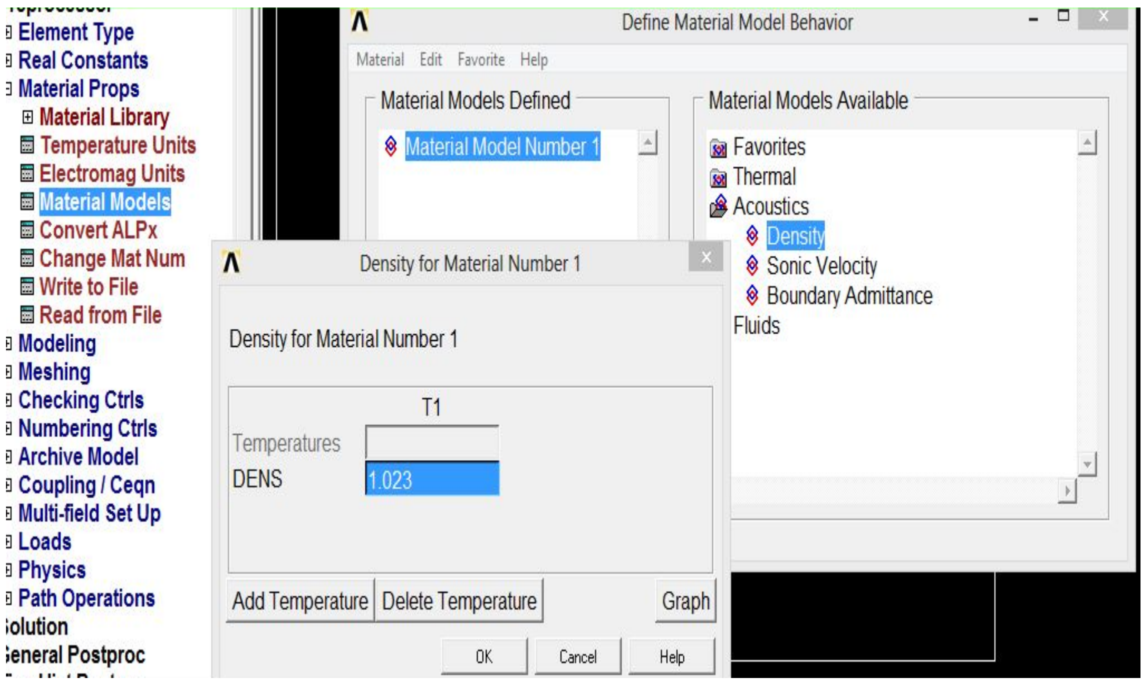
Material properties (such as Density, Sonic Velocity) are to be defined.

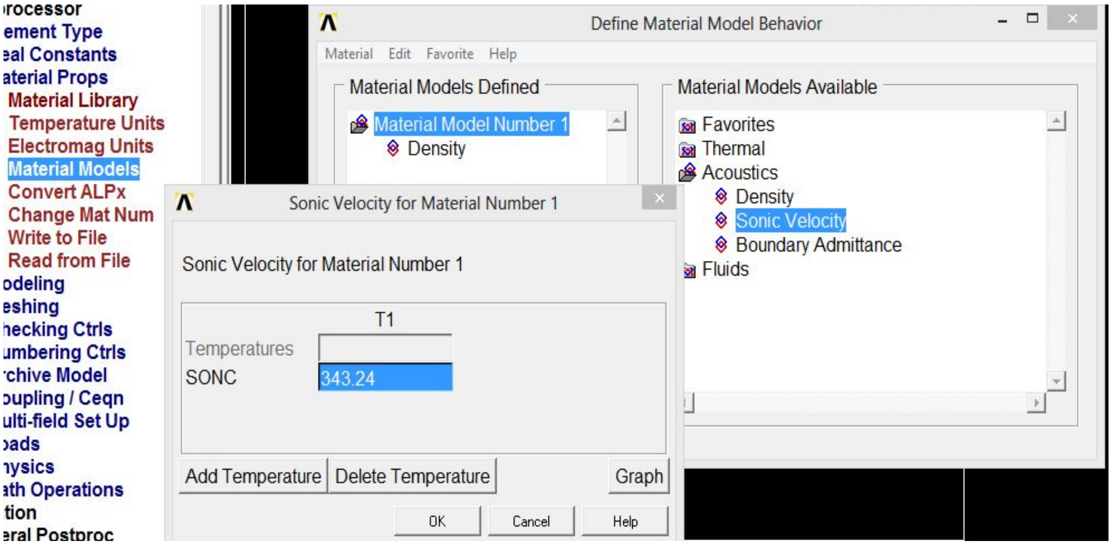
Steps involved:

Main Menu>Preprocessor>Material Properties>Material model>Acoustics

Density=1.2041

Sonic velocity=343.24



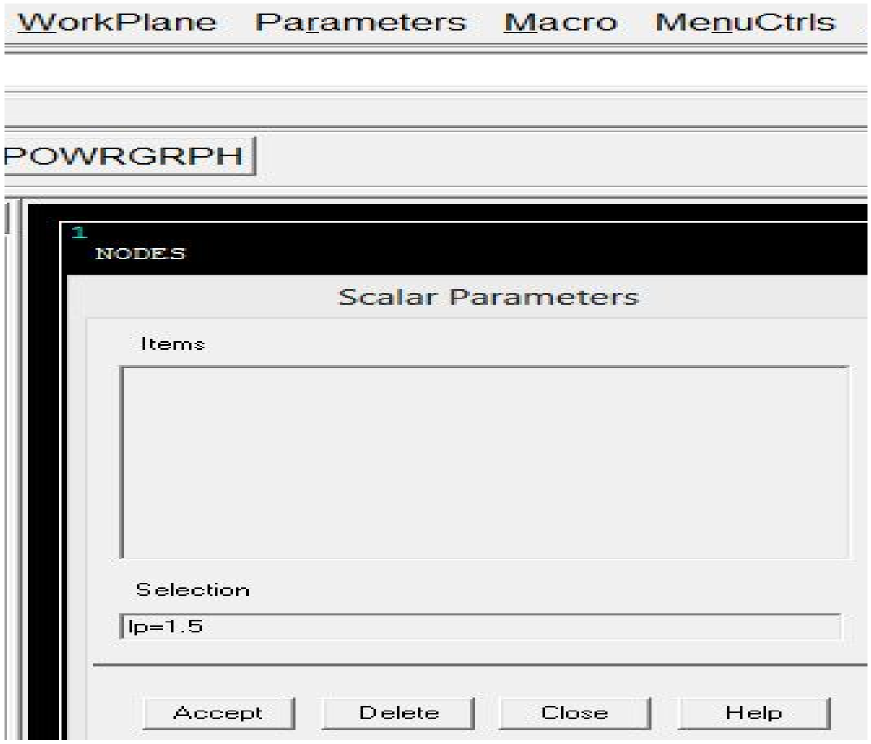


6. Defining Parameters

Go to Parameters on the top >scalar parameters > type each of the parameter with dimensions > click accept for each

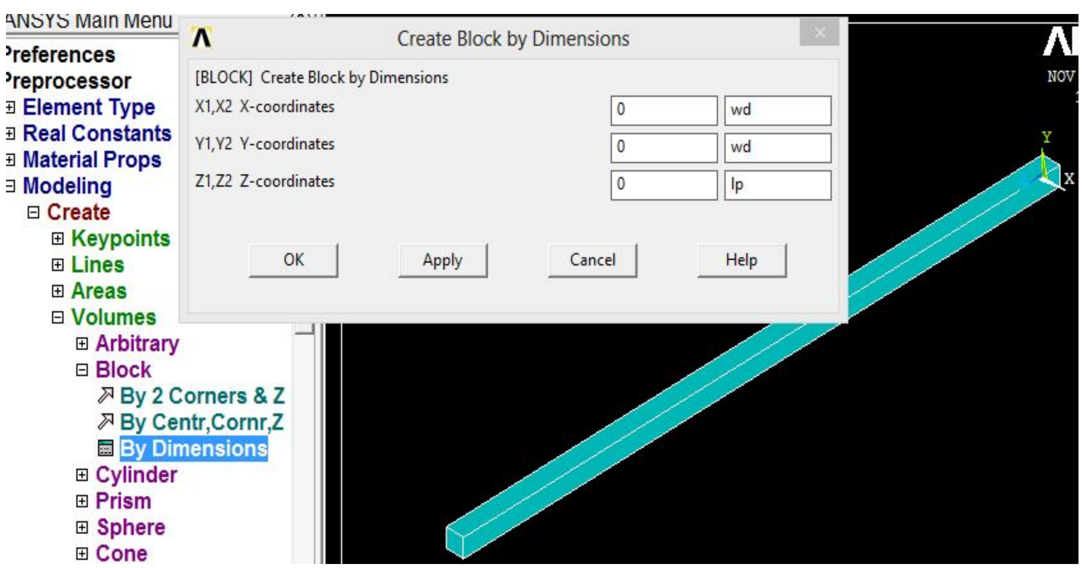
The parameters will be saved for further use

Lp= 1	Rc1=.0762
wd=.0242	Ln2=.0762
Ln1=.085	Rn2=.0175
Rn1=.02	Lc2=.1016
Lc1=.2032	Rc2=.0762



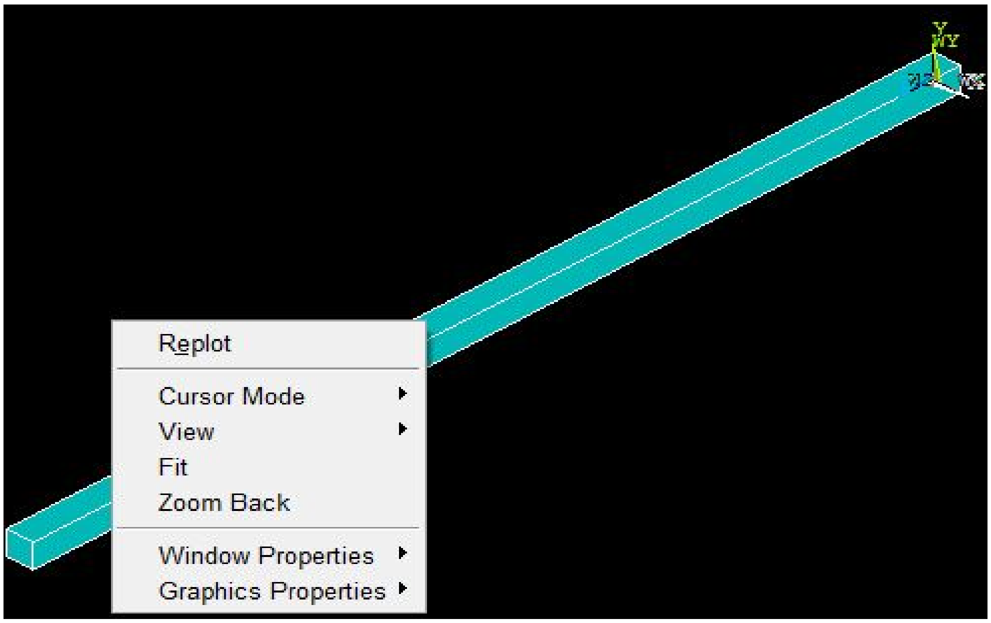
7. Creating Duct

Modeling > create > volumes > by dimensions > blocks > coordinates of x and y both (0, wd) and z= (0, lp)



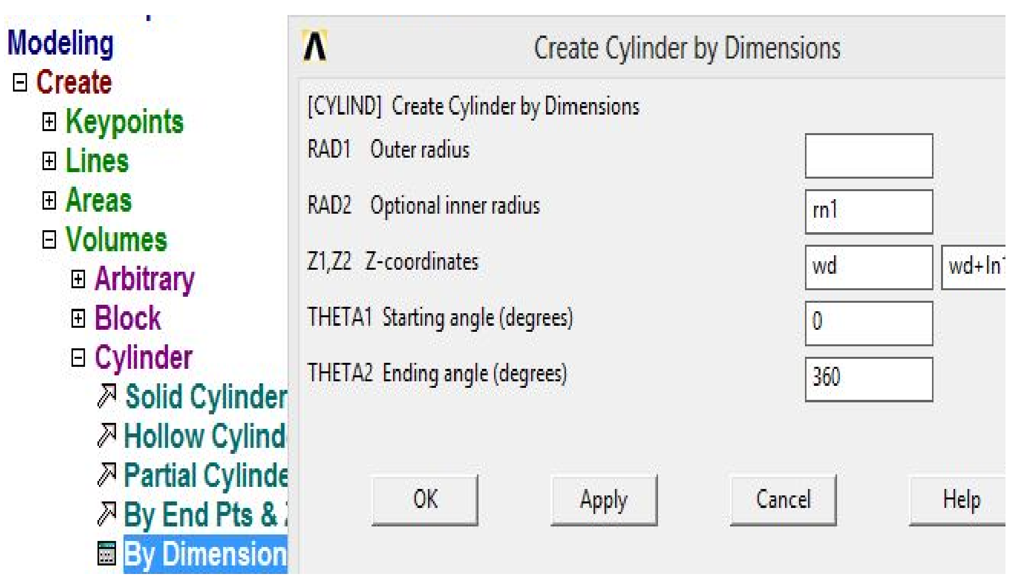
8. Changing working plane

Changing working plane by right click on screen > graphics properties > windows plane offset > Degrees xy, xz, zx middle > make degree to 90 and select x-clockwise first one



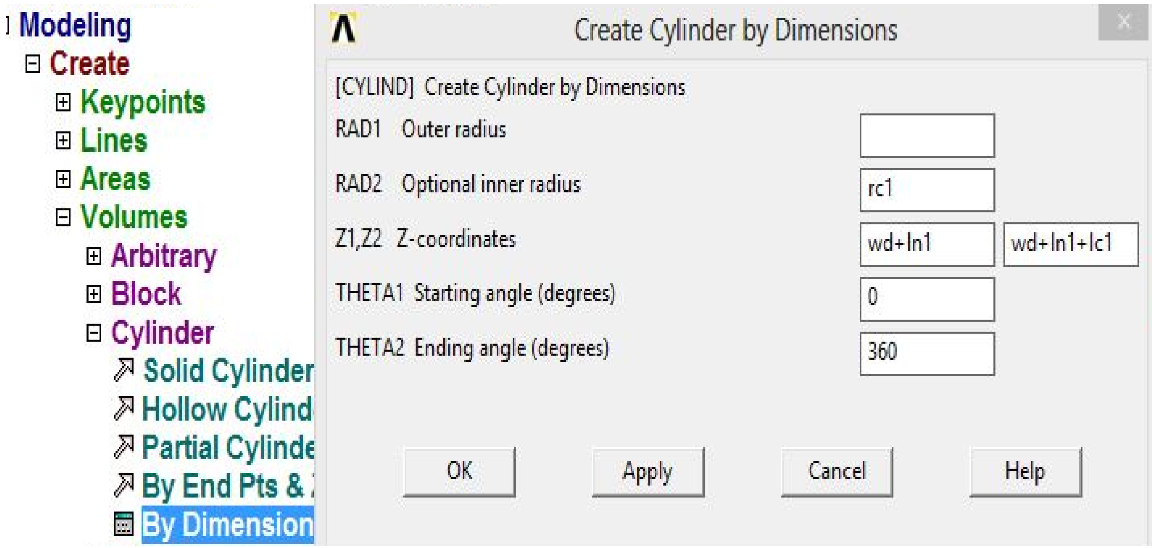
9. Modeling of Two Identical 1-DOF Resonator

9.1 First neck: Steps Modeling > create > volumes > cylinder > by dimensions > optional inner radius RAD2=rn1, Z coordinates z1=wd, z2=wd+ln1



9.2 1st Cavity

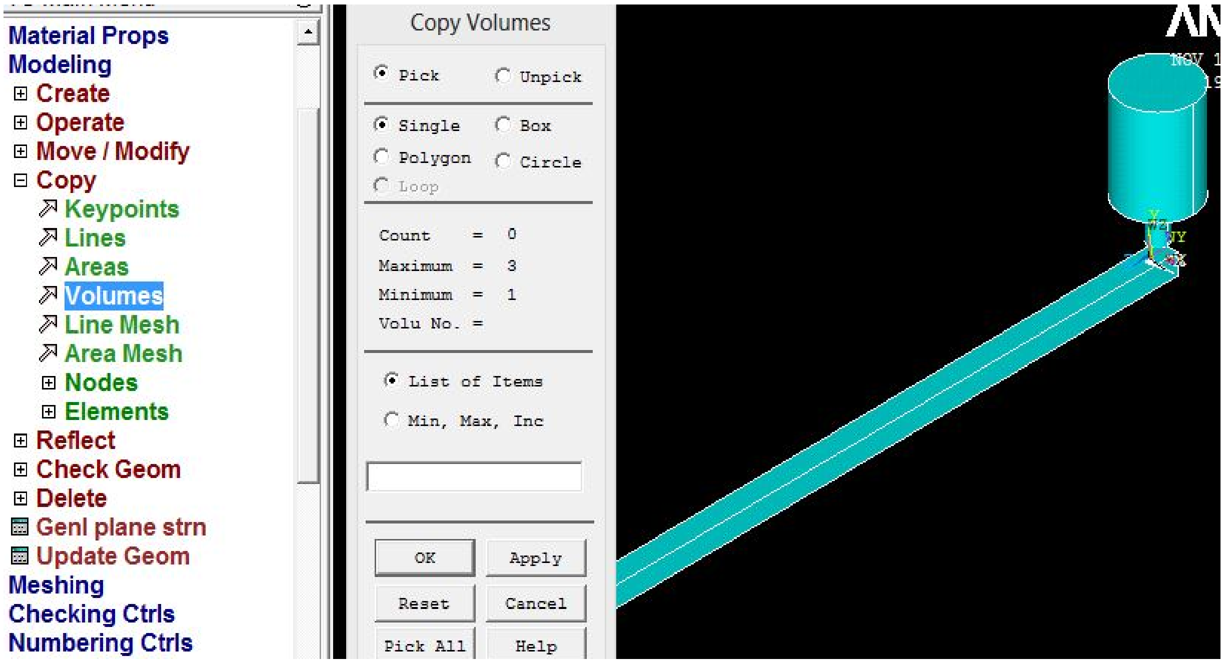
Similarly put RAD2=rc1, z1=wd+ln1, z2=wd+ln1+lc1



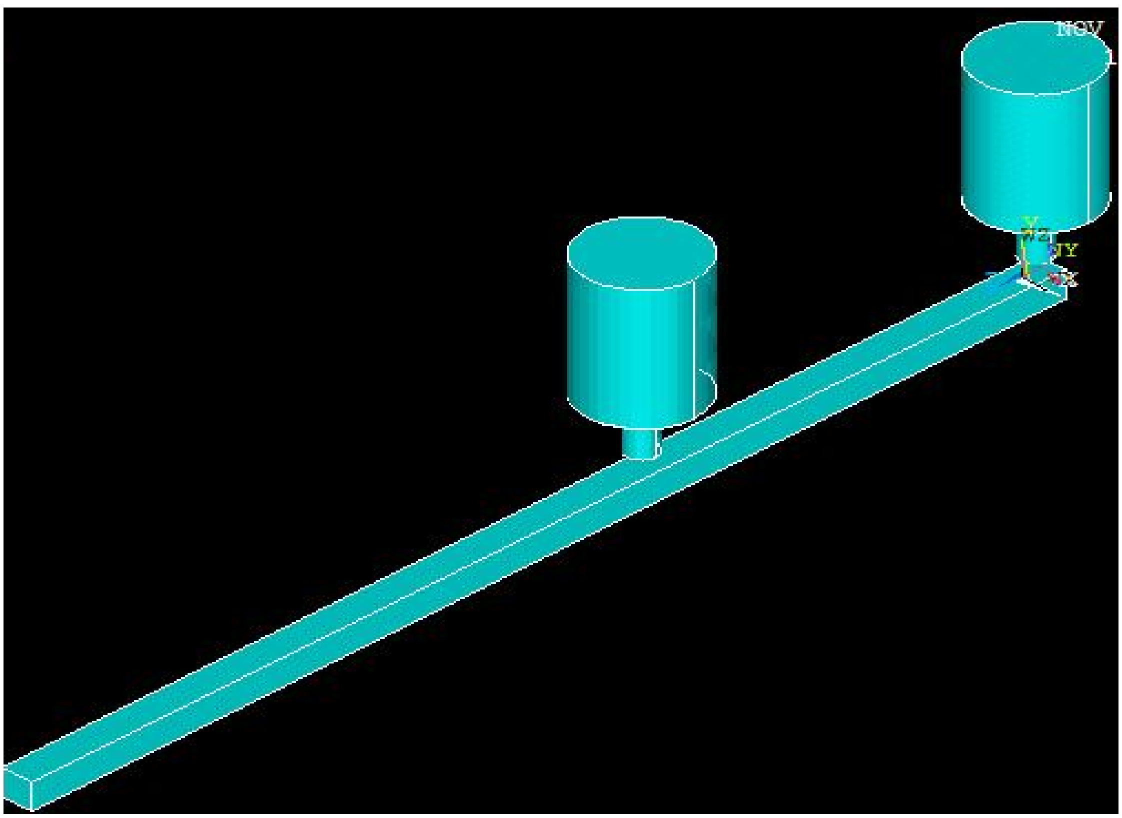
10. Copying the 2nd resonator for Identical Resonators

In case of identical resonator, second resonator are modeled by copying the 1st resonator by following steps

Modeling > copy > volumes >select the resonator manually > select pick all > DZ =0.572 > number of copies 2 > click ok

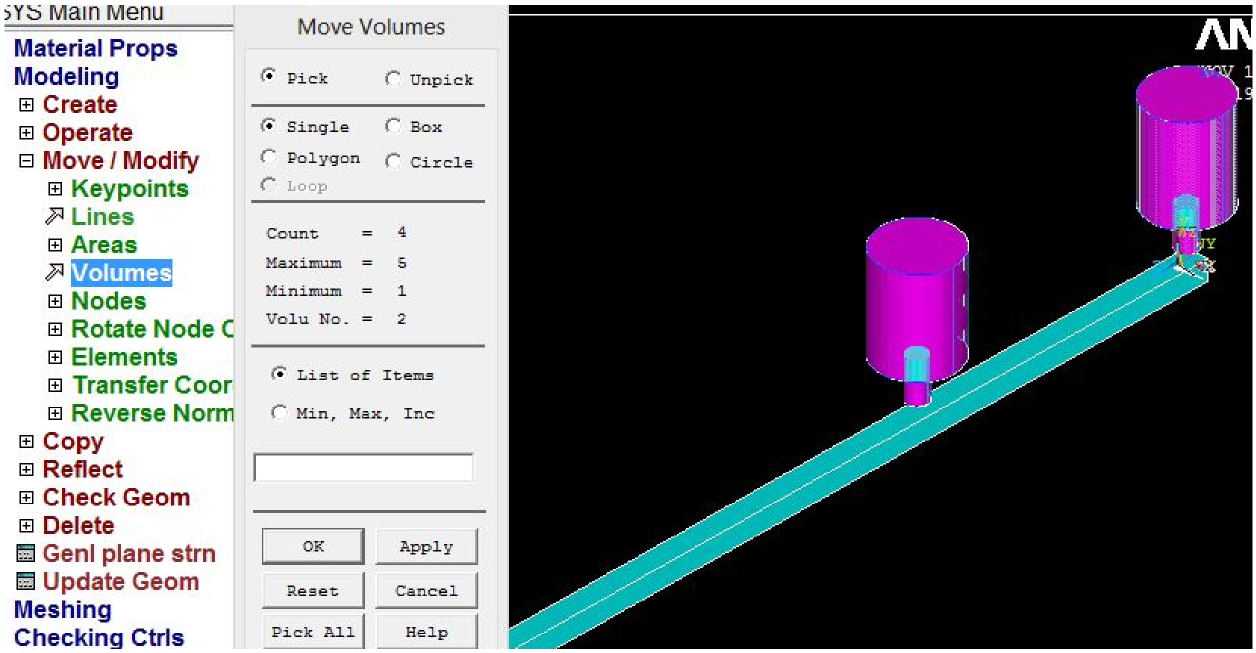


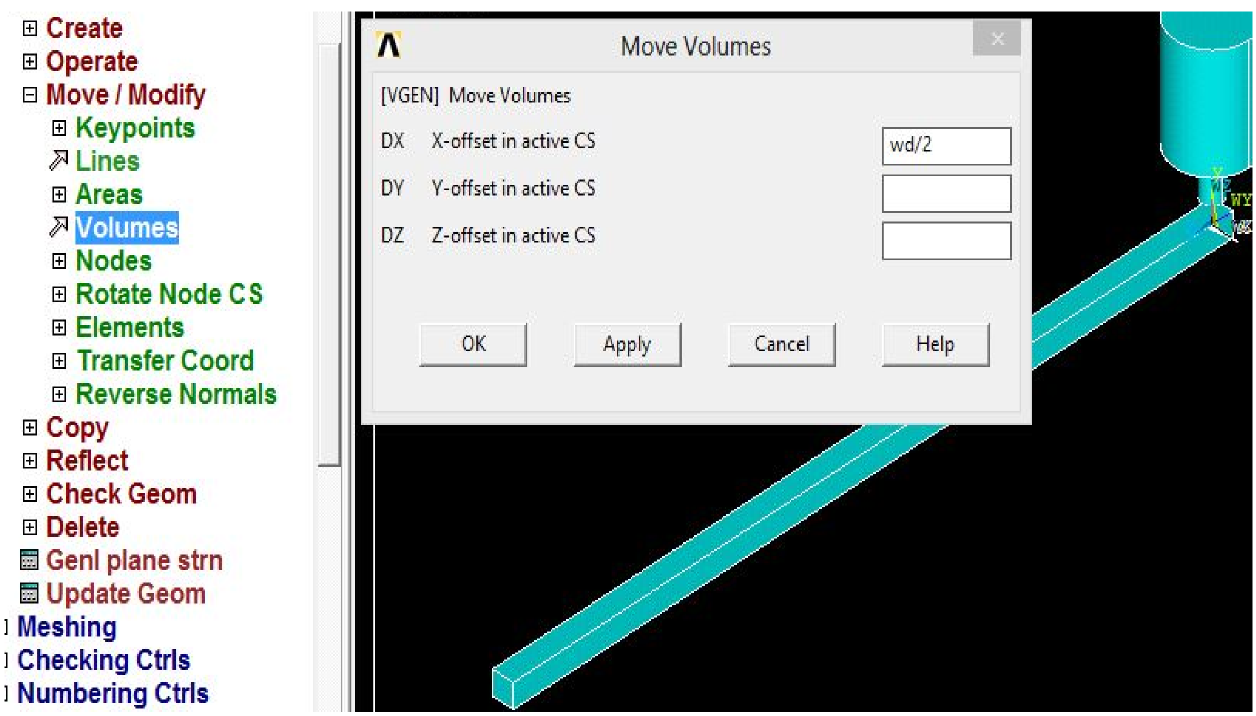
After copying models look like following

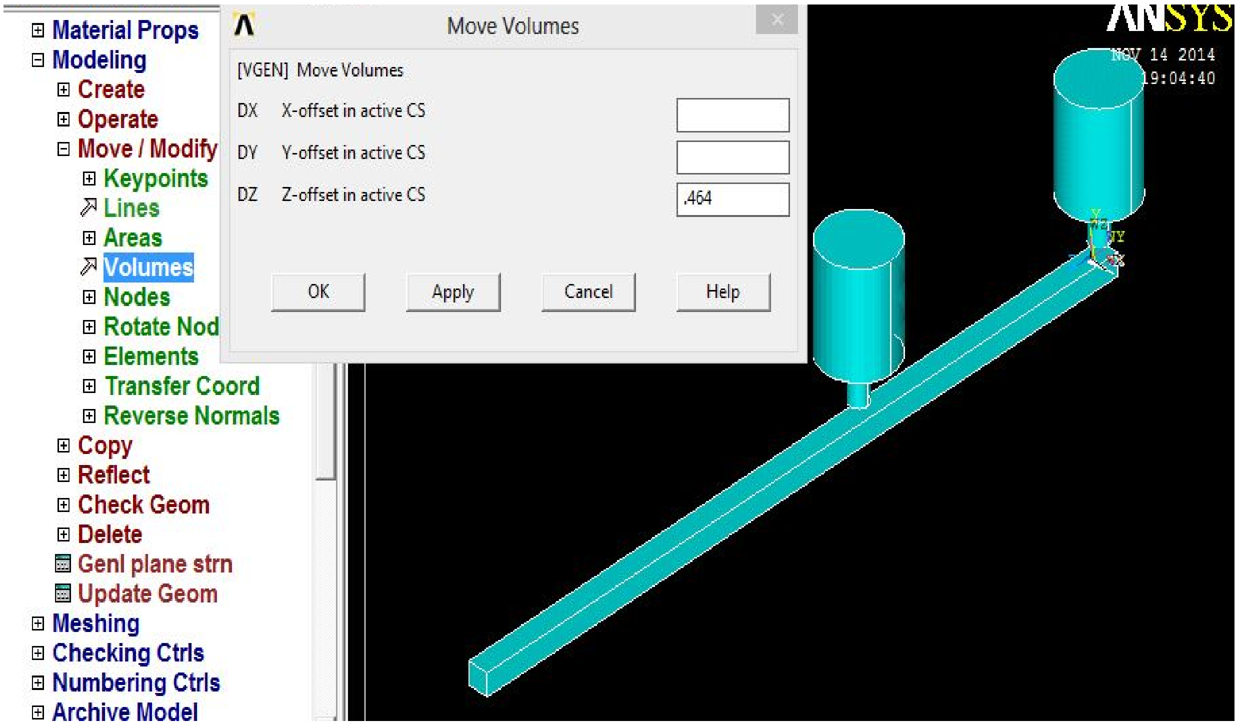


11. Move the Resonators

Modeling > move/ modify > volumes > select both the resonators manually > pick all > DZ=0.464, DX=wd/2





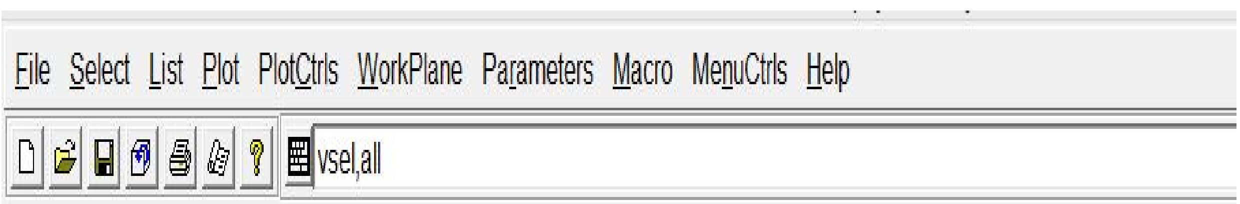


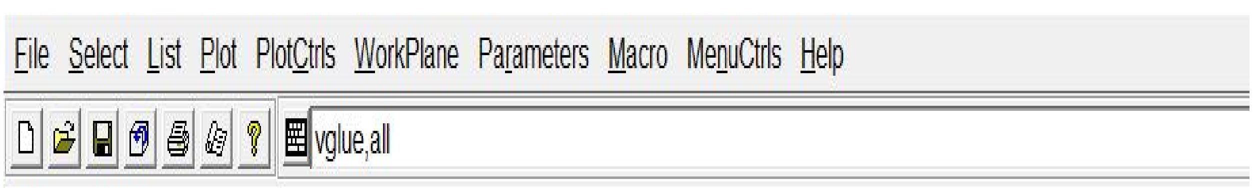
12. Select the entire volume and making glue of the duct and the resonators

Then type in command window

vsel,all

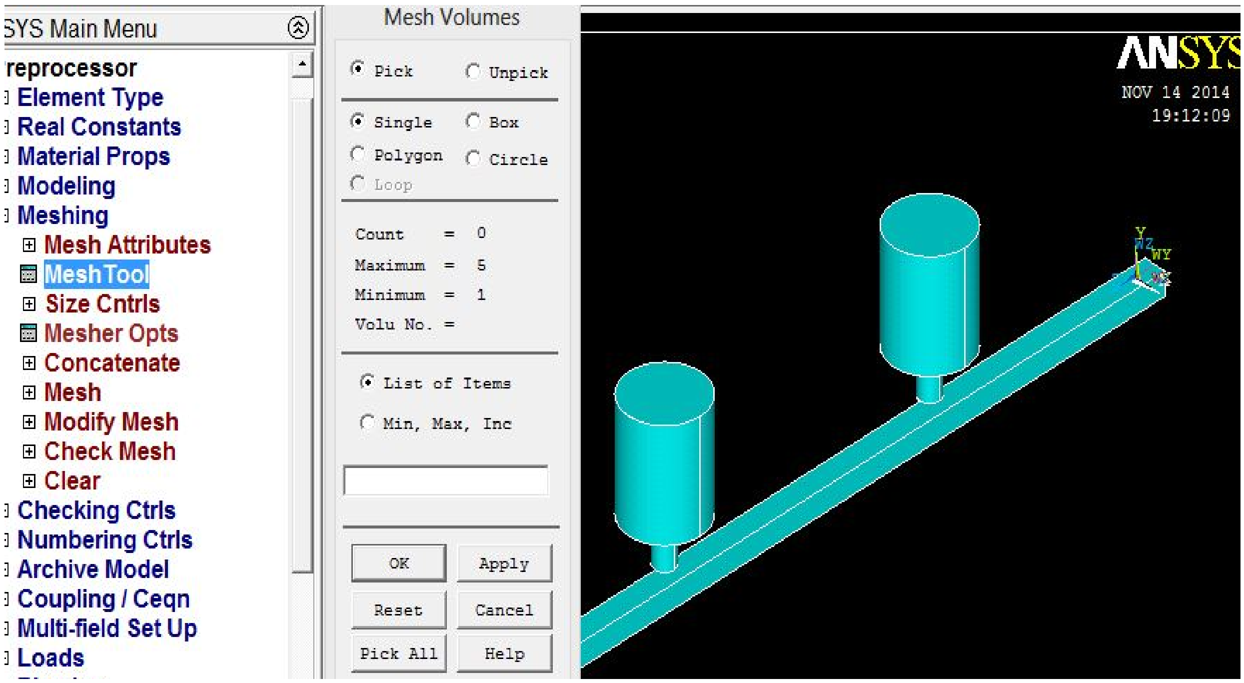
vglue,all





14. Meshing the models

Preprocessor > meshing > mesh tool > Set smart size 6 > size controls > Global > set > element edge lengthfor 300Hz=0.11>mesh>volumes >click on mesh>pick all



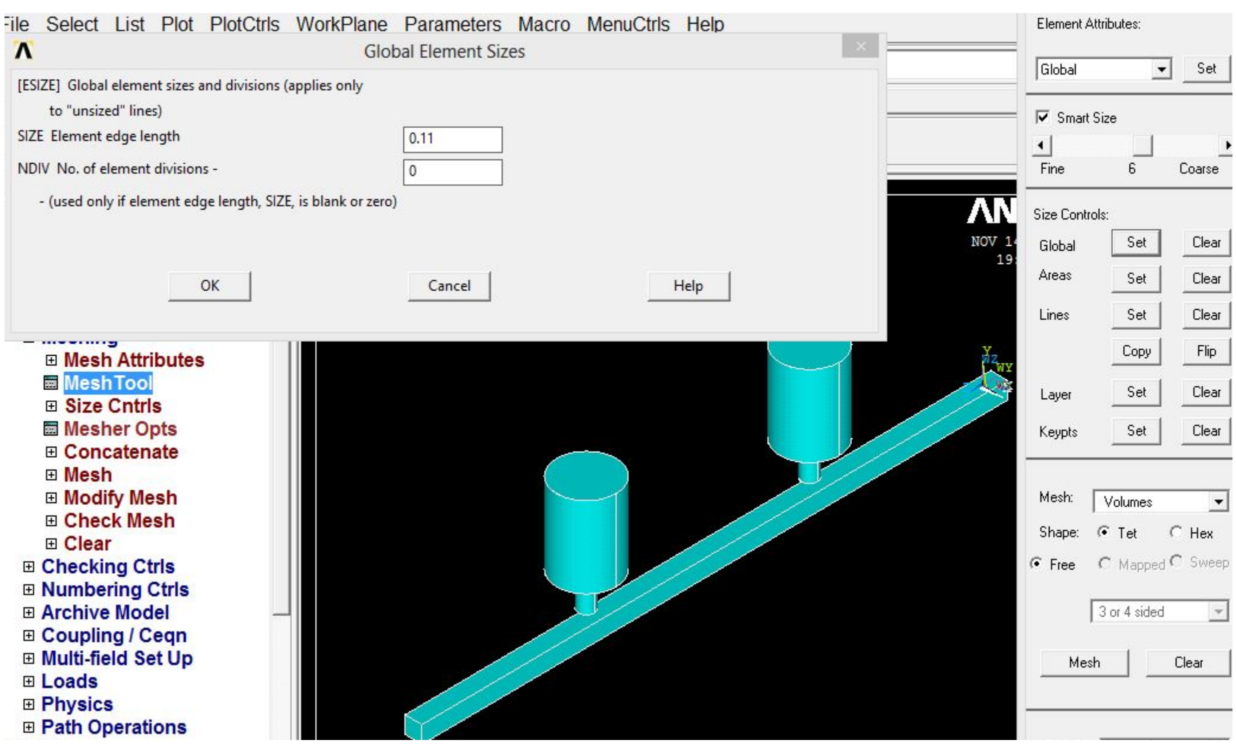
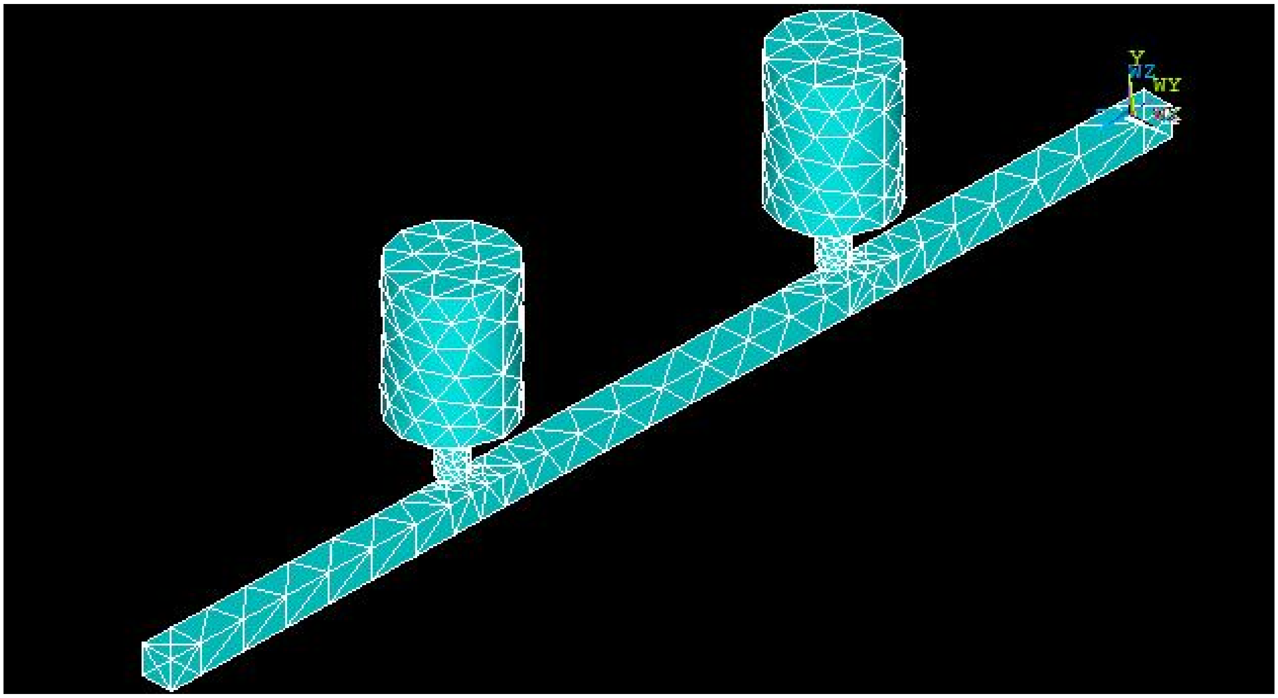
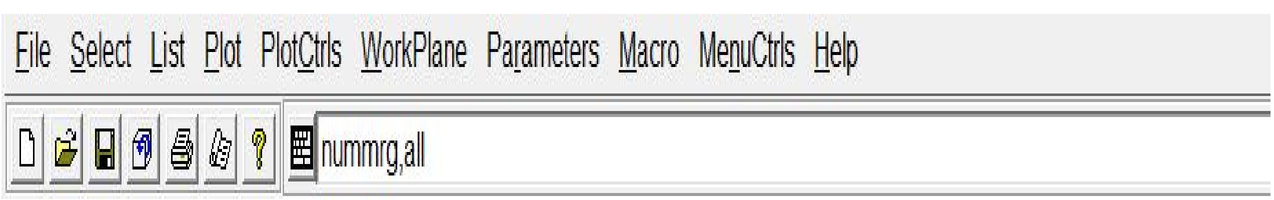


Figure: Meshed Model



15. Merging the elements

Type in command window nummrg,all

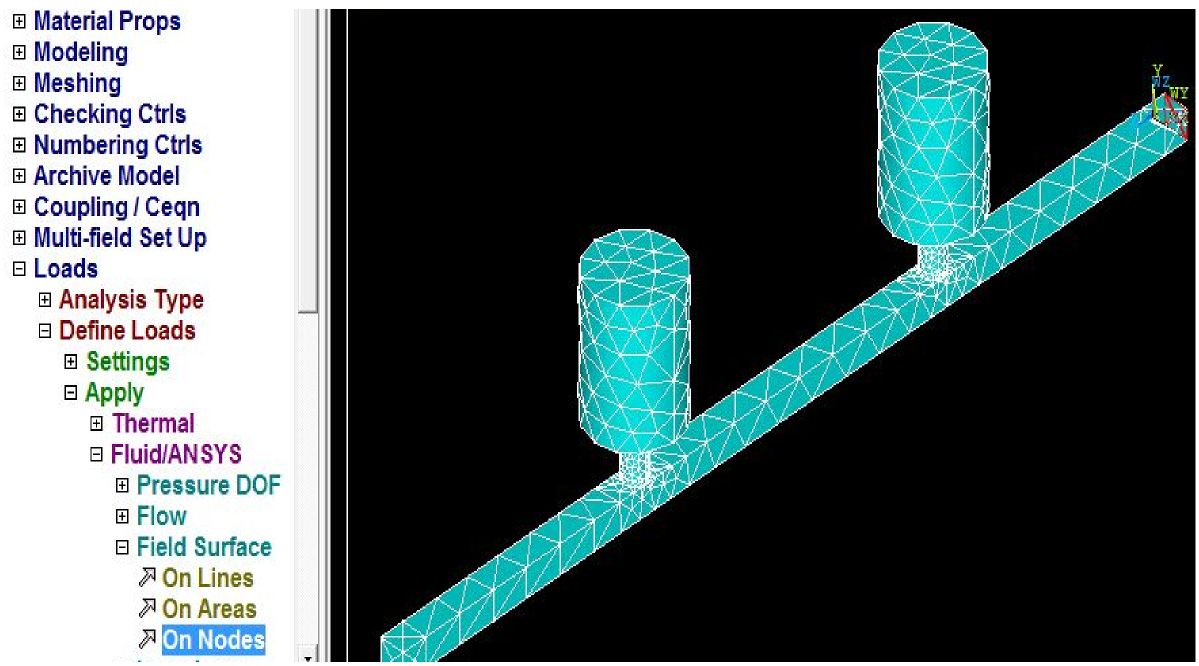


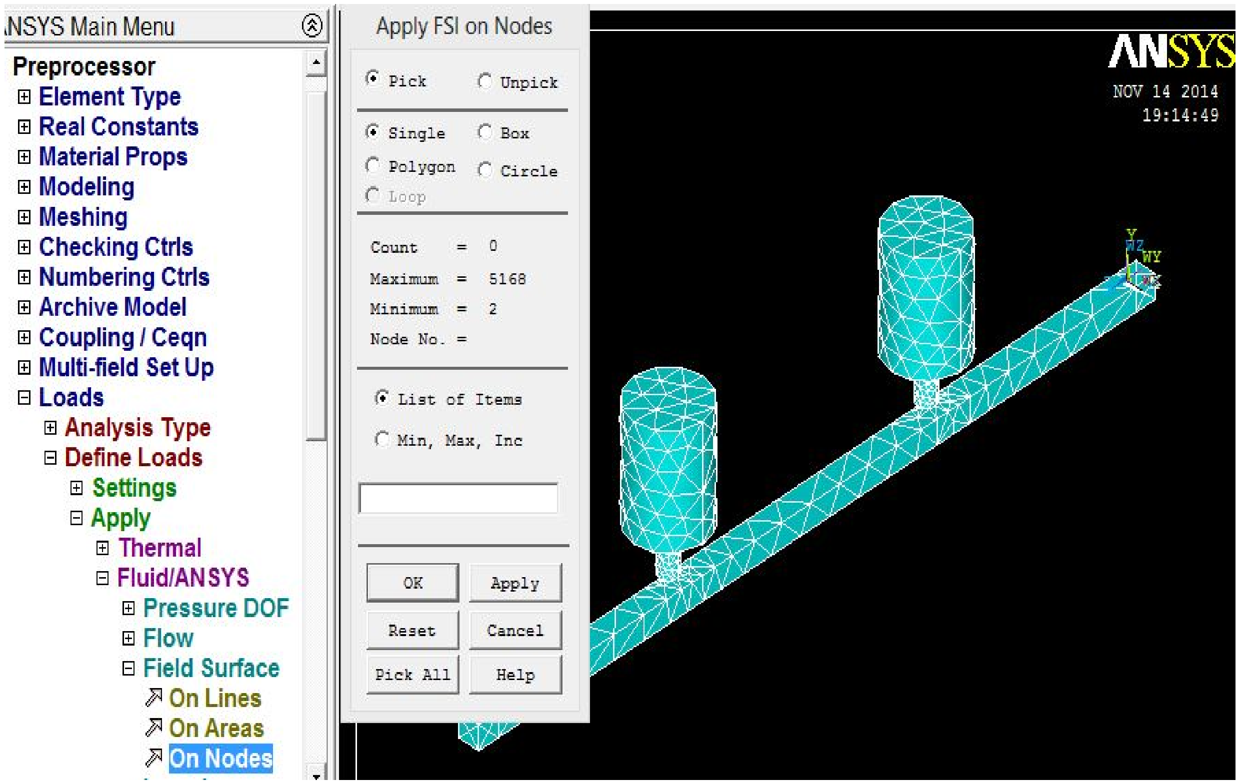
16. Selecting Nodes at inlet

Click on select on top > entities > nodes > by location > Z coordinates=0 > ok

17. Define loads at inlet

Preprocessors > loads > Define loads> apply > Fluid/ANSYS >field surface>on nodes>pick all





18. Applying excitation on the boundary

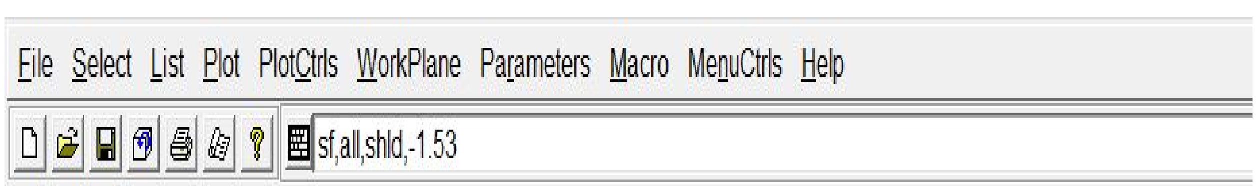
Excitation is applied on the inlet and outlet boundary in the form of normal velocity as follows:

vn=- sound pressure/ (density* sonic velocity)

For 150 dB, vn=-1.53 which is applied

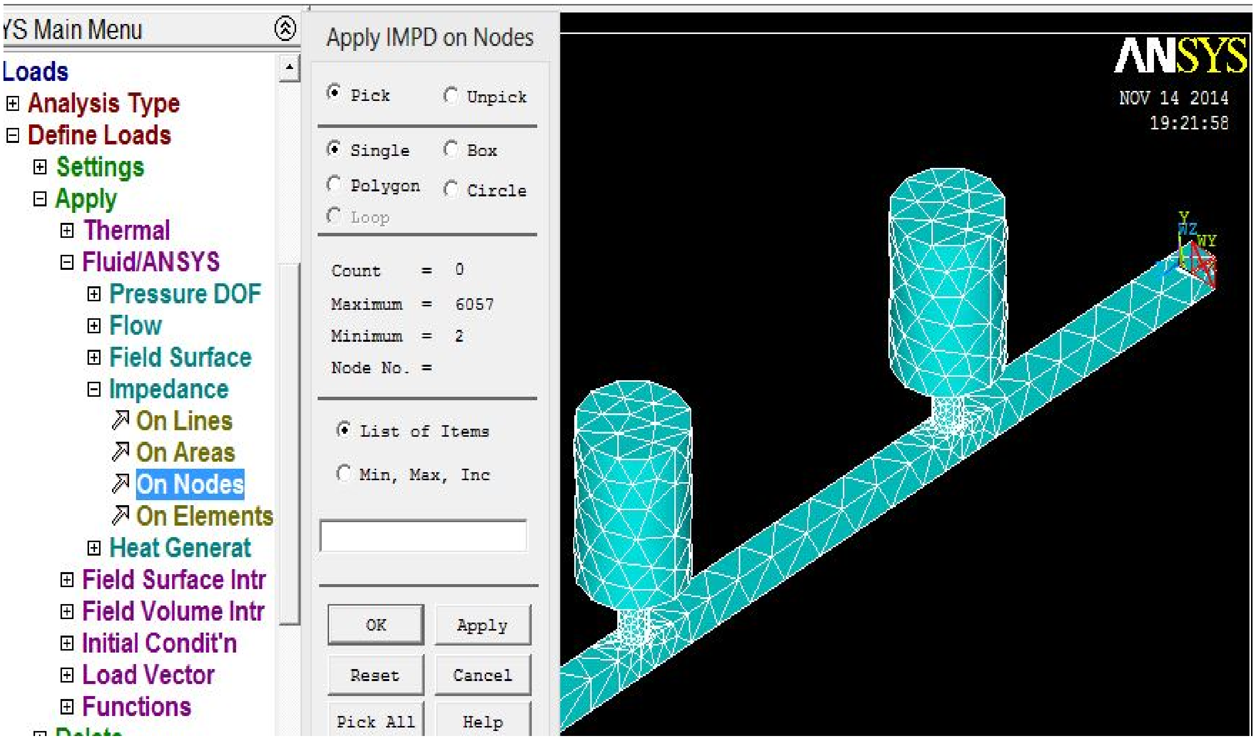
Type in command window

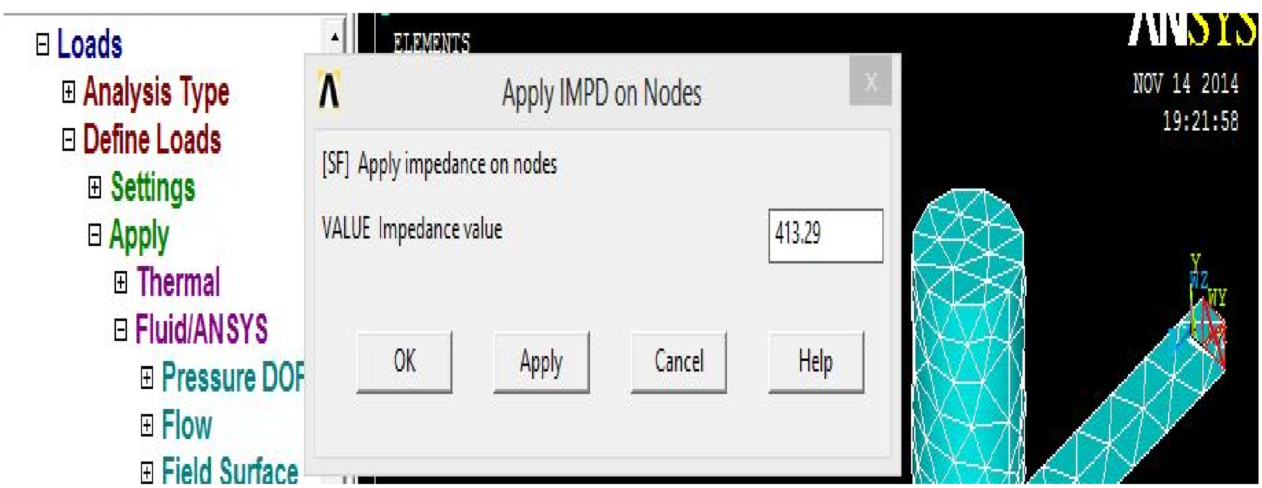
sf, all, shld,-1.53



19. Applying impedance boundary on inlet

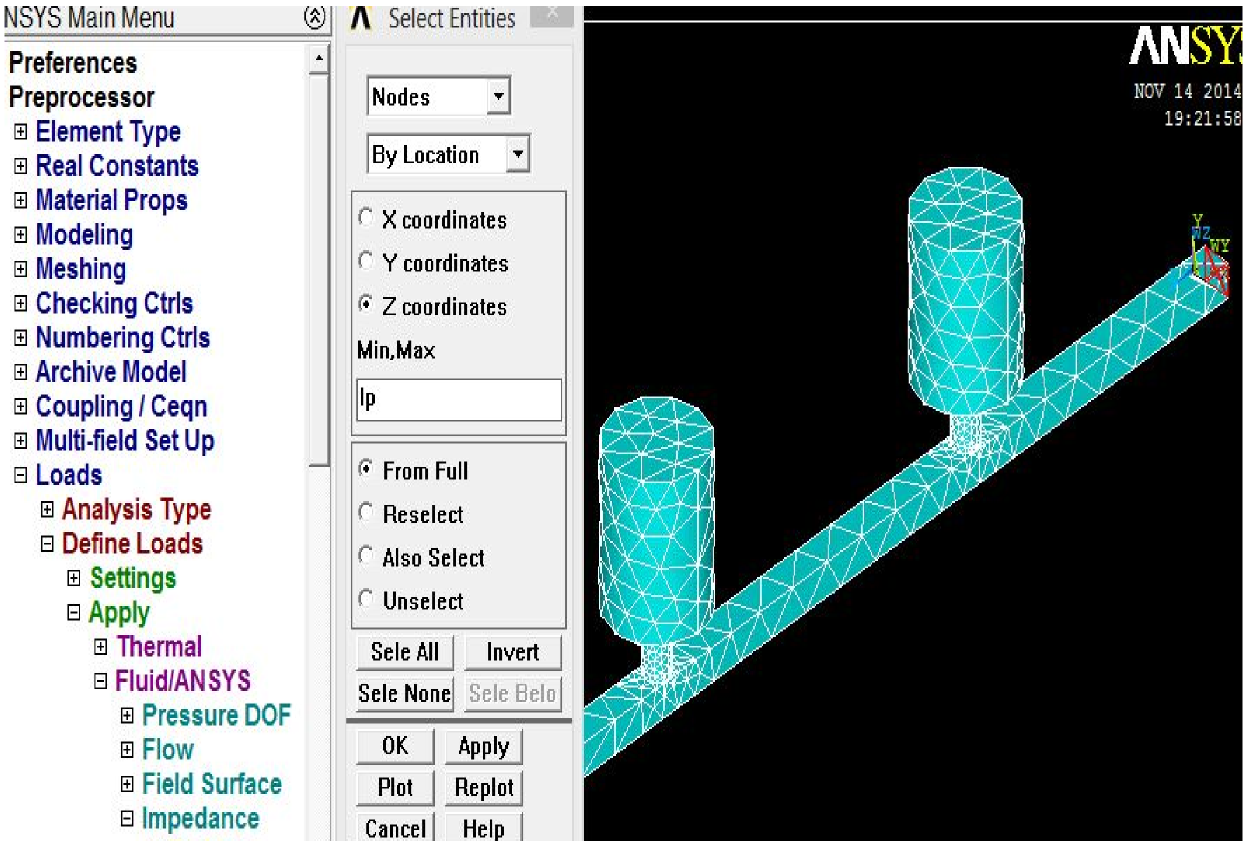
Preprocessors > loads > Define loads > apply > Fluid/ANSYS > impedance > on nodes > pick all > apply IMPD on nodes > value 413.29





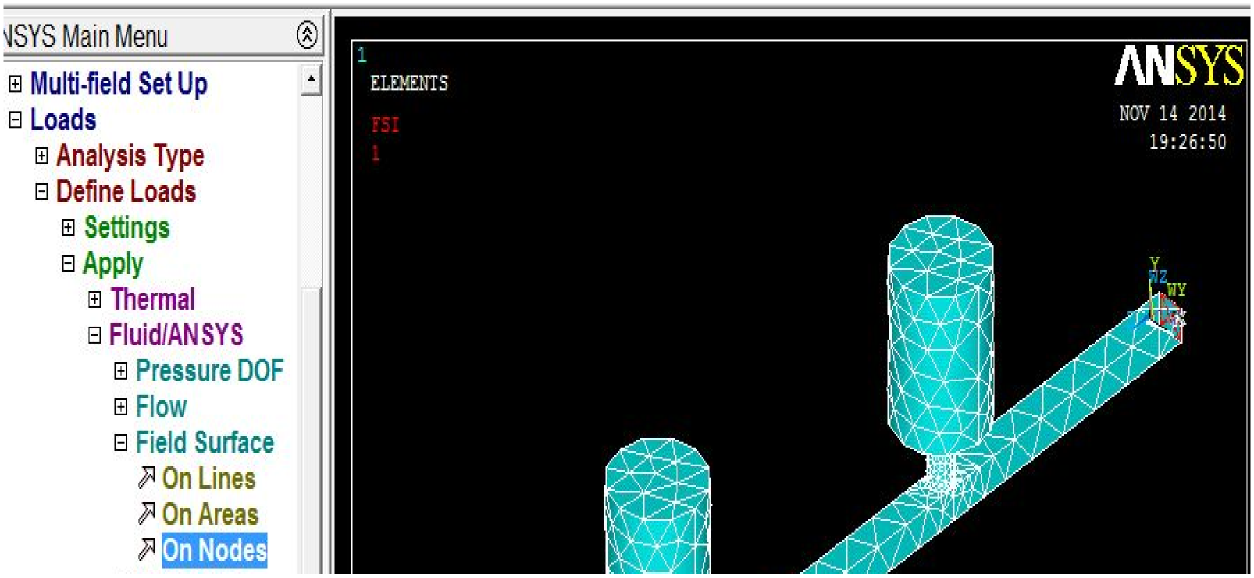
20. Selecting Nodes at outlet

Click on select on top > entities > nodes > by location > Z coordinates=lp> ok



21. Define loads at outlet

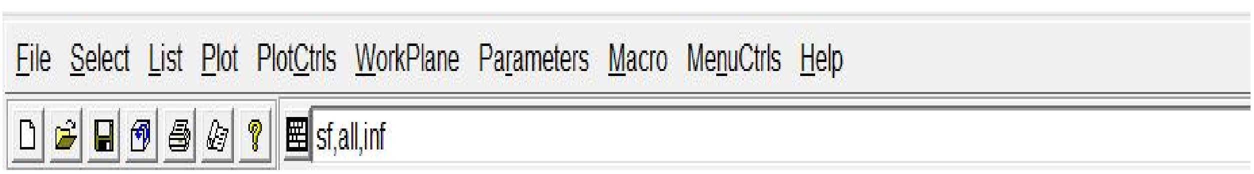
Preprocessors > loads > Define loads > apply > Fluid/ANSYS > field surface > on nodes > pick all



22. Radiation boundary on outlet

Then type in command window

sf, all, inf



23. Select everything

Then type in command window

alls

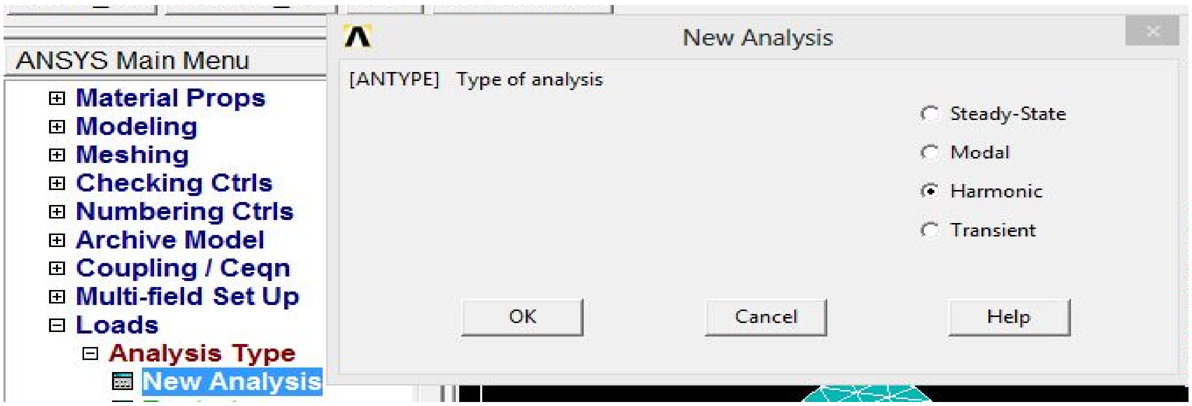
fini

24. Obtaining the solution

Specifying analysis type and options

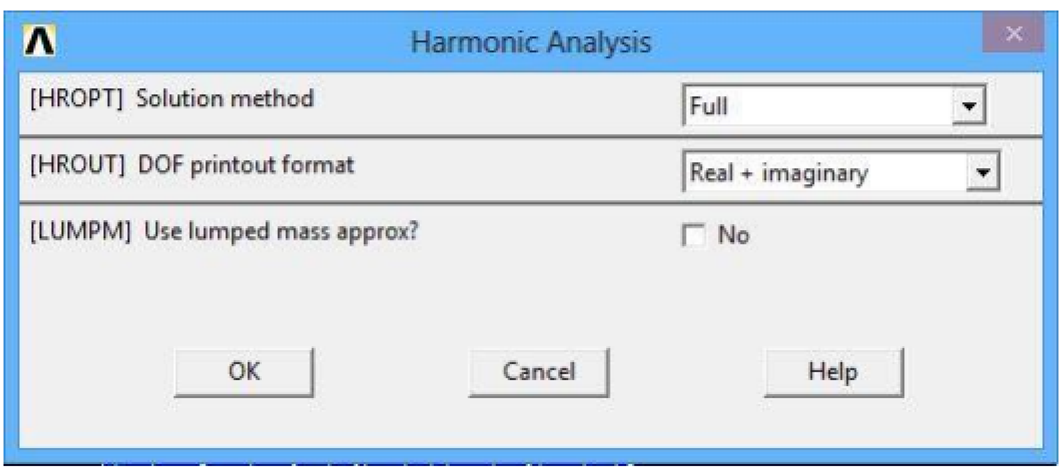
1. Define Analysis Type (Harmonic)

Solution >Analysis Type >New Analysis >Harmonic.

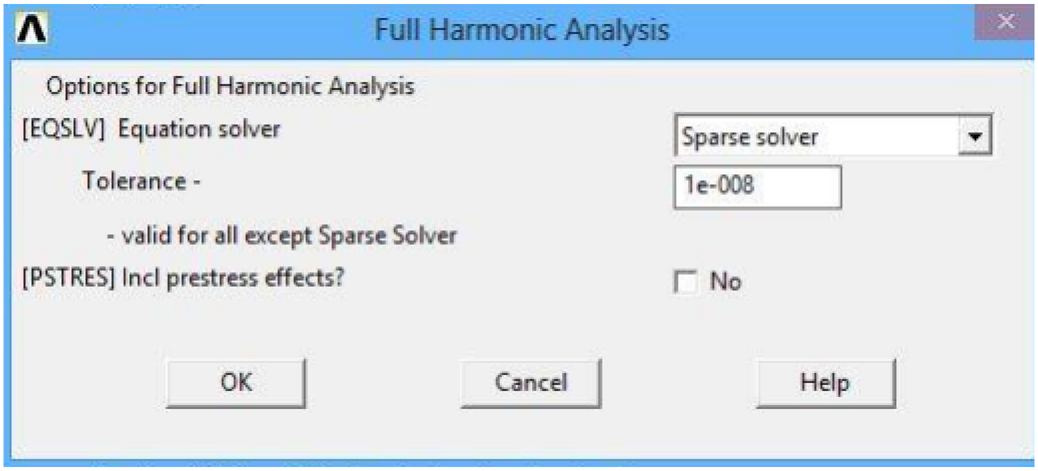


2. Setting options for analysis type

Select: Solution > Analysis Type > Analysis Options. The following window will appear

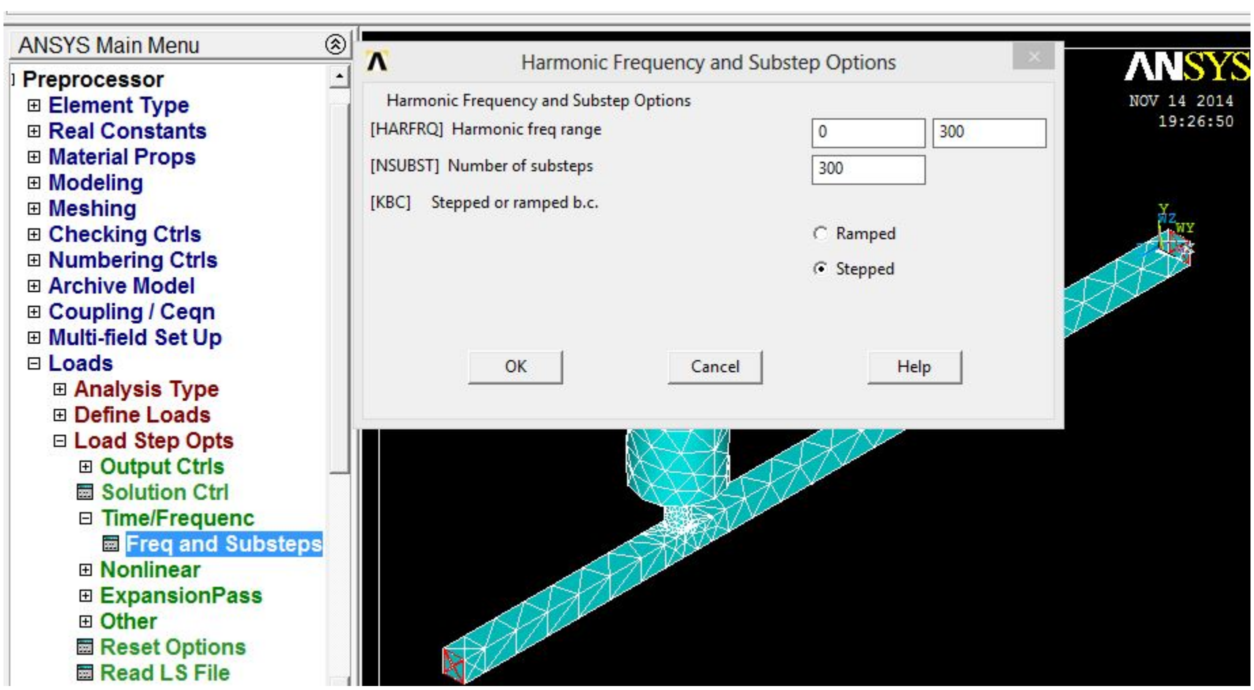


Clicking 'OK' will make the following window appear. The default settings (shown below) are used.

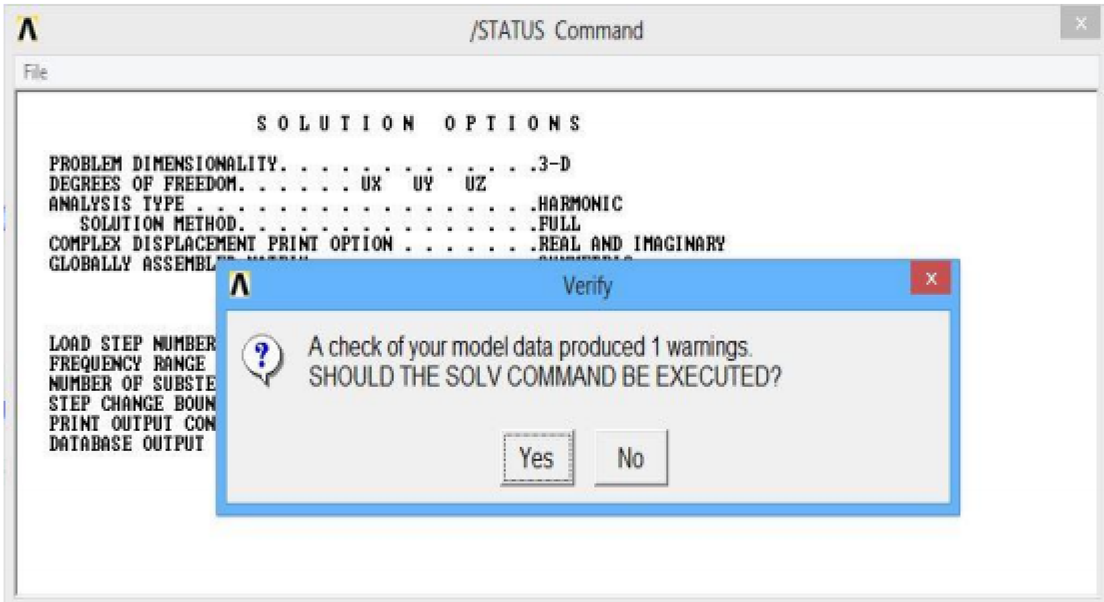


3. Defining frequency and Substeps

Loads > load step Opts > time/ frequency >freq and Substeps>freq range 0-300 > number of Substeps 300 > stepped



Solving: Solution > Solve > Current LS.



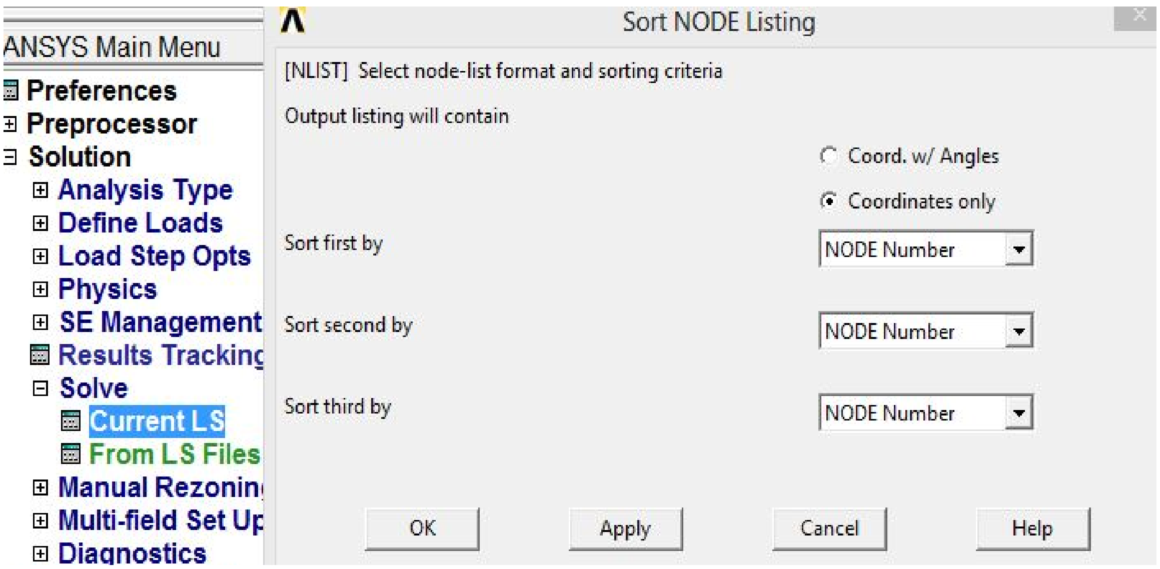
Verification: Yes

When the solution is done TimeHistPostPro is used to see the response of any node of the outlet of the duct.

Selecting the nodes at outlet

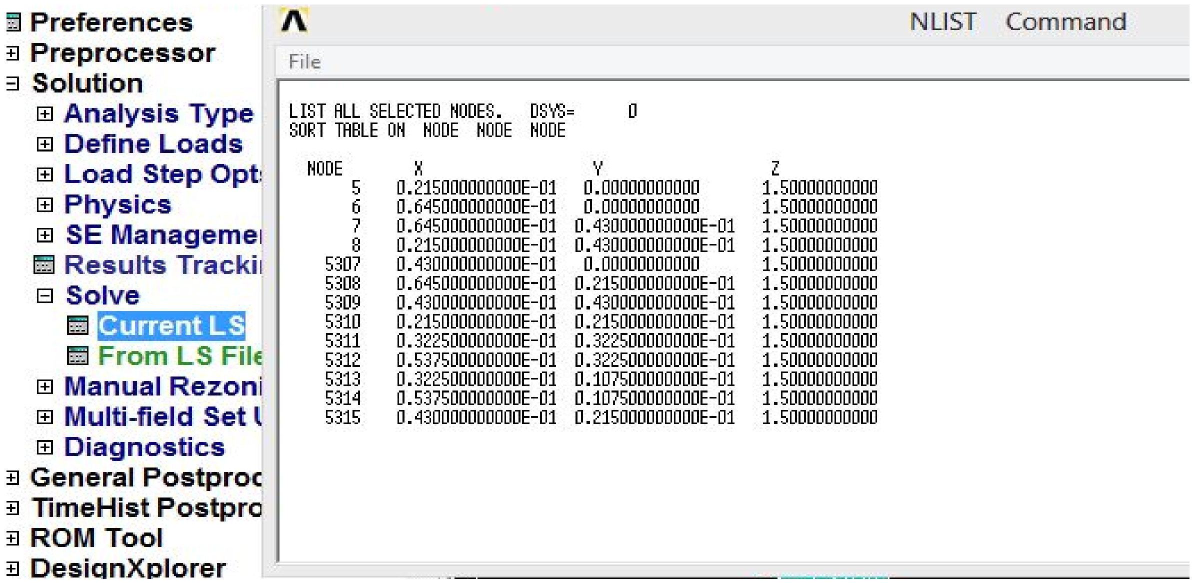
Node at outlet is selected like step 20.

Sort Nodes



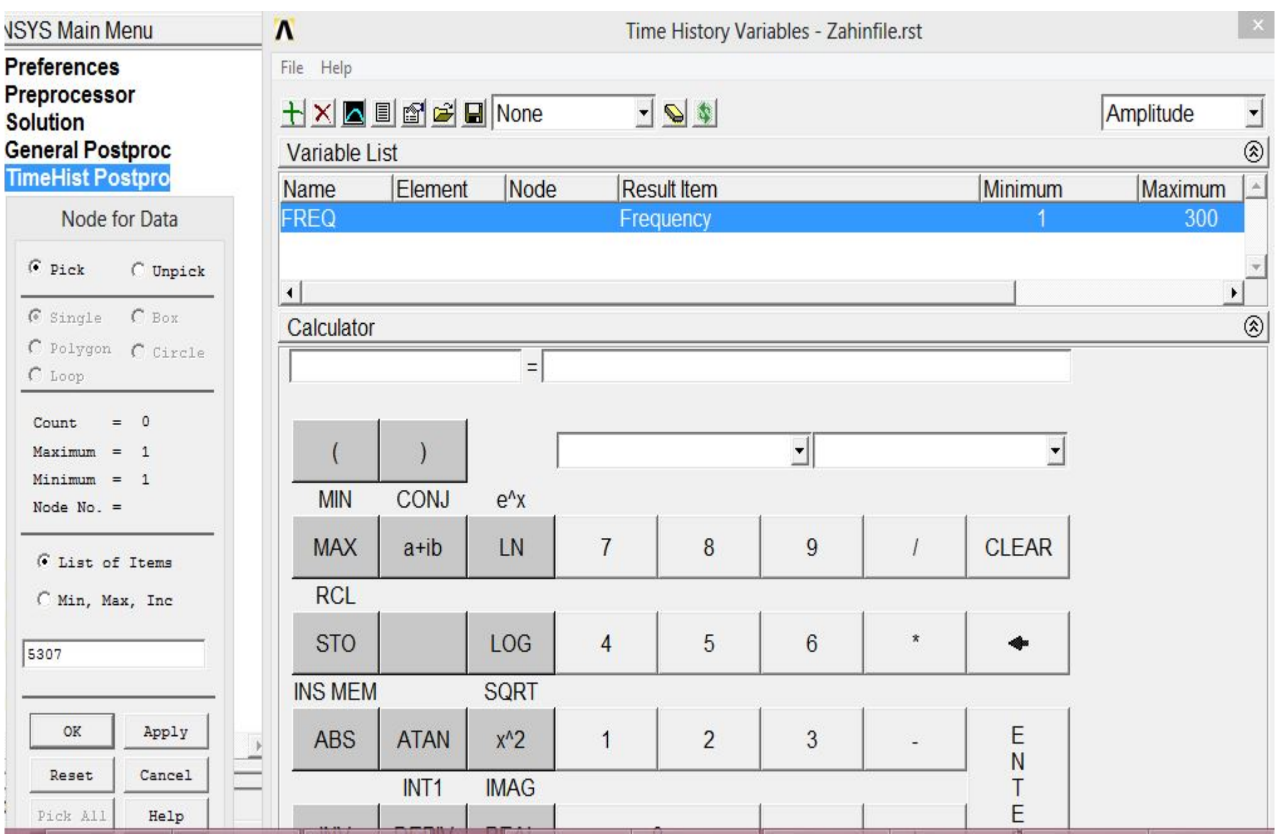
List nodes

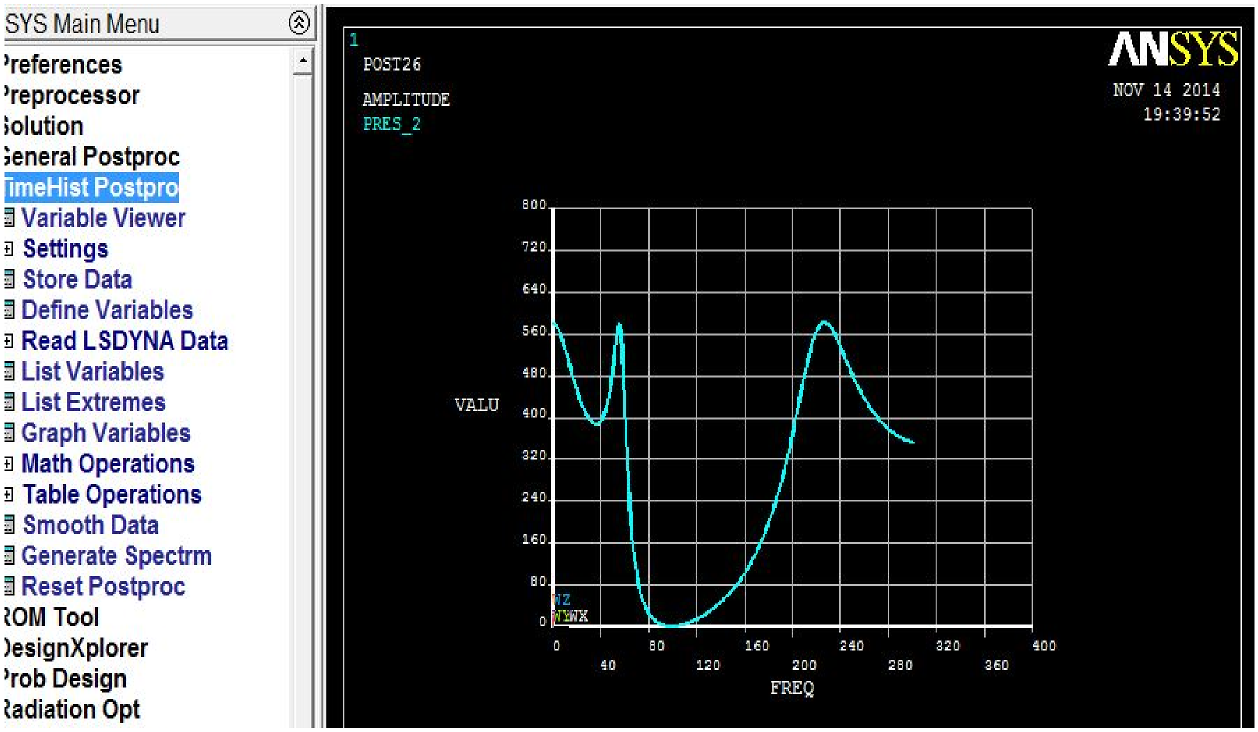
Nodes on top menu >list nodes> coordinates only > node number for each>Select a node for Z=lp



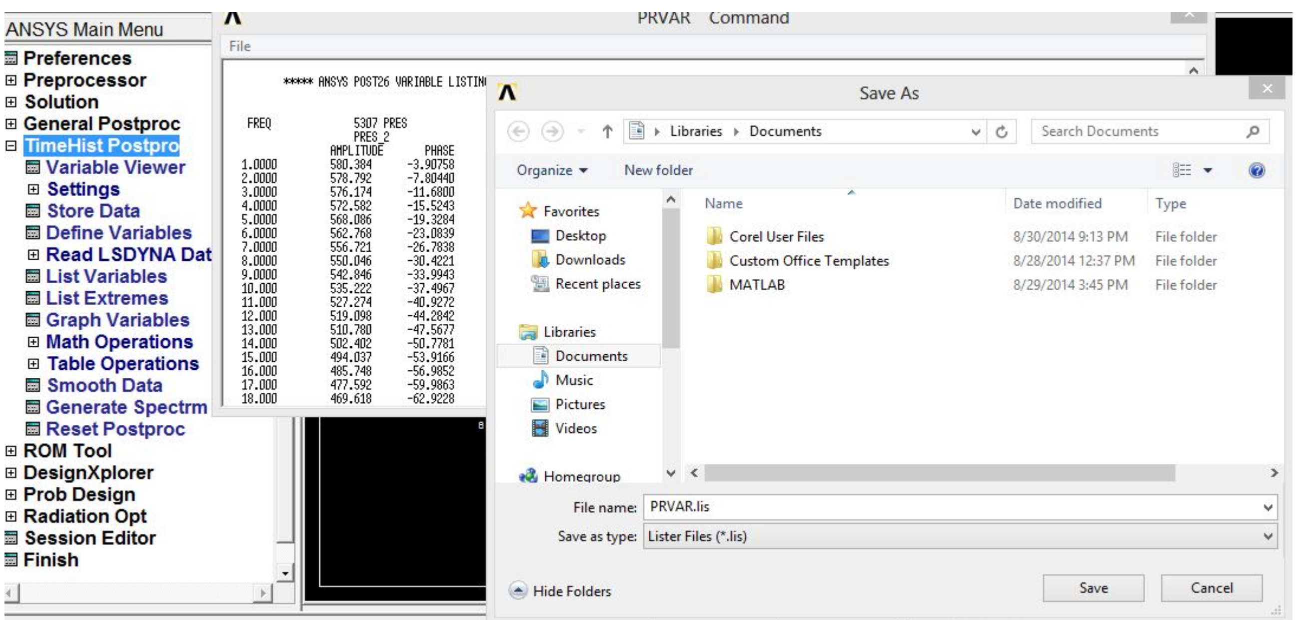
25. Post processing the results

TimeHistPostPro> variable viewer >add data > nodal solution > DOF solution > pressure > input node number selected earlier in 24.6 > click on graph data > list data > save data





Save File



The results are processed similar to Muffler in Chapter 4.3

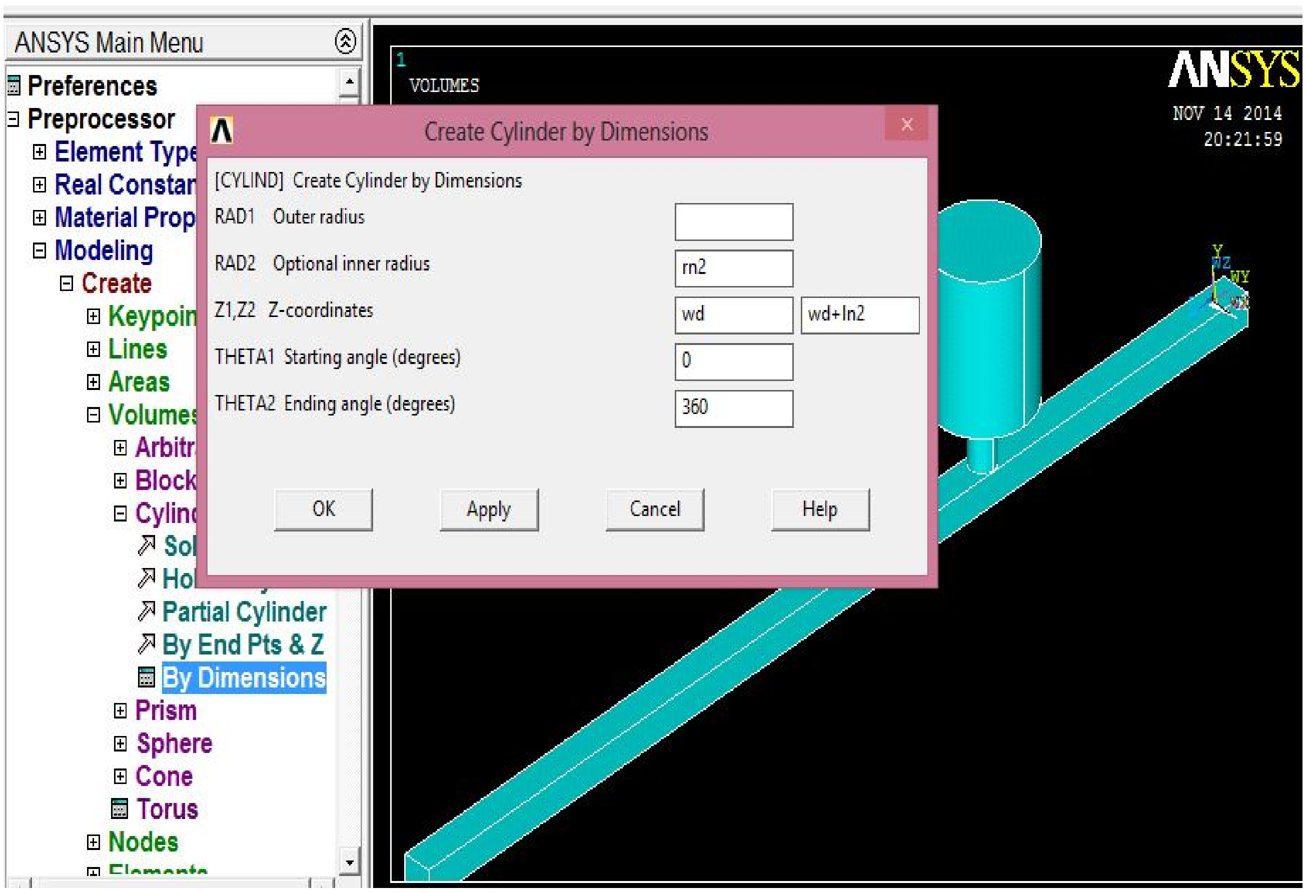
4.4.2 PROCEDURES FOR TWO DIFFERENT 1-DOF HELMHOLTZ RESONATOR

Here exactly same procedure is followed up to step 9 of two identical case mentioned in chapter 4.4.1. Then following steps are followed.

Modeling Of Two Different1-DOF Resonator

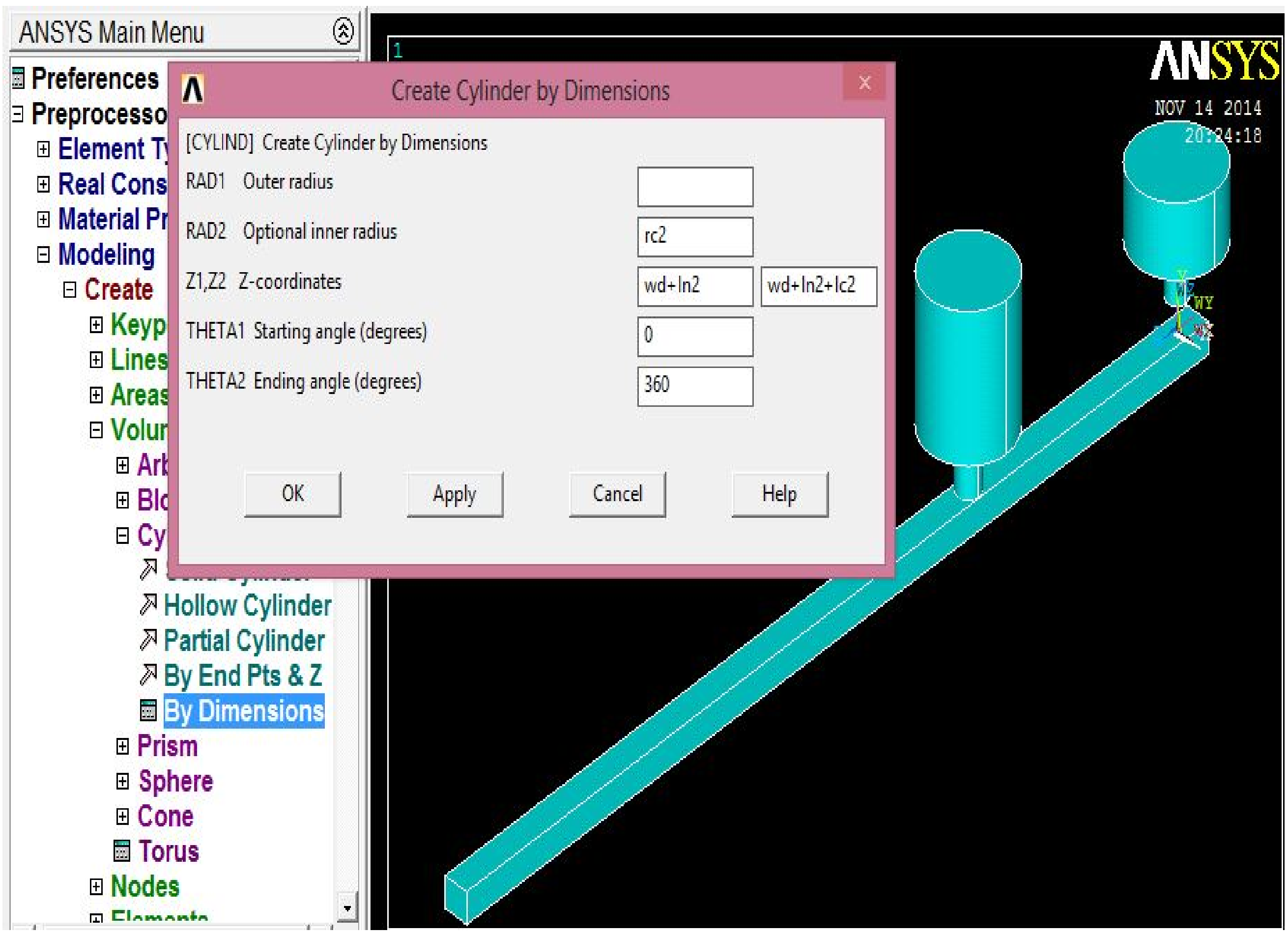
1. Neck of the second resonator

Steps Modeling > create > volumes > cylinder > by dimensions > optional inner radius RAD2=rn2, Z coordinates z1=wd, z2=wd+ln2



2. Cavity of the second resonator

Similarly put RAD2=rc2, z1=wd+ln2, z2=wd+ln2+lc2



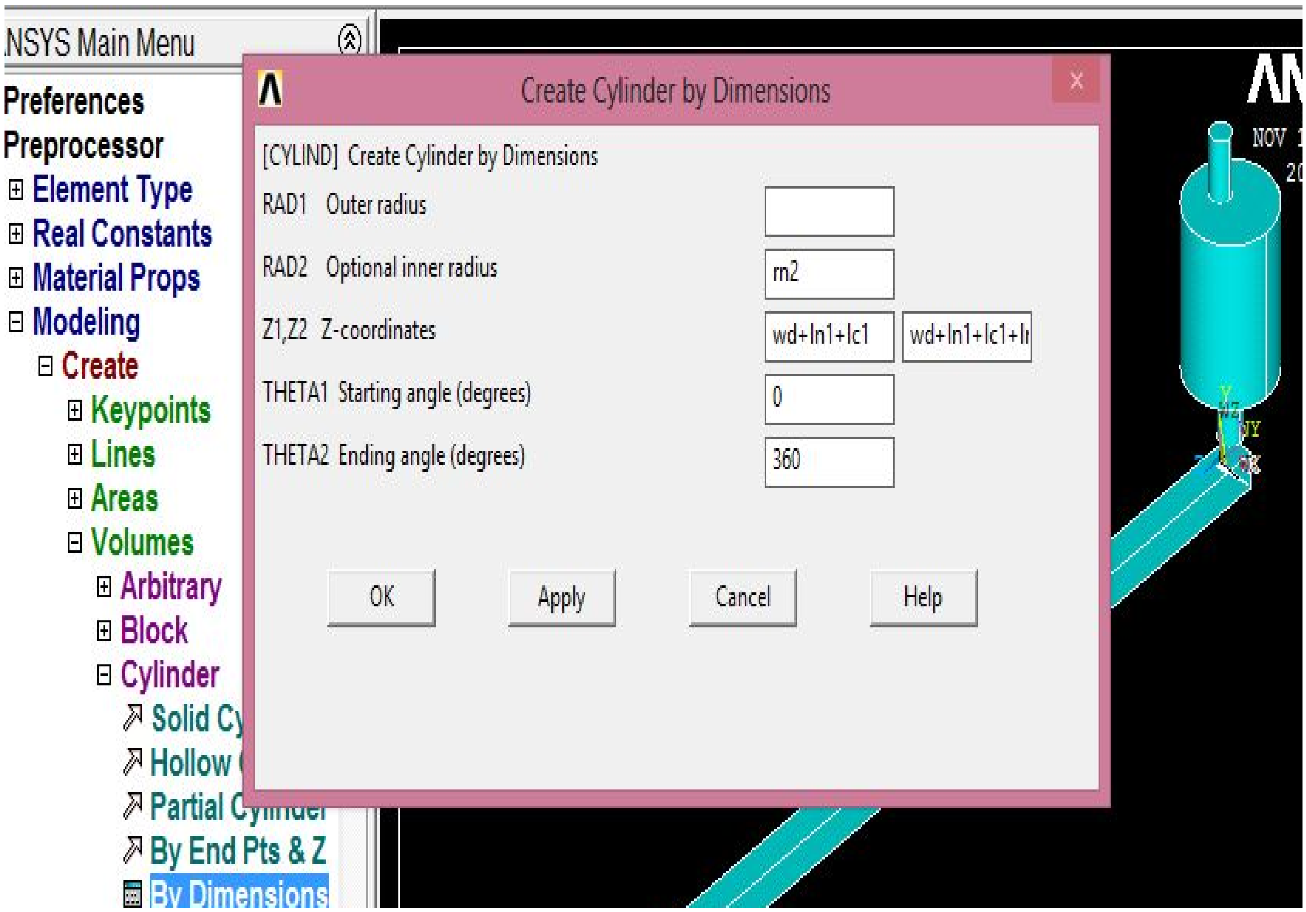
After this modeling remaining steps are exactly same as identical case chapter 4.4.2

4.4.3 Procedures for Two Identical 2-DOF Helmholtz Resonator

Here exactly same procedure is followed up to step 9 of two identical 1-DOF case mentioned in chapter 3.2.1.1. Then following steps are followed.

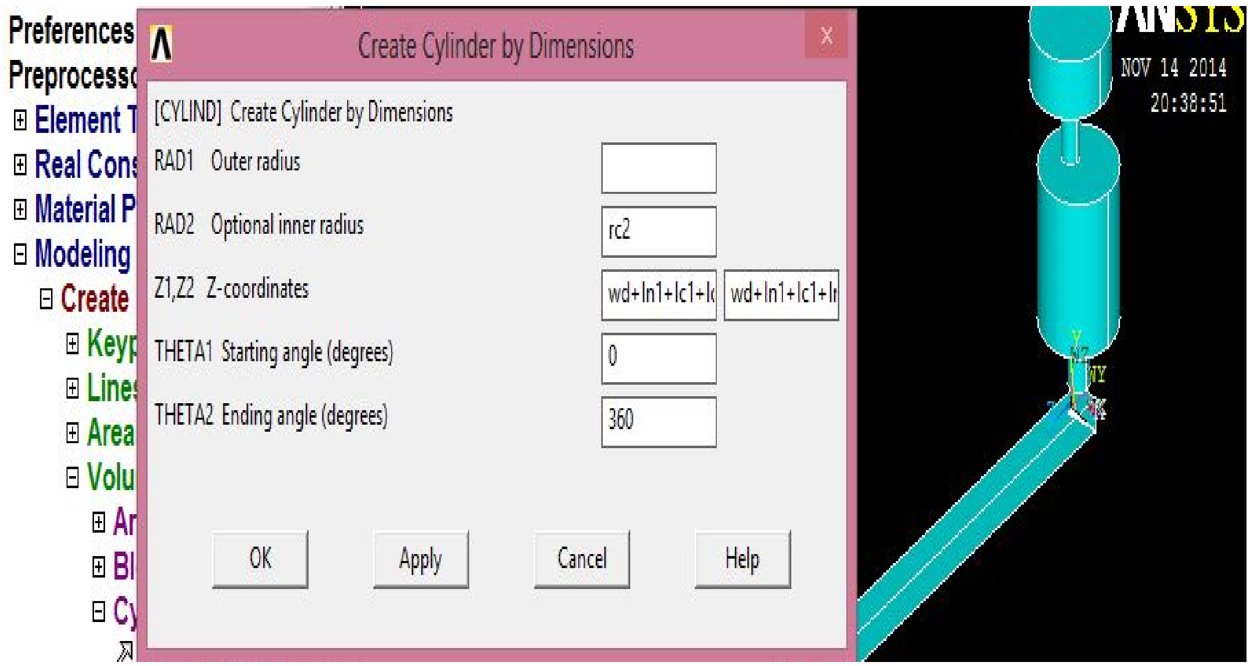
1. **2nd neck**:

Modeling > create > volumes > cylinder > by dimensions > optional inner radius RAD2=rn2,z1=wd+ln1+lc1,z2= wd+ln1+lc1+ln2



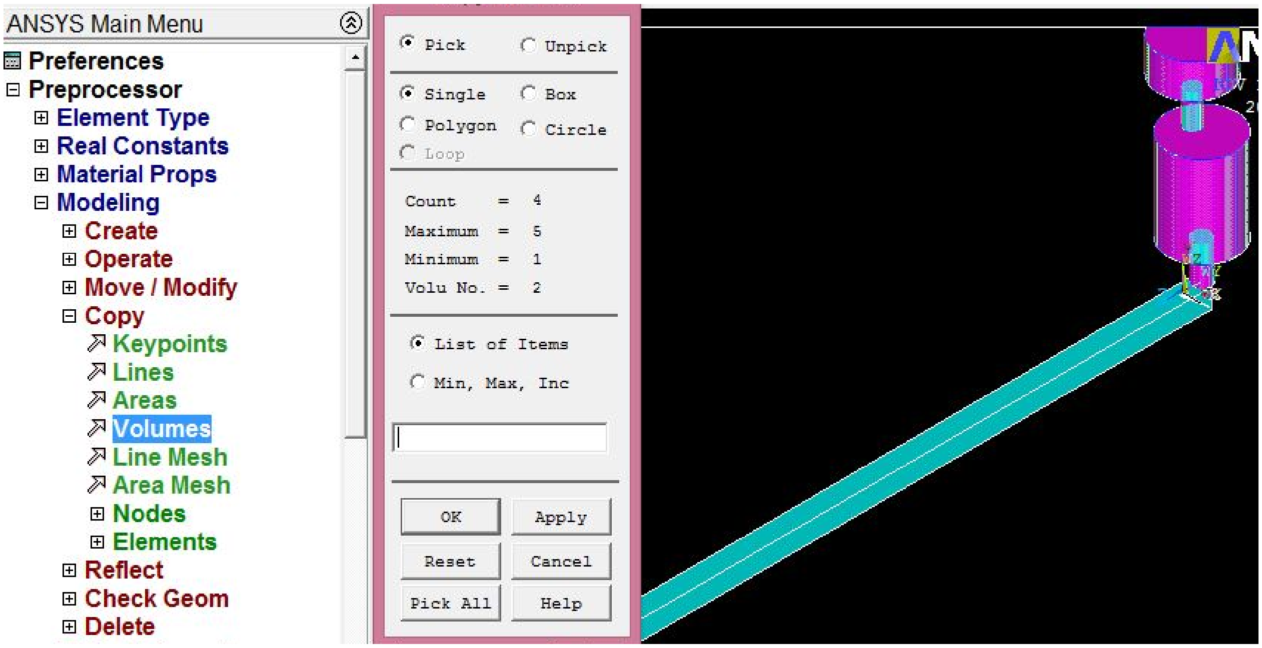
2.2nd cavity

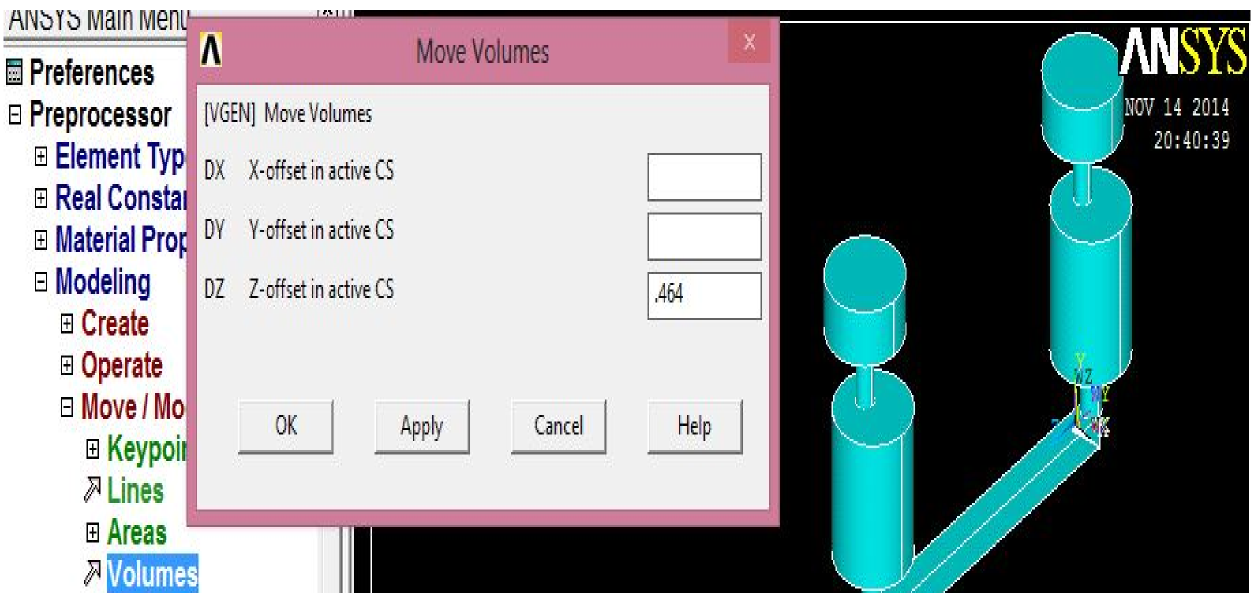
RAD2=rc2,z1=wd+ln1+lc1+ln2,z2= wd+ln1+lc1+ln2+lc2

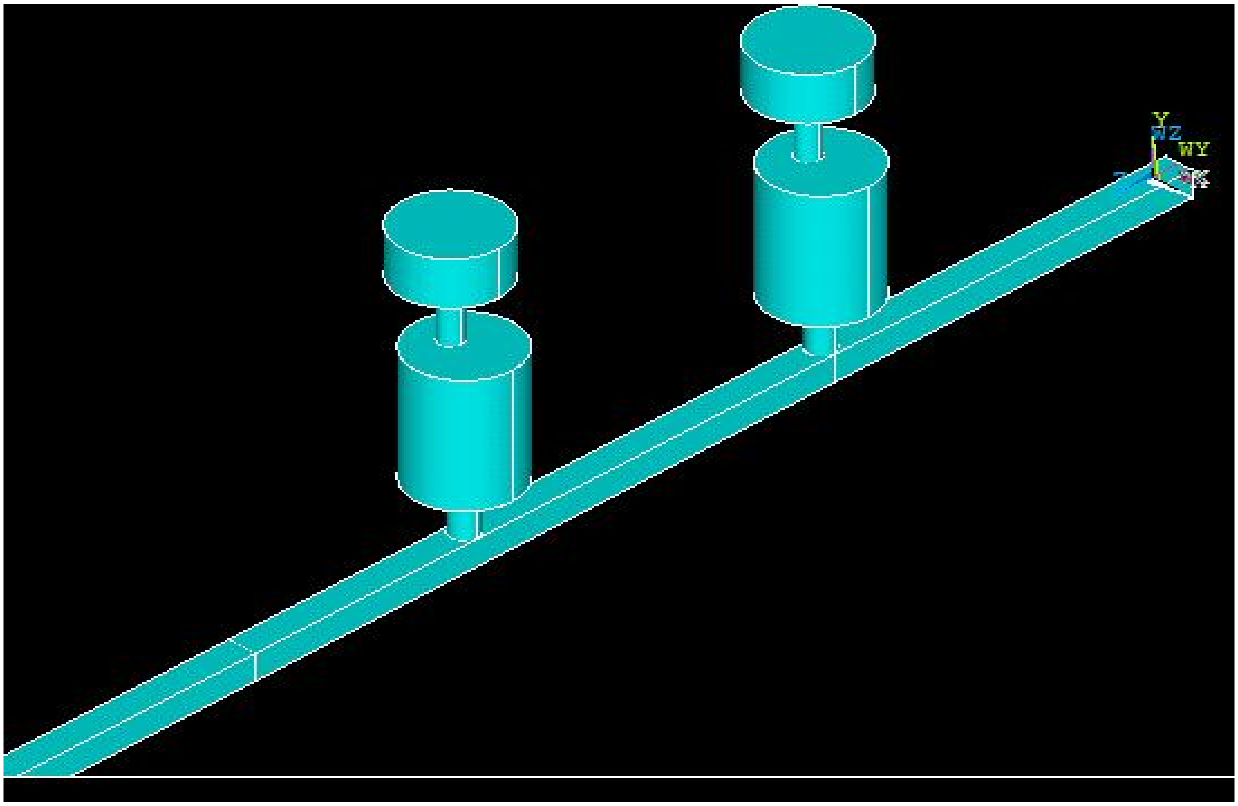


Copying the 2nd Resonator

Then 2nd resonator are copied like before


ANSYS Main Menu
Preferences
Preprocessor
Element Type
Real Constants
Material Props
Modeling
Create
Operate
Move / Modify
Copy
Keypoints
Lines
Areas
Volumes
Line Mesh
Area Mesh
Nodes
Elements
Reflect
Check Geom
Delete
Pick
Unpick
Single
Box
Polygon
Circle
Loop
Count = 4
Maximum = 5
Minimum = 1
Volu No. = 2
List of Items
Min, Max, Inc
OK
Apply
Reset
Cancel
Pick All
Help


Move Volumes
[VGEN] Move Volumes
DX X-offset in active CS
DY Y-offset in active CS
DZ Z-offset in active CS
.464
OK
Apply
Cancel
Help
ANSYS
NOV 14 2014
20:40:39



After this modeling remaining steps are exactly same as identical 1-DOF resonator case in chapter 4.4.2

Chapter 5

Results and Discussion

5.1 Case I (Relative Position)

The measured value of maximum TL found at around the resonance frequency 89 Hz was around 97.174 dB, 92.33 dB and 89.45 dB due to the relative spacing $\left(\frac{\lambda}{2}\right)$,$\left(\frac{\lambda}{4}\right)$ and $(< \frac{\lambda}{4})$ respectively compared to that of 50 dB for a single 1-DOF resonator shown in Figure 4. The above numerical results show that the overall performance for the two resonators at $\left(\frac{\lambda}{2}\right)$ distance apart is significantly higher than that of a single 1-DOF resonator attached to a duct. In case of two closely spaced identical resonators, the measured value of maximum TL at around 89 Hz is also very higher than a single resonator. Another notable observation is that an antinode (increase of sound pressure level) has been created at around 70 Hz due to the interaction of the two resonators but its effect is negligible on overall noise attenuation performance after 70 Hz shown in Figure 4. So, the above investigation suggests that the optimum relative spacing between the two identical 1-DOF resonators is $\left(\frac{\lambda}{2}\right)$distance apart.

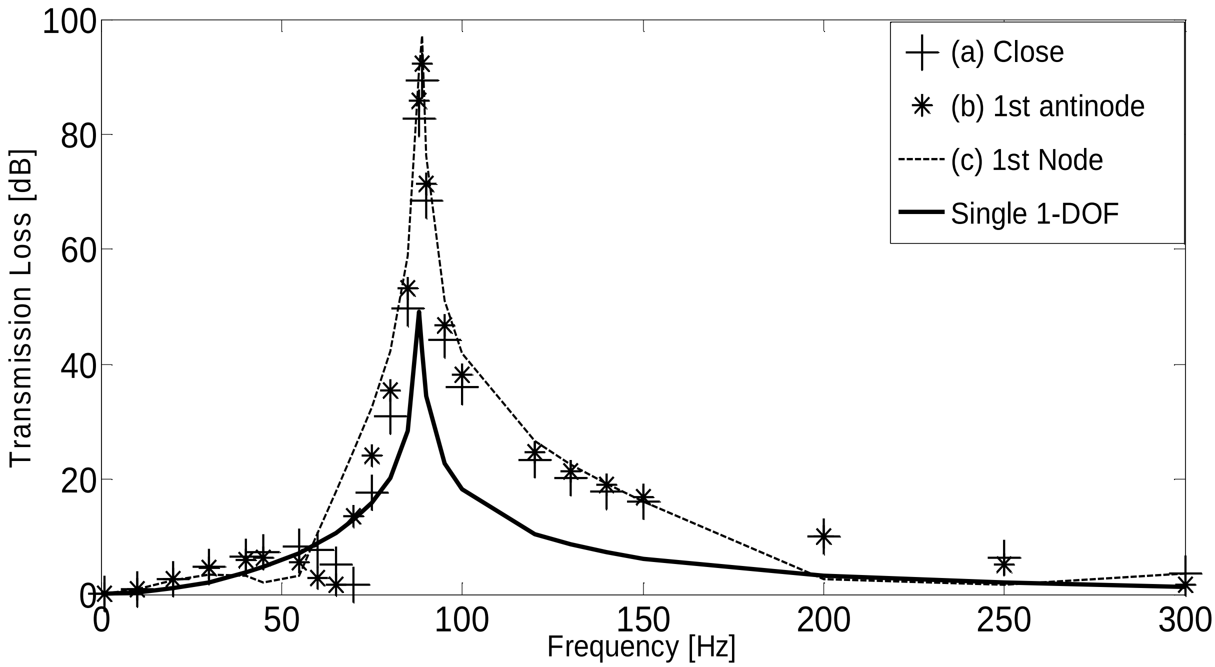


Figure 5.1: Numerical Results of Various Relative Positions

5.2 Case II (Varying Geometry of the Cavity)

The measured maximum value of TL at around the resonance frequency 89 Hz was 97.174 dB for optimum relative spacing; $\left(\frac{\lambda}{2}\right)$, the cavity ratio ; 1.59, neck ratio of 2.10 (4.1 Case I) for both the resonators. Then, by increasing the cavity ratio to 2, resonance frequency increases to 91 Hz and TL increases to around 118 dB. For further increasing the ratio to 2.77, the resonance frequency decreases to 88 Hz and TL drops to around 115 dB at around 88 Hz shown in Figure 5.

So, from this above investigation, overall noise reduction performance can be increased by 20 dB more for the cavity ratio 2.0 than that of 1.59.

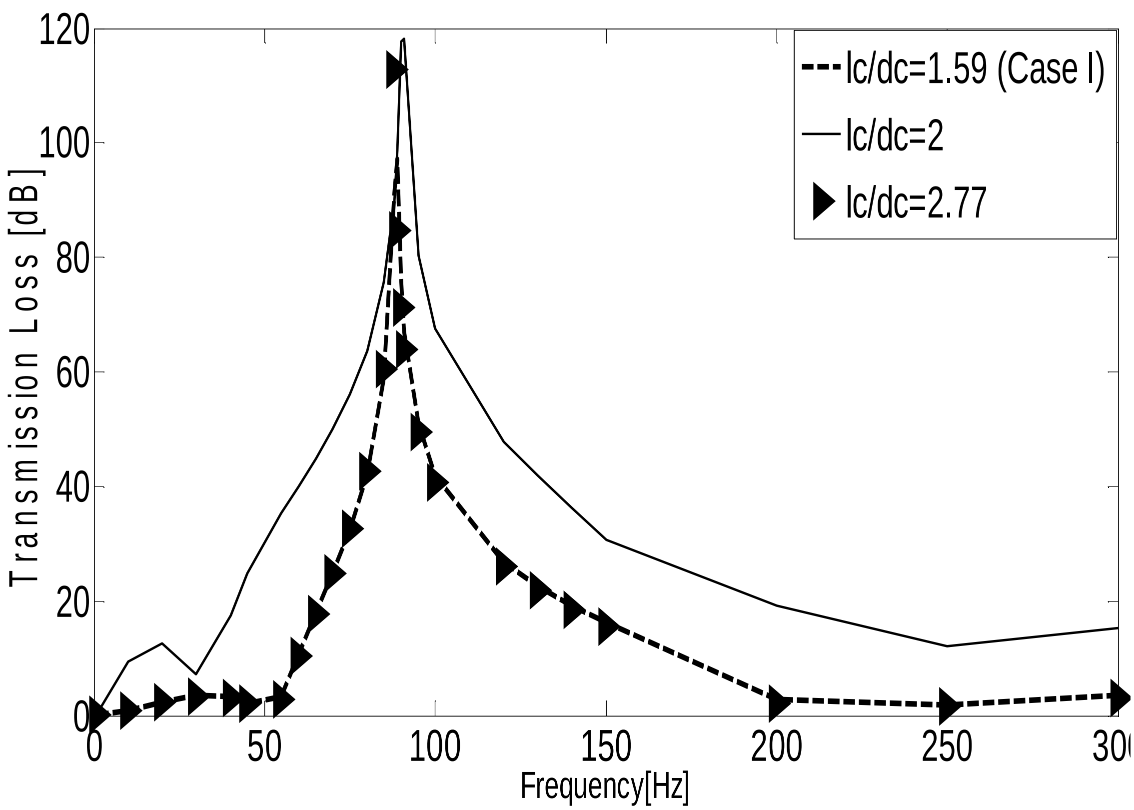


Figure 2.2: Numerical Results of Varying Geometry of the Cavity

5.3 Case III (Varying Geometry of the Neck)

For this investigation the neck ratio was changed keeping the cavity ratio 2.0 (4.2 Case I).The measured value of maximum TL for two resonators at $\left(\frac{\lambda}{2}\right)$ distance apart was 118 dB at around the resonance frequency (91 Hz) in Case II for the neck ratio 2.10. By increasing the neck ratio to 2.25, resonating frequency decreases to 82 Hz and the TL becomes 90 dB around 82 Hz. But, by increasing the ratio to 2.5, resonating frequency decreased to 80 Hz but the TL increases to around 130 dB at around 80 Hz. For further increasing the ratio to 3, resonance frequency decreased to 72 Hz but the TL increases to almost 120 dB at around 72 Hz shown in Figure 6.

So, from the above investigation, overall noise reduction performance can be increased by 12 dB more for the neck ratio 2.5than that of 2.10.

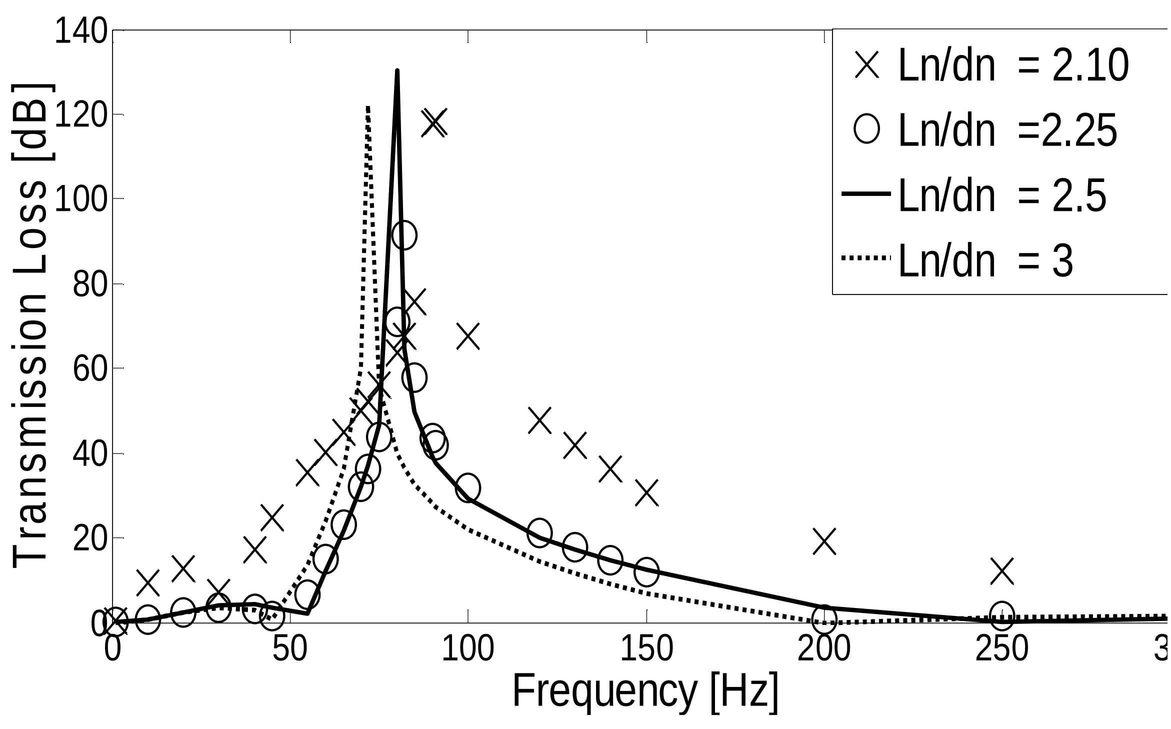


Figure 5.3: Numerical Results of the Varying Geometry of the Neck

Chapter 6

Chapter 6

Validation of Published Results

In this literature, a validation of maximum *TL* was performed using the dimensions of Figure 1for a single 1-DOF Helmholtz resonator. Analytical validation of *TL* was done for the equation 2 [6] and numerical validation was performed for the resonator of Figure 1 using finite element method. The analytical and numerical results of TL agree satisfactorily with the numerical results of TL of published article [6] shown in Figure 6.

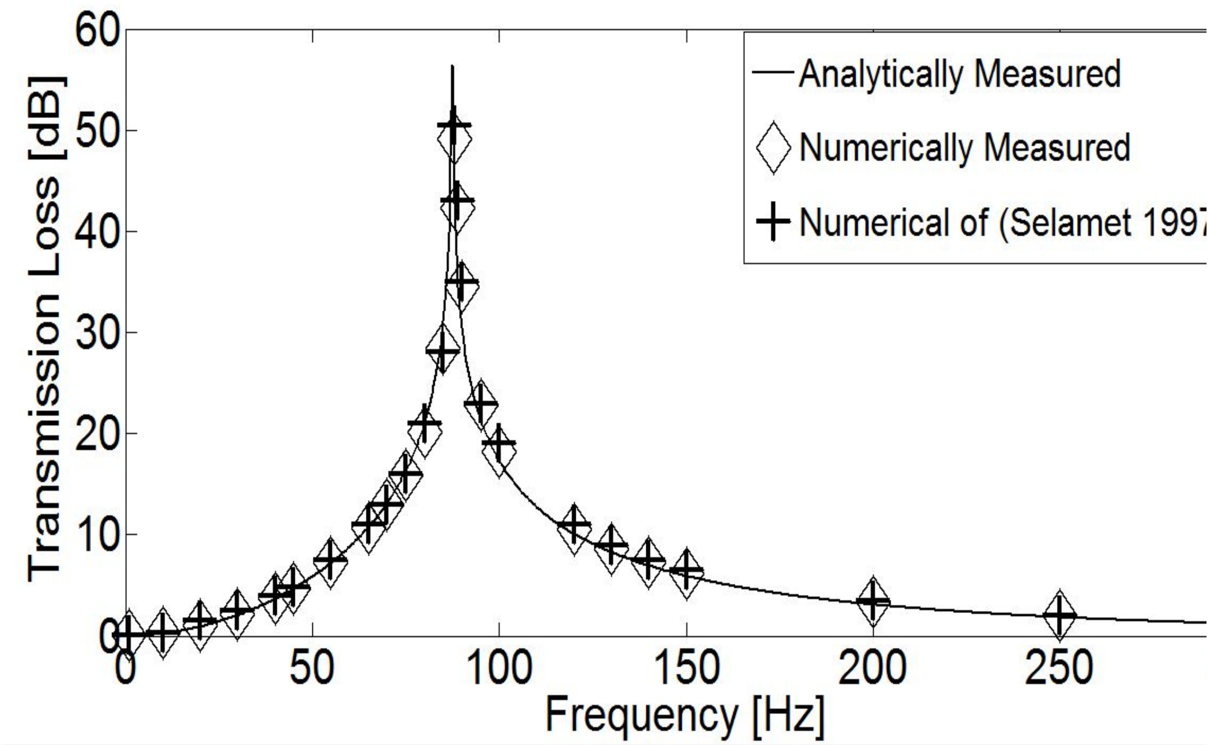


Figure 6: Validation of Analytical and Numerical Results with Published Numerical Results [6] for A Single 1-DOF Resonator

Chapter 7

Conclusion and Future Works

An optimized system for both two identical 1-DOF Hemholtz resonators in terms of relative position and geometry of both the cavity and neck has been investigated in this thesis. Maximum noise attenuation of 130 dB has been achieved compared to a single 1-DOF resonator at around the resonance frequnecy has been found numerically for two identical cylindrical 1-DOF Helmholtz resonators by finite element method which signifies the importance of this proposed procedure. The optimized system found from the above investigations is : relative spacing : $\frac{\lambda}{2}$, cavity ratio: 2.0 and neck ratio: 2.0.

Future works are:

- Experimental Investigation for two identical 2-DOF Helmholtz Resonator and comparison of results with numerical analysis.
- Numerical Analysis of two different frequency 2-DOF Helmholtz Resonators.
- Acoustic modeling of the resonators considering fluid structure interactions.

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