Noise optimization of a regenerative automotive fuel pump

J F Wang*, H H Feng, X L Mou and Y X Huang
School of Mechanical Engineering, Beijing Institute of Technology, Beijing, 100081 China
E-mail: wjfbit@163.com

Abstract. The regenerative pump used in automotive is facing a noise problem. To understand the mechanism in detail, Computational Fluid Dynamics (CFD) and Computational Acoustic Analysis (CAA) together were used to understand the fluid and acoustic characteristics of the fuel pump using ANSYS-CFX 15.0 and LMS Virtual. Lab Rev12, respectively. The CFD model and acoustical model were validated by mass flow rate test and sound pressure test, respectively. Comparing the computational and experimental results shows that sound pressure levels at the observer position are consistent at high frequencies, especially at blade passing frequency. After validating the models, several numerical models were analyzed in the study for noise improvement. It is observed that for configuration having greater number of impeller blades, noise level was significantly improved at blade passing frequency, when compared to that of the original model.

1. Introduction
Regenerative pumps have found applications in many industrial areas which require high heads at low flow rates including automotive and aerospace fuel pumping, booster systems, water supply, agricultural industries, shipping and mining, and chemical and food processing systems[1]. The fluid and NVH characteristics of regenerative fuel pump need more attention simultaneously. Both vibrations and noise can affect the regenerative pump performance and its life. In regenerative pumps, the sources of vibrations and noise may lie in hydraulic or mechanical aspects, but under normal operating condition, the blade passing frequency is the most usual excitation of vibrations and noise. This frequency results from the nonuniformity of the flow at the impeller outlet which is caused by the effects of the rotor-stator interaction.

Momentum exchange theory was used to explain fluid mechanism within regenerative pumps[2], and was improved by the following researchers[3-6]. Meanwhile, the fluid performance of regenerative pumps was improved by optimizing different parameters numerically[7-9] and experimentally[10-12]. Recent years, noise improvement has gradually got more attention[13,14]. Hence, it is important to conduct in-depth research on the noise performance of regenerative pump.

Therefore, the main purpose of this study is to explore the effects of factors such as the blade numbers on the noise performance of a regenerative automotive fuel pump. To achieve this objective, the CFD and CAA models were considered for the regenerative pump. For verifying the simulation results, the fluid and noise performance tests were applied. Accuracy of the simulations was assessed by comparisons to experimental data.

2. Regenerative fuel pump geometry and grid generation
The regenerative fuel pump-tank system consists of a fuel tank and a fuel pump module, as plotted in Figure 1. For CFD simulation, we just considered the flow domain in the fuel pump module, as shown in Figure 2. More detailed specifications are shown in Table 1, and sectional view of the impeller in Figure 3. Simplification was considered in the outlet domain.

For establishing grid independency, analysis was carried out using the meshing program ICEM (ANSYS Inc.), with number of elements restricted to approximately 1 million, 2 million and 4 million elements containing hybrid hexahedral/tetrahedral grids, and the grids were refined at the wall region where the flow properties change rapidly, the y+ value of the first grid point from the wall was adjusted to be less than 2. The mass flow rates of the three schemes and experiment are shown in Table 2. Since the variation in the mass flow rate values was less than 1.2% for models containing elements greater than 1 million, mesh size of approximately 1 million was adopted to save the computational time, and relative discrepancy between experimental and computational results was 1.9%.

### Table 1. Design specifications for the regenerative fuel pump.

| Specification          | Value (mm) |
|------------------------|------------|
| Impeller thickness     | 3.8        |
| Impeller diameter      | 32.8       |
| Blade thickness b      | 0.4        |
| Number of blades Z     | 47         |
| Chevron blade angle α  | 30         |
| Design mass flow rate  | 20.5       |
| Rotational speed       | 6000       |
| Pressure rise (kPa)    | 400        |

### Table 2. Grid independency test.

| Number of elements | Mass flow rate (g/s) |
|--------------------|----------------------|
| 1092018            | 20.11                |
| 2122816            | 19.99                |
| 4137091            | 19.85                |
| Experimental result| 20.5                 |

3. **Computational models set up**

3.1. **CFD model**

Three-dimensional RANS equations were solved in ANSYS CFX 15.0 to obtain the fluid performance. Gasoline was considered as the working fluid. The total pressure 0 kPa and 400 kPa were assigned at the inlet and outlet domain respectively, and the rotating speed, 6000 rpm, was set to the impeller domain. No slip wall condition was specified at the wall boundaries of the blades and the casing. The
turbulence is simulated using the k-ω-based shear stress transport (SST) turbulence model. A high-resolution scheme with second-order accuracy in space was used for the numerical scheme of the convection terms in the governing equations.

The interface between stationary and rotating domains was set to frozen rotor method in steady state computations, and to transient rotor stator method in unsteady state computations, respectively. The time step and coefficient loops for time scale control were set to $2.12766 \times 10^{-5}$ s and 10 times respectively in order to reduce all maximum residuals to a value below $10^{-5}$. This provided 10 time steps per impeller blade passage. Since the nature of flow is unsteady, it requires numerical analysis until the transient fluctuations of the flow field become time periodic as judged by the pressure fluctuations. In the present analysis this has been achieved after 5 complete rotations of the impeller. The pressure at the monitor points were recorded corresponding to each rotation of the impeller by time step advancement. Since the nature of flow is unsteady, it requires numerical analysis until the transient fluctuations of the flow field become time periodic as judged by the pressure fluctuations. In the present analysis this has been achieved after 5 complete rotations of the impeller. The pressure at the monitor points were recorded corresponding to each rotation of the impeller by time step advancement, as shown in Figure 4. The data extracted from the fifth revolution will be used for the following acoustic analysis.

![Figure 4. Monitor points and corresponding pressure fluctuations.](image)

3.2. CAA model

In this work, acoustic analysis was implemented using finite/boundary element method, vibration-acoustic coupling method and Automatically Matched Layer (AML) method in LMS Virtual. Lab Rev12. In this process, the pressure fluctuations on the inner wall of the fuel pump extracted from the unsteady flow were used for acoustic analysis.

![Figure 5. Acoustic domain of the pump-tank system.](image)  
![Figure 6. Acoustic field point.](image)

Figure 5 represents the whole acoustic domain for the acoustic analysis, which consists of pump boundary, oil pool boundary, air, gasoline and fuel tank domains. A tetrahedral grid system was constructed in the acoustic domain for acoustic analysis with approximately 136061 grid points to
satisfy the target element length. An observer point is set to measure the sound pressure, which is 200mm below the fuel tank, corresponding to the measure position, as shown in Figure 6.

4. Validation of the models
For validating the accuracy of the computational model, acoustic experiment of the regenerative fuel pump was implemented in the semi-anechoic room. The fuel pump was tested under installation condition, as shown in Figure 7. The 40AK type microphone and 26AK type preamplifier together were used to measure the sound pressure, located 200mm below the fuel tank. The LMS SCADAS III system was used for data acquisition.

Figure 7. Experimental set up. Figure 8. SPL comparison between simulation and experiment.

Figure 8 represents the sound pressure levels comparison between computational and experimental results at the monitor point. As we just considered fluid induced noise, the blade passing frequency noise was the interest we had in the fuel pump. Mechanical noise caused by rotor misalignment and electromagnetic noise caused by DC motor commutators were not considered, which related to the basic frequency and harmonic frequencies corresponding to low and medium frequency range. So the simulational results were lower than experimental results under 3000Hz. Between 3000 and 4000 Hz, sound pressure levels were higher in simulation than that in experiment, the probable reason was that some structures such as feed tubes, flange and damping layer were not considered. Similar trend between simulation and experiment was observed after 4000 Hz, especially at the blade passing frequency (BPF) where SPLs were 45.38 dB and 47 dB respectively, with a relative discrepancy about 3.4%.

5. Geometry modification
Effects of chevron blade angle $\alpha$, blade thickness $b$ and blade number $Z$ on noise performance of the fuel pump were studied. Different parameters are shown in Table 3, where $Z=47$, $b=0.4$ and $\alpha=30^\circ$ are the original parameters.

| Parameter | Blade number Z | chevron angle $\alpha$ | blade thickness b |
|-----------|----------------|------------------------|-------------------|
| Value     | 42             | 25$^\circ$             | 0.35 mm           |
|           | 47             | 30$^\circ$             | 0.4 mm            |
|           | 52             | 35$^\circ$             | 0.45 mm           |

Sound pressure level at the measure point was the criteria to evaluate different parameter modifications. Figure 9, Figure 10 and Figure 11 show the comparisons of the 1/3 octave band SPL spectra between numerical results of different parameters. It can be seen that around the BPF and higher frequencies, chevron blade angle and blade thickness have no much influence on the sound
pressure levels. On the contrary, the modification with 52 blades decreases the SPL significantly at high frequencies, especially at BPF. Figure 12 represents the comparison of the whole frequency band SPL spectra between 52 blades model and original model, value of SPL is decreased by 20.2 dB at BPF. At the same time, efficiency of the fuel pump only decreases from 56% to 53%.

6. Conclusion

The numerical analysis using CFD and CAA helped understand the fluid and noise performance of a regenerative pump. The CFD results of the mass flow rate only had a relative discrepancy of 1.9% compared to experiment, and the calculated sound pressure levels agreed well with experimental results at high frequencies, especially at blade passing frequency only a relative discrepancy of 3.4% existed. Three parameters were selected to study the effects on the noise level of the regenerative fuel pump. The modification with 52 blades can improve the noise level, and decreases the SPL by 20.2 dB at BPF, only decreases the efficiency from 56% to 53%.

7. References

[1] Y Teshome, E Dribsa 2007 M.S. thesis University of Addis Ababa, Addis Ababa, Ethiopia
[2] M Badami 1997 SAE Technical Paper Series No. 971074 45-55
[3] J W Song, A Engeda, M K Chung 2003 Proc. IMechE A J. Power and Energy 217 311-321
[4] M M Raheel 2003 Ph.D. thesis Michigan State University East Lansing MI USA
[5] I S Yoo, M R Park, M K Chung 2005 Proc. IMechE A J. Power and Energy 219 567-581
Acknowledgements
This research was financially supported by the National Natural Science Foundation of China (Grant No. 51006010) and a joint China-UK research programme “111” (No. B12022).