Purdue University
Purdue e-Pubs

International Compressor Engineering Conference   School of Mechanical Engineering

2004

Controling the Noise Radiation of Hermetic Compressors by Means of Minimization of Power Flow Through Discharge Pipes Using Genetic Algorithms

Andrey R. DaSilva
Universidade Federal de Santa Catarina

Arcanjo Lenzi
Universidade Federal de Santa Catarina

Edmar Baars
Embraco

Follow this and additional works at: https://docs.lib.purdue.edu/icec

CONTROLING THE NOISE RADIATION OF HERMETIC COMPRESSORS BY MEANS OF MINIMIZATION OF POWER FLOW THROUGH DISCHARGE PIPES USING GENETIC ALGORITHMS

Andrey Ricardo da Silva, MSc[1], Arcanjo Lenzi, PhD[2] . Edmar Baars, MSc[3]

[1,2]Universidade Federal de Santa Catarina, Department of Mechanical Engineering,
Florianópolis, Santa Catarina, Brazil
[1,2]Phone: +55 48 2340689, [1]E-mail: andrey@lva.ufsc.br, [2]E-mail: arcanjo@lva.ufsc.br

[3]EMBRACO, Laboratory of Acoustics,
Joinville, Santa Catarina, Brazil
Phone: +55 47 441 2698, E-mail: edmar_baars@embraco.com.br

ABSTRACT

One way to reduce the global noise level from hermetic compressors consists in minimizing the vibratory energy flow which originates inside the compressors and reaches their outer shell, where it can be efficiently radiated. The focus of this work concentrates on the hermetic compressor EM, used in domestic refrigeration systems, produced by EMBRACO. The energy flow can be transmitted to the outer shell through a significant number of ways. Here, the discharge pipe is taken to be an important energy transmitter and, therefore, is geometrically modified in order to minimize its capability to transmit vibratory energy to the system case in some specific frequency bands. An evolutionary optimization method known as Genetic Algorithms is used to find the ideal modifications. The Finite Elements Method is used to determine the objective function (the vibratory power flow between the pipe and the system case). Ten optimized candidates were achieved for each of four optimized frequency bands. Comparisons between the performances of optimized structures and the original geometry are presented in terms of power flow spectra. At last, the reproduction in practice of the best candidate of each optimization band is discussed, considering aspects of response sensibility due to geometric perturbations inherent to fabrication processes.

1. INTRODUCTION

Methods of vibration control are not only used to avoid structural damage in mechanical systems. Another important application concerns also to the necessity of reducing their global noise. This application has become more usual due to the increasing number of new health and environmental policies.

The goal of this work consists in reducing the noise radiation from the hermetic compressor EM, manufactured by EMBRACO, by avoiding the energy that originates inside the compressor to reach its outer shell, where it can be easily transformed into noise due to the shell's radiation efficiency. Concerning this, the system parts are categorized in three types of components: a vibration source (motor-kit), a vibratory energy transmitter (discharge pipe) and a radiating plate (system shell), presented in Figure 1. Usually in this type of systems, the greatest portion of radiated noise does not originate in the vibration source itself but in other components with higher sound radiation efficiency.

There are many methods available for vibration control. The most widely used are the, so called, classical passive control methods. These methods reduce the transmission of vibratory energy by adding mass and/or damping (Mead, 1988) to the system. However, these methods usually jeopardize many design goals which can limit their applicability. The application of methods based on active control has increased in the last few years. These methods act to produce countervibrations in order to cancel nominal vibrations. Unfortunately, there are still many restrictions (Gorman, 1975) to their applications related, mainly, to the high costs of implementation and to the specificity of the problem.

This work proposes, as an alternative, the reduction of vibratory energy flow that reaches the compressor's shell by means of geometric modifications on the component that plays an important role in the energy transmission mechanism: the discharge pipe.

The necessary geometric modifications are done under the control of an evolutionary optimization algorithm known as Genetic Algorithm. The objective function (the averaged power flow between the pipe and the system case) is calculated by the Finite Elements Method (FEM).

The optimization variables are 28 of the 30 keypoints that compose the original pipe geometry. Each keypoint can vary in space three directions within a limit defined by the geometric system restrictions (position of other system components that are not considered in the analysis). Ten optimized candidates were obtained for each one of the four analyzed frequency bands over 6 kHz : (i) 250 Hz; (ii) 500 Hz; (iii) 1000 Hz; (iv) 2000 Hz. The criterion used to obtain the best candidate in each process was the minimum averaged power flow. Comparisons between the performances of optimized structures and the original geometry are presented in terms of power flow spectra.

This paper is divided in five parts. The second part describes the mechanical system studied in this work and the role played by the Finite Element Method in the optimization process. The third part briefly presents the Genetic Algorithm theory and describes the computational strategy used to implement it on the geometric optimization of the discharge pipe. The fourth part explains the adopted criteria to choose the best candidate among then finalists of each of four optimization processes and presents their comparison with the original pipe in terms of power flow spectra. At last, considerations of robustness and practical feasibility of the chosen geometries are presented in the fifth part.

2. THE SYSTEM

The system analyzed is this paper is considered to have three parts (Figure 1): a motor-kit, a discharge pipe and the system case. The motor-kit is considered to be the only vibration source of the system, which energy is transmitted to the compressor's shell through the discharge pipe (Figure 2). The system case, for topologic reasons, is considered to be the only radiator of the system (Silva, 2003). Therefore, the sound radiation from the other components to the overall noise radiation is being neglected. It is also considered that the only way for the vibratory energy to reach the system case is through discharge pipe.
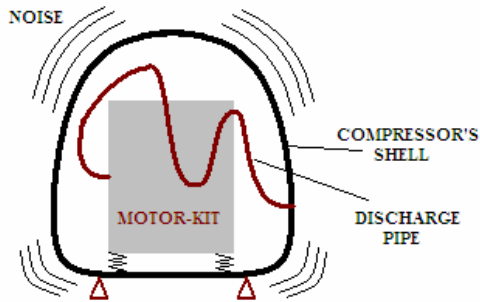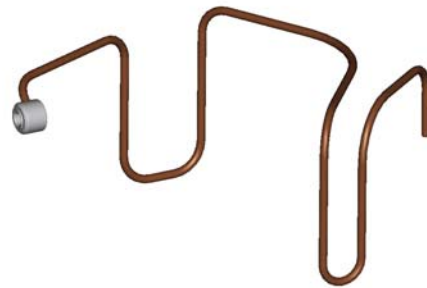

Figure 1 – System simplification.

Figure 2 - Original pipe geometry.

The objective function is taken to be the mean power flow, $\overline{W}_{IN}$, averaged over a frequency band, calculated at the connection between the pipe and the case, expressed by:

$$\overline{W}_{IN} = \frac{1}{n}\sum_{i=1}^{n} W_{IN_i} \tag{1}$$

where, n is the number of frequency values within the frequency band and W_{IN_i}, the sum of the power flow in the directions x, y and z for the i-th frequency, so that:

$$W_{IN_i} = \frac{1}{2}\Re\left\{\overline{F}_{X_i}\overline{V}^*_{X_i} + \overline{F}_{Y_i}\overline{V}^*_{Y_i} + \overline{F}_{Z_i}\overline{V}^*_{Z_i}\right\} \tag{2}$$

where, $\overline{F}_{X_i}$, $\overline{F}_{Y_i}$ and $\overline{F}_{Z_i}$ are the complex forces for the i-th frequency in the directions x, y and z, respectively, and $\overline{V}^{*}_{X_i}$, $\overline{V}^{*}_{Y_i}$ and $\overline{V}^{*}_{Z_i}$ are the complex conjugates velocity values for the i-th frequency in the directions x, y and z, respectively.

The flexural and torsional moments were not considered.

2.1 The Finite Element Method

The Finite Element method was used to calculate the averaged power flow for each frequency band of interest. This procedure followed four steps: modeling the system components; solving the eigenvalue problem; creating the frequency response functions (FRF's) and post-processing of nodal results (forces and velocities) obtained from the previous analyses, from which the averaged power flow is calculated.

The software ANSYS, version 7.0, was used to execute all the FEM analysis. This software has available many different methods to fulfill the analyses. For the solution of the eigenvalue problem, the Block of Lanczos Method (Maia, 1997) was used. The Full method was used to carry out the determination of the FRF's.

The Genetic Algorithm was implemented inside the software ANSYS environment, version 7.0 using APDL (ANSYS Parametric Design Language).

2.2 The System Parametrization

Some simplifications were made in order to build up the FE model to represent the system. The vibration source (motor-kit) was represented by three unitary harmonic forces in three orthogonal directions, acting on keypint 1 (Figure 3). The pipe geometry is build by connecting 30 keypoints by using cubic splines. The connection of the keypoins results in a line model, which is then, meshed using pipe elements (PIPE16) available in the software library. The right end of the pipe model is connected to a super element, which is intended to represent the same boundary conditions of the tube when connected to the compressor's shell. Both the pipe mesh and the shell, reduced into a super element, were built using 12 elements per flexural wavelength, considering that the higher analyzed frequency was equal to 8 kHz.
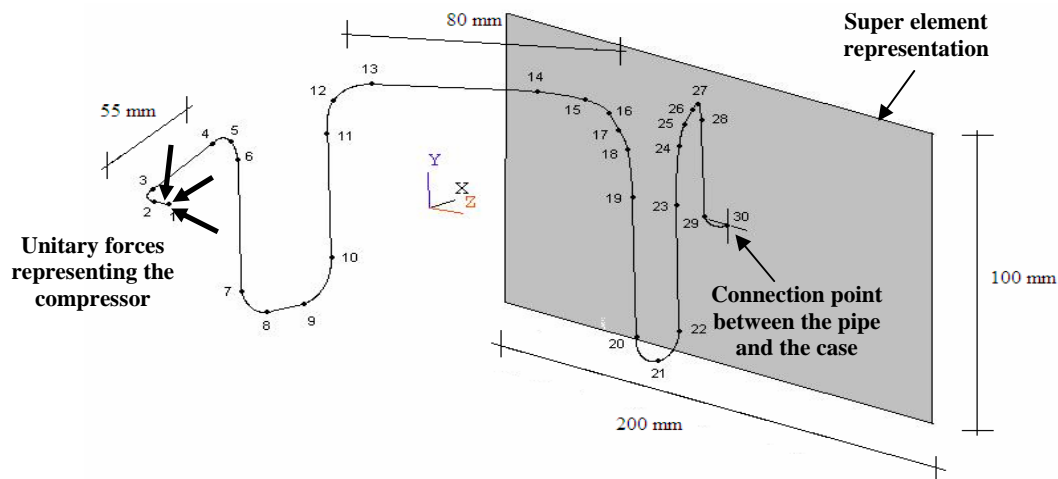

Figure 3 – FE model used during the optimization process.

The material properties attributed to the pipe model were: Young Modulus $= 2.1x10^{11} N/m^2$; Poisson Coefficient = 0.32; Density = 8370 Kg/m^3. For the case model the attributed material properties were: Young Modulus $= 1.9x10^{11} N/m^2$; Poisson Coefficient = 0.30; Density = 9070 Kg/m^3. These properties were found experimentally. Both components use a proportional damping model with a value equal to 0.001 s^{-1}. This damping value was chosen so that a modal response was clearly evident in the structure's response, but not so low such that large resonant peaks caused noise problems due to a large dynamic range.

3. OPTIMIZING WITH EVOLUTIONARY ALGORITHMS

To describe any optimization method it is necessary, first of all, to present some keywords that compose this field. In optimization, the parameter known as **objective function** is maximized or minimized in order to increase the performance of the system. This is achieved by the adjustment or combination of other parameters known as **optimization variables**. The domain covered by all the combinations of optimization variables is known as **search space**.

The goal is to search for the best combination of optimization variables available in the search space. Most of the optimization methods, i.e. classical methods, search for the best combination based on the gradient of the objective function. On the other hand, the evolutionary optimization methods (Genetic Algorithms, Simulated Annealing, etc…) are a stochastic-based class of optimizers (Bäck, 1996), that are not random searchers but have random elements in their algorithms. This permits them to be used in problems where the search space is multi-modal and discontinuous (Keane, 1996).

3.1 Optimizing the Discharge Pipe

The evolutionary optimizer used in this paper is known as Genetic Algorithm. This method is based on the Darwinian model of nature evolution (Holland, 1975) and is carried out, basically, by four operators: generation, fitness, cross-over and mutation.

The first operator (generation) randomly generates a population of ten discharge pipes, each one having a different combination of variables. The variables are the coordinates of 28 from the 30 keypoints that form the pipe geometry (See Figure 3). The keypoints 1 and 30 remain at the original coordinates due to the boundary conditions (vibration source and system case). Considering that each keypoint needs three variables to describe its position in the space, the total amount of variables of the system is 84. Each keypoint may have its position modified in one direction by the addition of an integer number that can vary between –2 mm to 2 mm. That means that each variable has a search space of 5 possibilities. Considering that all the variables have the same variation range, the system search space has $5x10^{54}$ possible combinations, i.e. the number of possible geometries achieved by combining the optimization variables.

The next operator (fitness) uses FEM to calculate the mean power flow, $\overline{W}_{IN}$, from each of the ten geometries previously generated. The geometries receive a probability number according to their performance, so that the sum of probabilities is equal to 1. Naturally, the geometries with better performances (low mean power flow value) have greater probability numbers. Ten geometries are then selected for the next operation. The geometries with higher probability numbers are more likely to be repeatedly selected.

The selected geometries are used in the next operator (cross over). The goal here is to create a new generation of geometries by exchanging characteristics (genes) between the previously selected ones. This exchange is achieved by turning each variable into a binary string. The concatenation of the 84 strings that represent a geometry is called a chromosome. The chromosomes are mated in pairs. Two matting chromosomes swap informations (genes) beyond a crossover point randomly selected, and two offspring thus result. At the end of this process a generation of ten new chromosomes is obtained, each of them representing a new geometry.

The new chromosomes are then used in the next operator (mutation). This operator consists on a random bit change in a chromosome with small probability. This provides random diversity in the evolution and helps to prevent premature convergence before too little evolutionary experience has been gained (Goldberg, 1989). At last, the chromosomes are transformed into geometries again, represented by decimal numbers and then submitted to the second operator, starting the iterative process. The iteration continues until a criteria of convergence is reached.

4. RESULTS

This section presents the results obtained for four optimization processes considering different frequency bands. The selection of the best geometry at the end of each process considered only their performances in terms of objective function. The geometries that showed the minimum values for the mean power flow were selected. Such criteria did not take into account the sensitivity of each geometry to small geometric perturbations, which could completely change their performance in terms of objective function. Very sensitive geometries are not feasible in practice due to geometric uncertainties inherent to fabrication processes. Keane (1995) proposes an interesting method to evaluate the robustness of optimal design solutions.

Table 1 shows the frequency resolution and the number of frequencies used to calculate the mean power flow for each process. Table 2 presents the results of the mean power flow, in watts, for the ten finalist geometries from each optimization process.

Table 1 – Frequency resolution and number of frequencies used to calculate the averaged power flow

Frequency band	Δf	Number of frequencies
6,0 – 6,25 kHz	5 Hz	50
6,0 – 6,50 kHz	5 Hz	100
6,0 – 7,0 h kHz	5 Hz	200
6,0 – 8,0 kHz	5 Hz	400

Table 2 – Averaged power flow of the original and finalist geometries.

Geometry	Band 1 6,0 – 6,25 kHz	Band 2 6,0 – 6,50 kHz	Band 3 6,0 – 7,0 kHz	Band 4 6,0 – 8,0 kHz
1	$0{,}304 \times 10^{-5}$	$0{,}323 \times 10^{-4}$	$0{,}551 \times 10^{-5}$ *	$0{,}230 \times 10^{-5}$ *
2	$0{,}304 \times 10^{-5}$	$0{,}104 \times 10^{-2}$	$0{,}551 \times 10^{-5}$	$0{,}322 \times 10^{-5}$
3	$0{,}170 \times 10^{-5}$ *	$0{,}154 \times 10^{-4}$ *	$0{,}551 \times 10^{-5}$	$0{,}230 \times 10^{-5}$
4	$0{,}175 \times 10^{-5}$	$0{,}149 \times 10^{-3}$	$0{,}753 \times 10^{-5}$	$0{,}455 \times 10^{-5}$
5	$0{,}304 \times 10^{-5}$	$0{,}359 \times 10^{-4}$	$0{,}689 \times 10^{-5}$	$0{,}297 \times 10^{-5}$
6	$0{,}377 \times 10^{-5}$	$0{,}323 \times 10^{-4}$	$0{,}551 \times 10^{-5}$	$0{,}297 \times 10^{-5}$
7	$0{,}170 \times 10^{-5}$	$0{,}149 \times 10^{-3}$	$0{,}551 \times 10^{-5}$	$0{,}122 \times 10^{-4}$
8	$0{,}304 \times 10^{-5}$	$0{,}323 \times 10^{-4}$	$0{,}551 \times 10^{-5}$	$0{,}305 \times 10^{-5}$
9	$0{,}281 \times 10^{-5}$	$0{,}149 \times 10^{-3}$	$0{,}721 \times 10^{-5}$	$0{,}297 \times 10^{-5}$
10	$0{,}304 \times 10^{-5}$	$0{,}323 \times 10^{-4}$	$0{,}689 \times 10^{-5}$	$0{,}321 \times 10^{-5}$
Original pipe	$0{,}36 \times 10^{-2}$	$0{,}18 \times 10^{-2}$	$0{,}23 \times 10^{-2}$	$0{,}32 \times 10^{-2}$

(*) Best geometries according to the used criteria

4.1 Power Flow Spectra Results

The spectra presented below (Figures 4, 5, 6 and 7) show the comparisons between the power flow of the original and optimized pipes, represented by continuous and dashed lines, respectively. The regions in blue represent the bands in which the optimization processes were carried out. At the right side of each spectrum the history of the objective function is presented against the number of generations, which shows the value of the best objective function achieved after each generation.
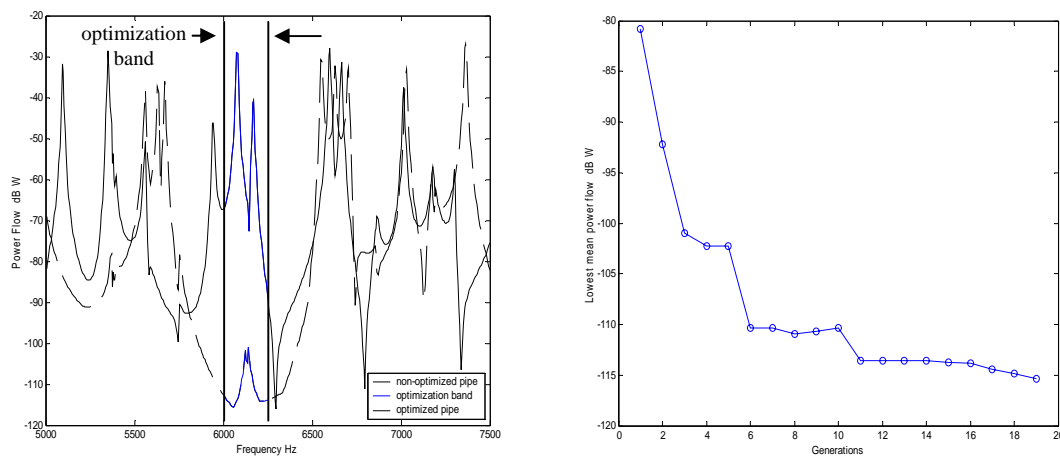

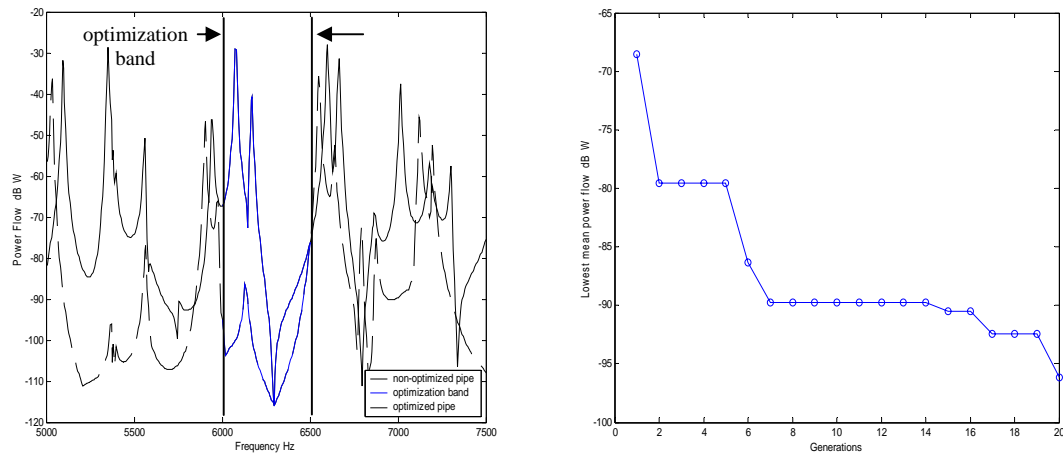*Figure 4 – Power flow spectrum comparison and optimization history for the band within 6.0 kHz to 6.25 kHz.*

International Compressor Engineering Conference at Purdue, July 12-15, 2004

Figure 5 – Power flow spectrum comparison and optimization history for the band within 6.0 kHz to 6.50 kHz.
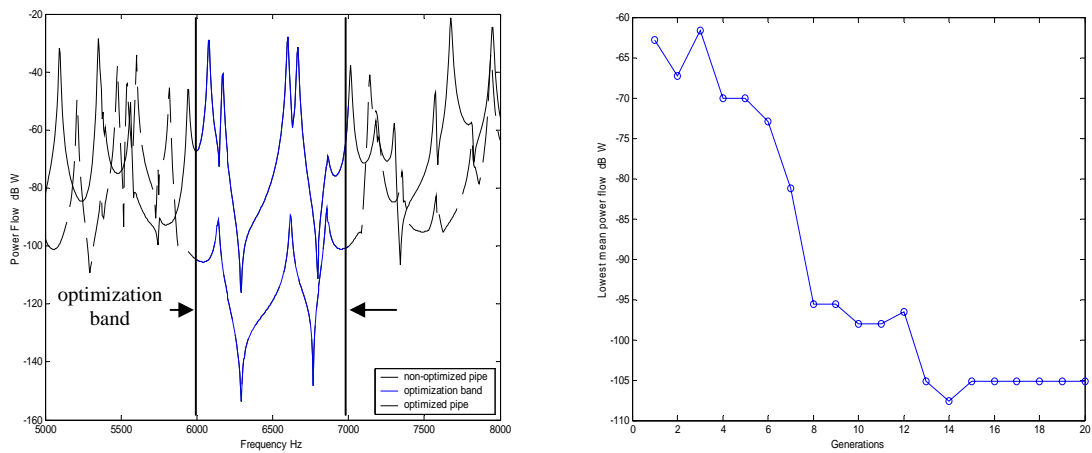

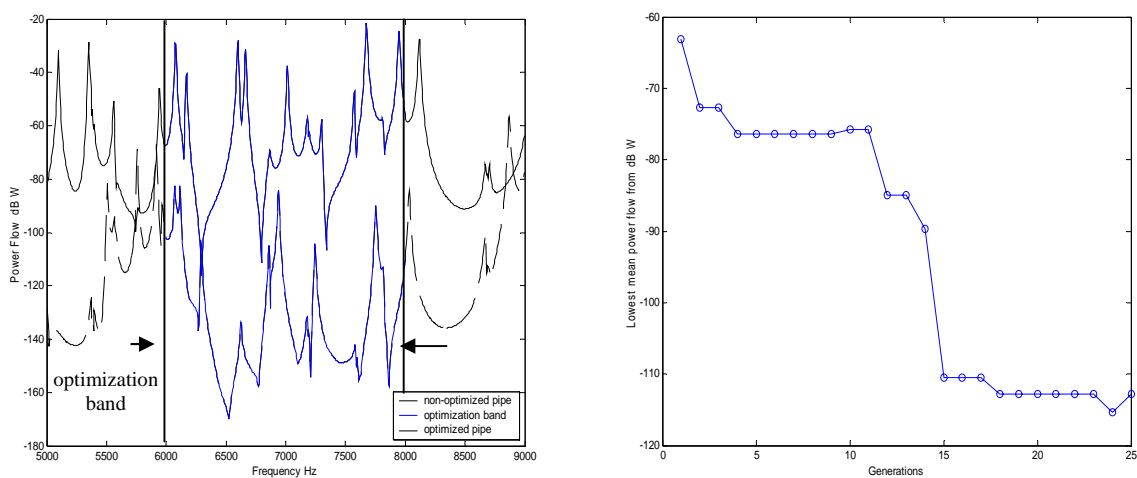*Figure 6 – Power flow spectrum comparison and optimization history for the band within 6.0 kHz to 7.0 kHz.*

Figure 7 – Power flow spectrum comparison and optimization history for the band within 6.0 kHz to 8.0 kHz.

International Compressor Engineering Conference at Purdue, July 12-15, 2004

4.2 Best Geometries

The best geometries obtained for each frequency band are shown below in isometric overview. Figure 13 shows the original geometry (non-optimized), so that the geometrical modifications suffered by the optimized pipes can be evaluated by comparison.
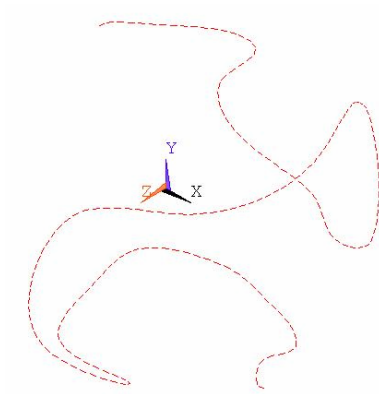

Figure 8 – Best geometry for the optimization band between 6.0 kHz and 6.25 kHz
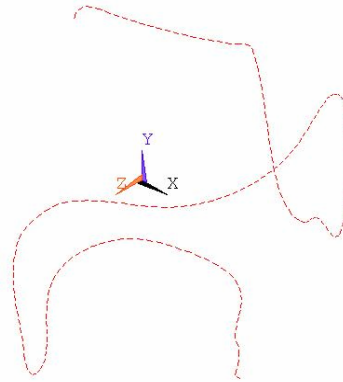

Figure 9– Best geometry for the optimization band between 6.0 kHz and 6.50 kHz
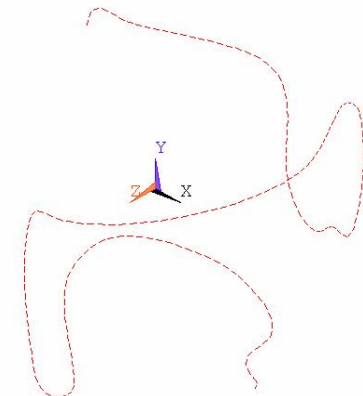

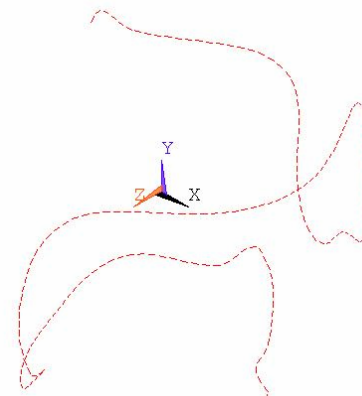*Figure 10 – Best geometry for the optimization band between 6.0 kHz and 7.0 kHz*

Figure 11 – Best geometry for the optimization band between 6.0 kHz and 8.0 kHz

Figure 12 – Original pipe geometry.

International Compressor Engineering Conference at Purdue, July 12-15, 2004

. **5. CONCLUSIONS**

In this work, no correlation between optimization bandwidth and mean reduction of the objective function (mean power flow) was reported. However, such correlation in which the wider the bandwidth the smaller the mean reduction in the objective function, was reported by Elliot (2000) in the optimization design of a 2-D truss using Green functions as objective function evaluator.

The optimizations in wide bands here performed are achieved by the expense of increases in power flow outside the optimization window (blue regions). This can be explained by the optimization mechanism, which is though to be by changing the many modes of vibration of the structure such that the power flow is affected in two ways. Firstly, moving the resonant responses out of the optimization region. In this case, increasing the modal density outside the optimization limits. Secondly, the structural mode are aligned as to destructively interfere.

The sensibility of the chosen structures due to geometric perturbations that may occur within the fabrication tolerances was no studied in this paper. Therefore, one has no guarantee that the chosen structures will have the same performance after manufactured.

Even so, the results obtained seem to be workable in obtaining structures with broad band filtering characteristics.

REFERENCES

Bäck , T., 1996, Evolutionary Algorithms in Theory and Practice, Oxford: Oxford ,University Press.

Beranek, L., L., and Vér, I., L., 1992, Noise and Vibration Control Engineering: Principles and Applications, John Wiley & Sons, INC.

Elliott, S, J., 2000, Robustness of Optimal Design Solutions to Reduce Vibration Transmission in a Lightweight 2-D structure: Part I, *Journal of Sound and Vibration, vol. 299*, pg. 505-528.

Goldberg, D. E., 1989, Genetic Algorithms in Search, Optimization and Machine Learning, Cambridge, MA: Addison-Wesley.

Gorman, D. J.,1975 , Free Vibration Analysis of Beams and Shafts, John Wiley & Sons, INC.

Holland, H., J., 1975, Adaptation in Natural and Artificial Systems: An Introductory Analysis with Applications to Biology, Control and Artificial Intelligence, Ann Arbor, MI: The University of Michigan Press.

Keane, A., J., 1995, Passive Vibration Control Via Unusual Geometries, *Journal Of Sound And Vibration,* vol. 185, pg. 441-453.

Mead, D., J., 1988, Passive Vibration Control, John Wiley & Sons, INC.

Silva, A., R., S., Lenzi, A. and Baars, E., 2003, Using Genetic Algorithms to Minimize Power Flow, *XII Seminário de Elementos e Encontor de Usuários do ANSYS –* SMARTTECH, São Paulo.

Silva, A., R., S., Lenzi, A. and Baars, E., 2004, Minimization of Power Flow in a 3-D Structure Using Genetic Algorithms, (submitted) *VI SIMMEC, Itajubá.*

ACKNOWLEDGEMENT