Predicting Noise Radiation for Full Frequency Engine Design

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Abstract

In various industries, engine design has an important impact on noise radiated by a manufactured product. In recent years, low and mid frequency analysis of engine has shown that many design decisions influenced negatively the acoustic performance of engines. To counteract this problem, design engineers have started to design engine covers that actually reduce the noise radiated before it gets too far from the engine. These methods basically modify the path the noise would take to get to a vehicle occupant ear. Another approach is to modify the design of the engine construction itself in order to reduce the noise generated at the source. This paper presents a method to predict radiated noise from an engine for the full frequency domain of analysis (0-10000 Hz). This approach combines the Finite Element Method (FEM) to represent the structure and its load cases and the new Fast Multipole Boundary Element methods (FMM-BEM) to represent the fluid surrounding the engine. This approach allows engineers to add acoustics to their existing structural predictions and also evaluate the effect of adding covers in the vicinity of an engine. Typical engine noise radiation results with structural changes along with cases with and without acoustic covers are presented and discussed.

Introduction

Traditionally, design of automobile engine radiated noise performance has been based primarily on the prediction of vibration levels. Controlling vibration levels and removing as much as possible vibration peaks provided a simple way of preventing noise issues at an early stage of design. Today, new advances in radiated noise predictions make acoustic predictions also available at an early stage of design and can contribute to better design in the whole frequency range of interest.

Gumerov and Duraiswami [1] have developed a fast multipole method (FMM) which is an accelerated iterative solution of the boundary element method (BEM) for the Helmholtz equations in three dimensions. This method has been implemented in the commercial software VA One [2]. This method has gained in popularity in recent years because it significantly increases the speed of computation, solving 1 million DOFs in the time it took to solve the 10,000 DOFs with BEM. It also reduces the amount of memory used to store BEM matrices (O(n2) vs O(n)) and the maximum number of nodes it can handled has increased from 10,000 to 1 million nodes.

Vibro-acoustics applications have very specific accuracy and cost criteria. Algorithms that work well for electromagnetics or Laplace's equation do not necessarily perform well for vibro-acoustics. The best algorithms are optimized to trade off accuracy requirements that are unique to vibro-acoustics having the appropriate balance between BEM discretization errors, FMM approximation errors, iteration/convergence errors in order to not "overcompute" certain parts of the solution. FMM is based on the use of iterative solvers, the convergence is therefore an important performance criteria. Convergence, accuracy, memory usage and cost all vary with "kD" values (where k is the wavenumber and D the domain size). It is important to make sure that an FMM solver performs well over a large range of kD values when applied to vibro-acoustics. These are the basis for the VA One implementation of FMM BEM.

Modelling process

The global process to follow to predict acoustic response of an engine includes the definition of the loads, the representation of the vibro-acoustic paths and the post-processing of the acoustic results. In fact the vibro-acoustics model can be described as a structural path and an acoustic path (Figure 1).

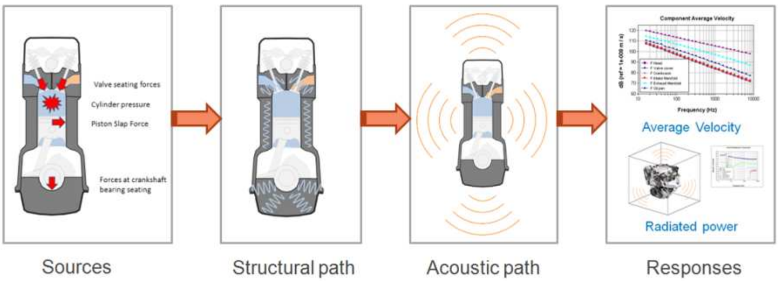


Figure 1: From source to receiver modelling process: i) Sources and Structure are represented in FEM and ii) Acoustic path is represented in FFM-BEM

The different steps of the modelling process are described in details in the next sections.

Sources

This paper focuses on the addition of acoustics to the current process of engine vibro-acoustics performance evaluation. It assumes that the main forces acting on the structure are well understood and represented in the FEM structural model created in the current process. The main excitations are forces at crankshaft bearings coming from the internal moving parts such as pistons, cranks, etc. It should also include the piston slap excitations, the valve seat and bearing forces and the contribution from the combustion pressure. In this study, a simple 1/3 octave force spectrum is used as structural excitation to illustrate the modelling process and analysis of the results.

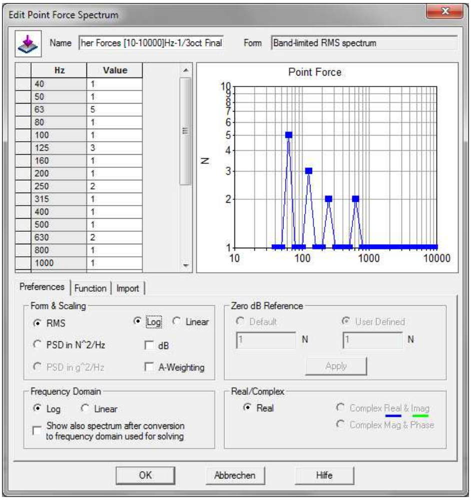


Figure 2: Force spectrum used for excitation

Structural model

The FEM model of the structure is composed of five different parts. The valve cover, valve train, engine block, bulkheads and the oil pan. This partitioning is arbitrary and any other partitioning would also be valid. One could have chosen to split the parts further into exhaust and intake sides. This would allow a more refined identification of parameters such as panel radiation efficiency and average velocity (Figure 3 and Figure 4).

The FEM mesh resolution is sufficient to allow the computation to be performed to 10 kHz. The structure's first mode is at 1169 Hz and the total number of modes up to 10 kHz is 263 modes.

To investigate structural design changes effect on radiated noise, different FEM models were built. From a nominal configuration, 3 modifications were made to the engine structure: the cylinder coolers were rebuilt into a round shape, a square shape and beads were added to the oil pan (Figure 5 and Figure 6). The beads are 260mm long, 24 mm wide and have a radius of 12mm.

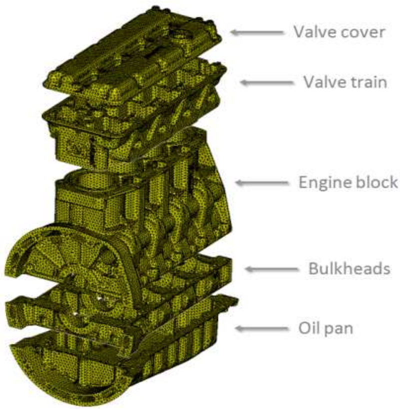


Figure 3: Description of engine studied

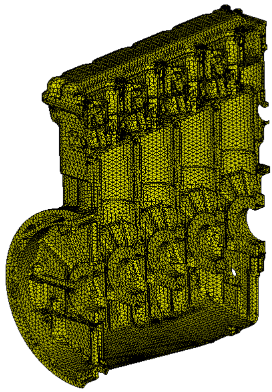


Figure 4: Structural FEM model internal view

Once modal basis is computed for all cases, a skin only model is used to compute the average velocity on the outer skin of the engine. This can be done since only the outer node velocities are necessary to compute the radiation of noise from the structure. This skin model has the advantage of solving very quickly since it does not include any of the interior nodes except where the excitations are applied.

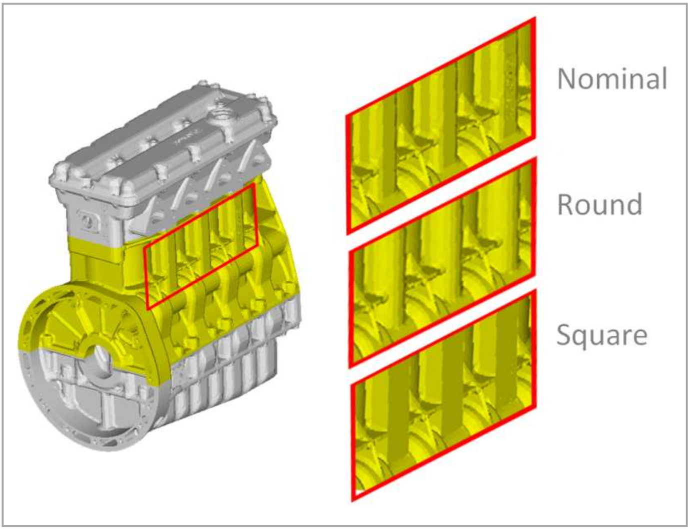


Figure 5: Cylinder cooler shape modification

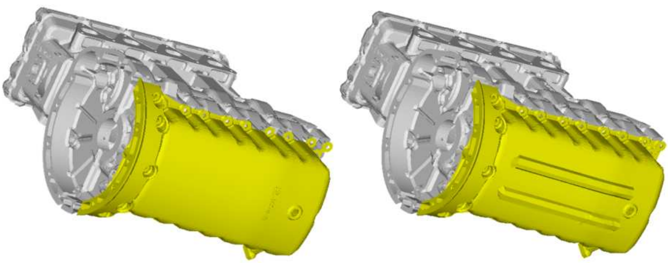


Figure 6: Oil pan reinforcement using beads: Nominal (left) and beaded (Right)

Acoustic model

To represent the fluid around the engine, the FMM-BEM method has been selected. This method provides a detailed description of the scattering of waves around the complex contour of the engine and is therefore appropriate to predict radiated noise by the vibrating engine.

Furthermore, its implementation into VA One provides an easy and quick way of creating the BEM fluid element. The boundary mesh is created automatically using a shrinkwrapper that creates a quality mesh with uniform element size (Figure 7). Finally, the FMM BEM is multicore aware and will automatically make use of any number of cores available to compute the BEM intermediate results.

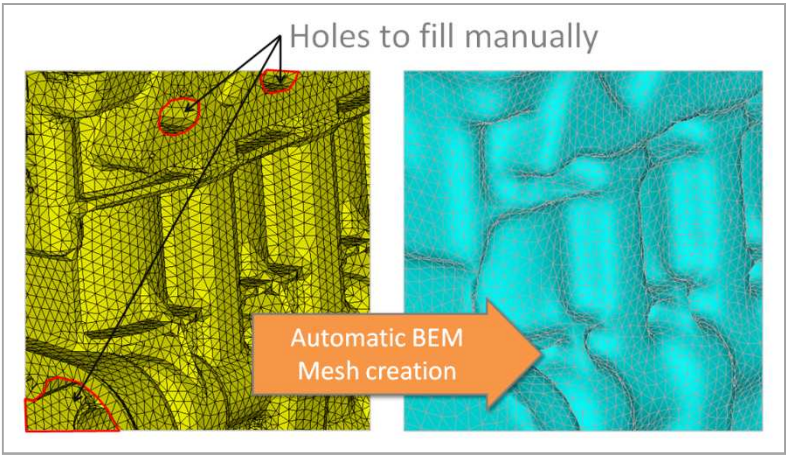


Figure 7: Smooth BEM mesh automatically created with uniform element size for optimum computation speed

The intermediate results are the relationships between the boundary mesh nodes, the BEM object and the data recovery nodes predefined by the user. This allows the computation of total power radiated and SPL at any predefined point in space. Radiation efficiency is also computed automatically.

A projection algorithm is used to project the velocity computed on the skin of the structure onto the BEM acoustic mesh eliminating the need to build compatible meshes.

Computation Process

The general computation process is illustrated in Figure 8. The iterative process is performed until all the load cases are completed. Usually these include different engine RPM and torque. These loads can be defined as forces on the FEM structural model or as modal participation factors. Using the precomputed modal basis, the velocity on the outer skin of engine can be computed. Note that the modal basis needs to be computed only once.

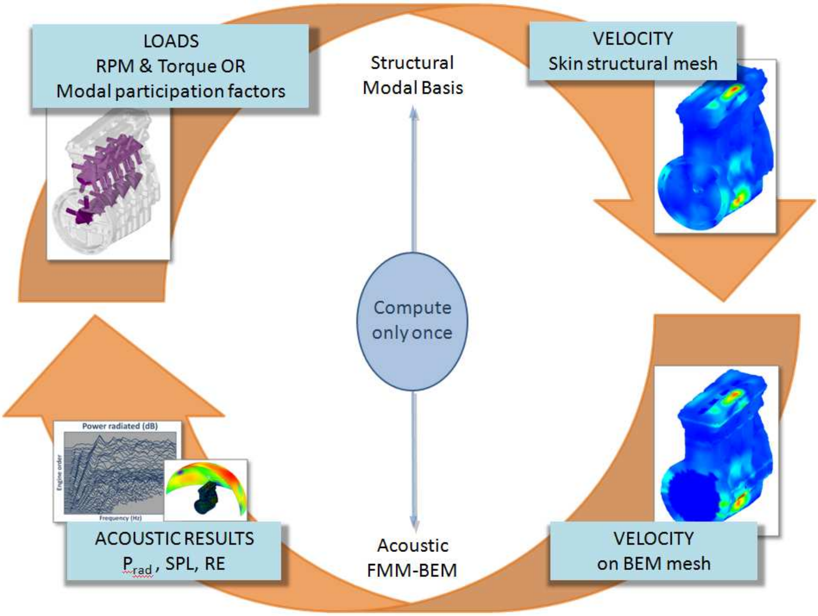


Figure 8: Loadcase iterative computation process: Top - Structure velocity computation, Bottom - Acoustic computation

Once the velocity response is computed, it is automatically projected onto the acoustic BEM mesh. In Figure 8, note the location of the transmission which is not coupled to the BEM fluid (dark blue area), since no radiation from this part will occur in the fully assembled powertrain. Using the acoustic FMM-BEM intermediate results, the acoustic responses can be computed. Note that the BEM intermediate results need to be computed only once.

Results

Skin velocity

The average velocities have been computed for the different structural changes discussed earlier.

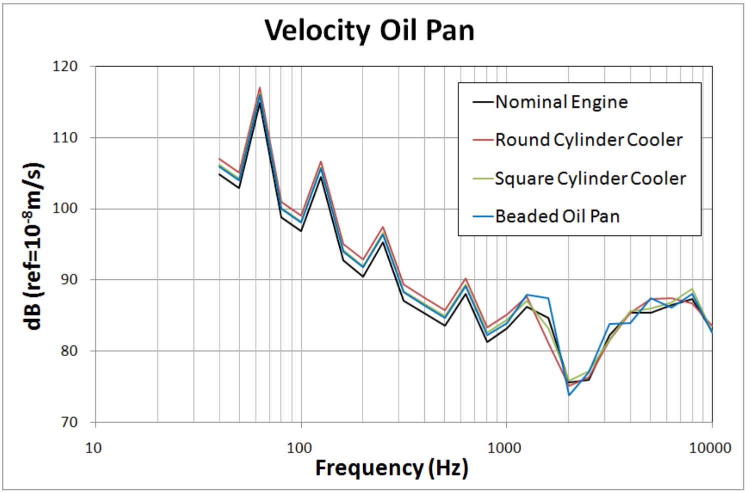


Figure 9: Oil pan average velocity

In Figure 9 note that even though the structural changes have been made on the engine block, the average velocity of the oil pan is impacted on the whole frequency range. The same is through for the average velocity of the engine block when a change is made to the oil pan stiffness (Figure 10).

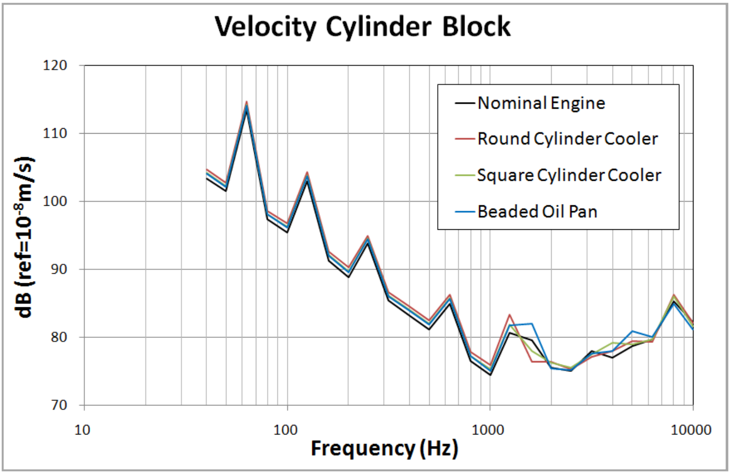


Figure 10: Engine block average velocity

Sound power radiated

The sound power radiated by the engine is presented in Figure 11 for the configurations considered. The sound power radiated graph alone does not provide insight as to what are the main contributors to the total sound power radiated. The same model can be used to compute panel contribution from each structural part. Figure 12 shows the panel contribution for the FEM parts previously described. Note that at lower frequency where no modes are present in the engine, the main contributor is the engine block. At higher frequency, the oil pan is dominating the radiated noise levels. It has been shown that sound power radiated is proportional to the product of the average velocity of a panel and its radiation efficiency.

$$\Pi_{rad} = \sigma A \rho_0 C v_{rms}^2$$

Therefore to control radiated power, one can influence average velocity or radiation efficiency of a structural part. Considering only velocity as an indicator of sound power radiated might lead to large errors. Below 1000 Hz the velocity is fairly high but on the contrary radiation efficiency is fairly low, therefore assuming that sound radiated would be solely proportional to velocity as in the so-called ERP (Equivalent Radiated Power) method, one would dramatically overestimate the radiated power.

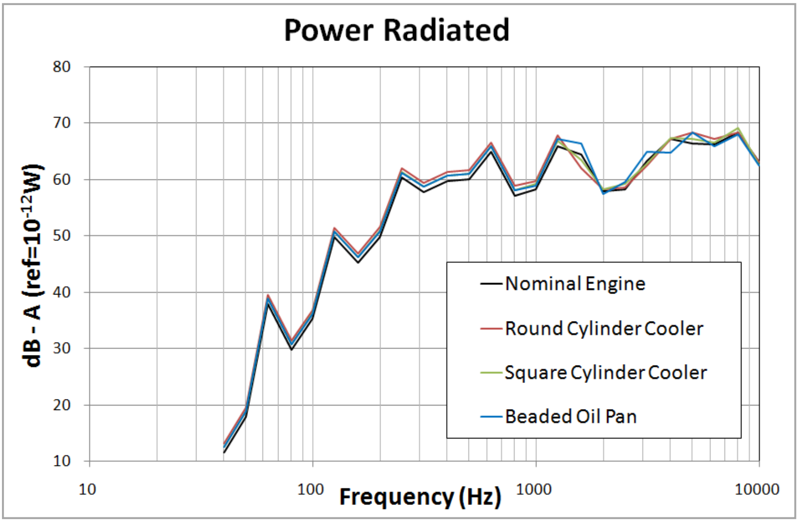


Figure 12: Sound power radiated for different structural design changes

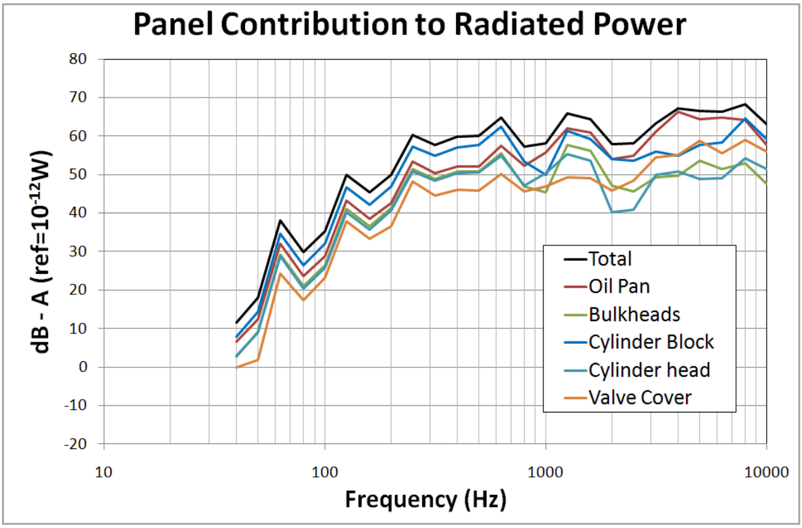


Figure 11: Panel contribution from each FEM subsystems

Radiation efficiency

Radiation efficiency is defined as “the acoustic power radiated by the plate into a half space, divided by the acoustic power that an infinite piston (all parts vibrating in phase) would radiate into the same half space if it were vibrating with the same rms velocity as the plate” [3]. The radiation efficiency can be viewed as the ability of a panel to radiate noise.

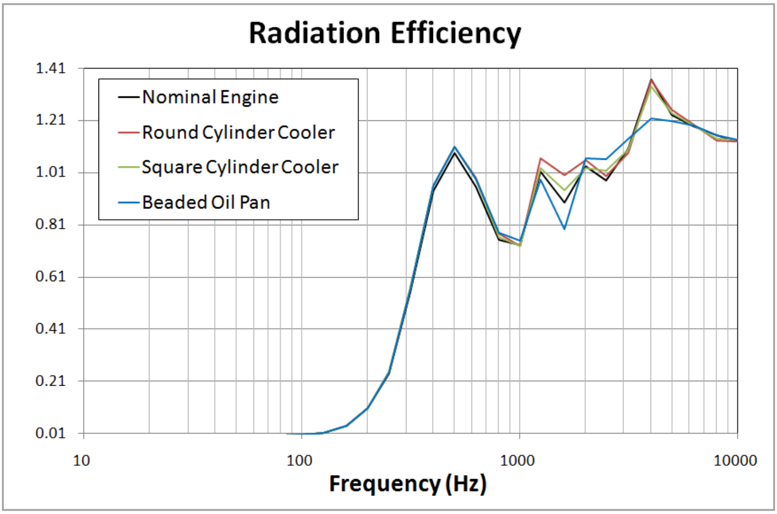
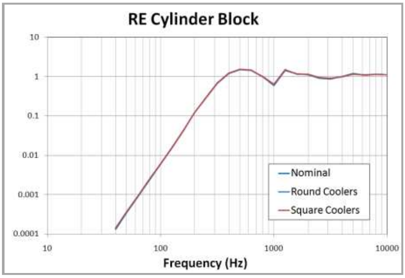


Figure 13: Average Engine Radiation efficiency

Figure 13 shows the radiation efficiency of the different FEM parts. Note that at low frequency, radiation efficiency is low and that at higher frequency, RE increases significantly.

It is of interest to examine the change in radiation efficiency a structural design change can bring. In the present case, the radiation efficiency is barely modified by the structural design changes as can be seen in Figure 14 and Figure 15.



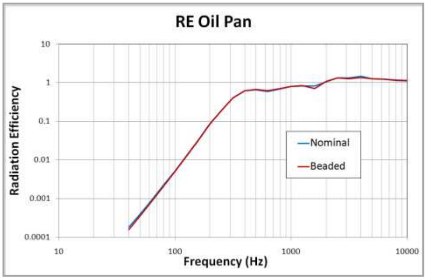


Figure 14: Radiation Efficiency of engine block variations

Figure 15: Radiation Efficiency of oil pan variations

Pressure distribution

SPL can be predicted anywhere around the engine at predefined node locations. The nodes can be part of a grid where SPL will be displayed in a contour plot. Pressure spectrum can also be retrieve when virtual microphone locations are defined before the computation of BEM intermediate results.

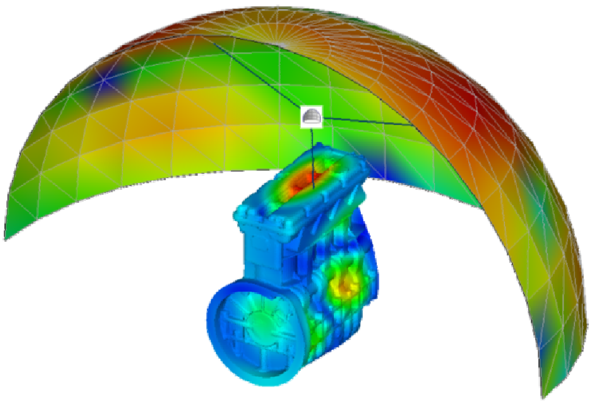


Figure 16: Pressure distribution as a contour plot

Computation time

The main advantage of the FMM-BEM approach is the computation speed. The results generated in this paper were generated for a single design change in less than an hour once the structural modal basis is computed. For a different frequency domain of interest, the computation time is also provided par frequency. Note that these are performed on a computer with 2 dual core processors of 3 years of age.

	Time (sec/freq)	Time (Sec. for 0-10 kHz)
Structural forced response	15.4	385
FMM intermediate results	112	2800
Acoustic response	1.24	31
Total	129 sec.	3216 sec.

Conclusion

This paper has presented an approach that allow engine designer to use combined vibration and acoustic data to base design decisions on. It has also demonstrated that in order to evaluate sound power radiated by an engine, it is not sufficient to consider that the radiated noise is directly proportional to the vibration level; it is the product of the radiation efficiency and the average velocity that determines the level of noise generated by a vibrating body.

Acknowledgements

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References

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