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COMPUTER-AIDED DESIGN AND EVALUATION OF LOW-FREQUENCY EFFECTIVE MUFFLERS

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INTRODUCTION

Fluidborne noise in machines and piping at low frequencies often influences thermal performance and flow characteristics, and could cause structural vibration and reliability problems. The severity of the problem may depend on many factors such as noise magnitude and phase, source characteristics, fluid flow system, coupling of acoustical and structural systems, fluid flow-sound interaction, etc.

DESIGN REQUIREMENTS FOR LOW FREQUENCIES

A muffler/manifold designer is often faced with the following design requirements and recommendations. Some typical values, based on author's experience with a few fluid machines are also given; these values vary with the specific application.

1. At a given location, Lp (See Nomenclature for definition) should not exceed the acceptability criterian based on structural vibration considerations. For the example cases given here, Lp should be lower than 130 dB re 2×10^{-5} Pa over 0-200 Hz range.

2. With the introduction of a muffling system, machinery thermofluid performance penalty should be minimum. In our cases, drops in η_E and η_F should be less than 5%. This requires that only reactive muffling elements be chosen with no or very little built-in energy dissipation mechanism.

3. Limited muffling space is usually available; consequently smaller volumes and lower expansion ratios yield little TL at the low frequencies. For example, in our case area expansion ratio usually does not exceed 10, and the product of wave number and length is generally smaller than unity; consequently, only 0-12.5 dB TL could be obtained over 0-200 Hz for a simple expansion chamber muffler [1].

4. Muffler/manifold system should be acoustically effective over a broad range of machinery operating conditions. Especially, TL values should not be very sensitive to the fluid flow conditions [2].

5. Finally, acoustic resonances (and their coincidence with mechanical resonances as well) should be avoided. This can be accomplished by shifting the acoustic natural frequencies away from the excitation frequencies. In the case of coupled machines, in-parallel or in-series, constructive acoustic interference mechanisms should be avoided.

Some of the above considerations are contradictory in nature, thus necessitating a design compromise. Because of the complexity of the design problem, a computer-aided design solution method is required.

COMPUTER SIMULATION MODEL

In order to predict the effectiveness of a muffler in a given operating system, we must be able to model not only the muffler but source, load and all dynamic interactions as well. Space limitations prevent us from discussing all mathematical models in detail some of which are described in Reference [3]. Here, a listing of various models is given as follows.

1. Fluid machine source model. This describes the basic thermo-fluid processes in time domain. It also generates the excitation function for the manifold and muffler system.

2. Source - Manifold/muffler interaction model. The purpose of this model is to incorporate the effect of back pressure on the source processes. This model also includes Fourier and Inverse Fourier transform algorithms.

3. Manifold and muffler models. Since we are interested only in the low frequency, the assumption of plane wave propagation holds good. We assume that manifolds and mufflers can be modeled, in the frequency domains, using linear acoustic transmission line theory. A building-block type of a transfer function approach is employed to build the overall model. In this, geometric and kinematic interactions due to multiple machinery excitations are also accounted for.

4. Load model. Mathematical models for various typical terminations are included.

In order to completely describe all interactions, an interative computer solution algorithm is employed. Table I presents the comparison of computer simulation model and experiment; note the excellent correlation.

COMPUTER-AIDED DESIGN: EXAMPLE CASES

Now we will discuss, in brief, four design studies. Our aim is not to present any novel muffler design; rather, we will illustrate the capabilities of the computer-aided design simulation.

1. Evaluation of a Low-Frequency Effective Muffler

Problem: Evaluate the acoustic performance of a reactive fluid machine muffling system - of high TL at low frequencies.

Results: Table II presents the results of muffling system performance for the first six harmonics of the running speed. We note that the NR values are high, like the TL data, at 60, 90, 150 and 180 Hz. Thus, we are able to predict the acoustic performance of the muffler in an operating system with a fluid machine source and a load.

2. Optimal Location of a Muffler

Problem: To investigate the effect of connecting tube length on muffler performance and to determine where the muffler should be located.

Results: Table III presents some typical IL results for various connecting tube lengths. We note that the selection of 0.07ℓ will lead to a significant increase in Lp at 60 Hz; conversely, at 120 Hz the selection of length ℓ causes problems. For this case, 0.75ℓ length seems to be optimum at 120 Hz provided we accept a slight increase in Lp at 60 Hz. Using the simulation, other suitable lengths could also be searched for.

Table I. Validation of the Computer Simulation Model

Quantity	Simulation compared to experiment, within
η_E	$\pm$ 4%
η_F	$\pm$ 2
Pressure drop in manifold & muffler system	$\pm$ 5%
Lp - in the manifold and upstream of muffler	$\pm$ 5 dB
Lp - downstream of muffler	$\pm$ 10 dB

Table II. Results of the Muffler Evaluation Study

	NR, dB					
	30 Hz	60 Hz	90 Hz	120 Hz	150 Hz	180 Hz
Predicted	5	16	15	7	13.5	10
Measured	7	16	17	5	13.0	19.5

Table III. Results of the Muffler Location Study

Muffler Location	IL, dB			
	60 Hz		120 Hz	
	Predicted	Measured	Predicted	Measured
.07ℓ	-12.5	-7.5	-4.5	-5.5
.75ℓ	-1.5	-4.0	6	14
ℓ	0	-2.0	-14	-15
ℓ = reference length used for relative comparisons				

3. Manifold Design for Thermofluid Performance Improvement

Problem: To redesign the inlet/outlet manifold plenum of a positive displacement fluid machine so as to improve its thermodynamic and fluid flow characteristics.

Results: Table IV shows the effect of increasing manifold plenum volume; we notice an improvement in η_E and η_F, and a corresponding decrease in Lp at the fundamental running speed.

Table IV. Results of the Manifold Design Study

Plenum Volume	η_E		η_F		Lp at 60 Hz, dB	
	Predicted	Measured	Predicted	Measured	Predicted	Measured
V *	1.005	1.000*	1.013	1.000*	2.5	0.0*
2V	1.012	1.030	1.043	1.036	−1.0	−0.5
3V	1.019	1.031	1.064	1.049	−5.5	−3.5
4V	1.021	1.031	1.077	1.054	−8.0	−5.5

*denotes reference quantity used for relative comparisons

4. Diagnosis of A Pulsations Problem for Coupled Machines

Problem: To diagnose the pressure pulsation problem of a system consisting of three fluid machines in parallel. These machines have a common manifold which is connected to the rest of the operating system through a pipe; it was observed that piping vibrations are worse when one or two machines are unloaded.

Results: Table V presents the results of this problem in the form of amplification achieved over the case when all machines I, II and III are loaded. We note that depending on the nature of the unloading, a unique amplification spectrum is obtained. From the simulation model, we identified this to be due to the interactions between individual manifold elements.

Table V. Results of the Coupled Machines Pulsations Study

Machine(s) Unloaded	Amplification*, dB					
	60 Hz		120 Hz		180 Hz	
	Predicted	Measured	Predicted	Measured	Predicted	Measured
II	2.5	1.5	17	2.0	−2.5	−1.0
III	4	10.0	−3.5	−2.5	4.5	4.0
II and III	8	11.5	9.5	4.5	−8	+1.0

*Ampl. over the case when all machines I, II and III are loaded.

NOMENCLATURE

IL = Insertion loss, dB

Lp = Sound pressure level, re 2×10^{-5} Pa

NR = Noise Reduction, dB

TL = transmission loss

η_E = Thermodynamic efficiency index

η_F = Fluid flow efficiency index

REFERENCES

1. D.D. Davis, Jr., et al, NACA Report No. 1192, 1954.
2. L.L. Beranek, Noise and Vibration Control, McGraw-Hill, 1971
3. R. Singh and W. Soedel, J. Sound Vib., 62(4), 125-143, 1979.