REDUCING NOISE IN AN ELECTRO-HYDRAULIC VALVETRAIN

Michael Söderback[1], **Pauli Valkjärvi**[1], **Xiaoguo Storm**[1], **Kari Saine**[2]**, Maciej Mikulski**[1]

[1] University of Vaasa
School of Technology and Innovations (TEKKINO)
Efficient Powertrain Solutions (EPS) research group
firstname.lastname@uwasa.fi

[2] Wärtsilä Finland Oy
firstname.lastname@wartsila.com

Abstract

The issue of noise, vibration and harshness (NVH) from machinery is an increasing societal concern. Knowledge regarding the levels and origin of vibroacoustic phenomena is essential when striving to reduce the NVH footprint of an internal combustion engine. Variable valve timing allows adjustment of the timing of a valve lift event to improve internal combustion engine performance and fuel economy, and to reduce emissions. This study mounted an electro-hydraulic valve actuation system on a medium-speed Wärtsilä 4L20 marine diesel engine. Accelerometers and a microphone were used for vibroacoustic measurements of the valve opening and closing events. Different valve profiles were designed and the associated noise and vibration of each profile were evaluated. Measurements were performed in the crank angle domain to enable detailed analysis of vibroacoustic phenomena, together with other events related to the operation of the valves. This study found that the NVH footprint of the variable valvetrain was reduced by altering the valve profile design. Lower levels of noise and vibration were associated with valve profiles that had a softer landing. A sound pressure level reduction of up to 8 dB was achieved for the quietest valve lift profile compared with the noisiest.

1 INTRODUCTION

Efforts to create ever-quieter combustion engines include a focus on reducing noise, vibration and harshness (NVH) of the valvetrain. In this context, noise caused by valve closure in particular plays a key role. In pursuit of high volumetric efficiency, a valve must be opened and closed as quickly as possible within the valve timing constraints imposed by the target of reducing engine pumping losses. Valve speed is even more important when using advanced variable valve actuation (VVA) strategies to support in-cylinder combustion [1] or exhaust gas thermal management [2]. However, valvetrain speed targets are often accompanied with trade-offs concerning NVH and durability issues.

A study by Hanaoka and Fukumura [3] found that valvetrain noise manifests mainly at low engine-speeds. They concluded that the impact noise is generated by impulsive excitation that occurs at valve opening and closing. Furthermore, the sound pressure generated by the closing event is proportional to the impact velocity. Working on a similar topic, Savage and Matterazzo [4] concluded that cam design is the most important parameter affecting valvetrain noise.

Suh and Lyon [5] identified two major sources of valvetrain vibrations. One is the valve seating impact; the other is the dynamic interaction force between the camshaft and tappet. This study included a method for separating valve seating noise and valve opening noise. In their analysis of vibrations in an internal combustion engine (ICE) valvetrain, Norton et al. [6] concluded that the most effective way to determine the vibratory response of a valvetrain is by means of acceleration measurement. They also concluded that both the time and the frequency domain information should be used for effective analysis of a valvetrain´s dynamic vibroacoustic behaviour. In his work on application of time-frequency representation techniques to impact-induced engine noise and vibration, Suh [7] found it important to include both timing and spectral information in the vibroacoustic data. This combined information is especially important when analysing sound signals induced by impact mechanisms. Valvetrains generate impact sound, especially at valve closing. In his valvetrain noise study, Suh was able to separate valve seating noise at the time of valve closing, as well as valve opening noise during the valve opening period.

In their review of state-of-the-art VVA applications, Lou and Zhu [8] predicted that ICEs will remain dominant vehicle power sources for decades to come. They recognised the need to replace mechanical systems with advanced technologies in order to meet the demands for continuous improvement of engine efficiency with reduced emissions. Full VVA is one of the most promising technologies with potential to improve fuel efficiency. There is abundant worldwide research on the topic. Notably, Jiang et al. [9], proposed a novel cam-less VVA system, based on a brushless direct current motor. The system uses an encoder to obtain the absolute position of the valve, and therefore control the velocity through its position. The authors implied that better control of the valve seat velocity can be used to reduce noise and vibration. This thesis has not been explicitly tested in the discussed work. One can note that NVH benefits in cam-less VVA systems can come from the lack of vibrations caused by camshaft acceleration and cam/follower friction.

In light of the above review, reducing the valvetrain´s NVH footprint is a highly relevant research issue, and one where synchronised spectral-temporal analysis is a key enabler. To this end, cutting-edge VVA systems with accurate control of the valve movements offer potential NVH reduction without impairing engine performance and efficiency. The latter advantage has been raised in the literature but, to date, no quality research has supported that with public results.

The present work focuses on the above knowledge gap. The research entailed mounting a fully variable electro-hydraulic valve actuation (EHVA) system on a medium-speed Wärtsilä 4L20 marine diesel engine. Accelerometers and a microphone were used to perform vibroacoustic measurements of the valve opening and closing events. Different valve profiles were designed and the associated noise and vibration of each profile were measured. Measurements were performed in the crank angle (CA) domain to enable detailed analysis of vibroacoustic phenomena, together with other events related to operation of the valves.

2 METHODS

The NVH experiments were conducted in the VEBIC engine laboratory at the University of Vaasa. The engine is a research version of a 200 mm-bore, medium-speed Wärtsilä 4L20 diesel engine, typical of marine or electricity generation applications. Its fundamental design is explained in the Wärtsilä 20 Product Guide [10]. Specifically adapted for research, the test unit is a four-cylinder version featuring an advanced common-rail fuel injection system and a turbocharger (ABB TPS48E01) with adaptable calibration and control functionalities. The engine operates at a full load of 848 kW, yielding 800 kW on the generator. The publication by Hautala et al. [11] provides greater insights into the engine and the measuring system supplying data for model calibration and validation. For the sake of brevity, only the part of the experimental setup relating to NVH measurement is described here.

2.1. Electro-hydraulic valve actuation

The experiments were done with the EHVA unit mounted on top of one of the cylinders of the 4L20 engine. To replace the original camshaft-driven valvetrain, the pushrods were removed and a new cylinder head with the EHVA was mounted on the engine. In addition to intake and exhaust valves, the main components of the EHVA system are the control system, control valves to regulate oil flow, analogue signal converters and linear position sensors that measure valve position. Control of EHVA is via the Speedgoat system, a rapid prototyping platform which allows fast-tracking of the control algorithm from the Matlab-Simulink environment level [12].

During operation, Speedgoat sends a control signal to the Parker D3FP proportional directional valves which, by regulating the hydraulic oil flow, actuate the movement of the engine´s valves. Figure 1 shows the layout of the EHVA system components and the operating principle. To follow a predefined reference valve profile, Speedgoat needs real-time intake and exhaust valve positions from the linear sensors synchronised with the CA reference from the engine gear tooth sensor. The hydraulic oil pressure is generated with a separate electric motor-powered hydraulic unit set to a feed pressure of 230 bar. The hydraulic unit draws the oil from the engine's sump. After being used by the EHVA, the oil returns to the sump, thus completing the oil circuit. Note that the main valve actuation force is created by a combination of hydraulic pressure/spring tension, as depicted in Figure 1, with the D3FP electro-hydraulic valves acting as low-power circuits. This cascade power transmission is more complex than state-of-the-art, purely electric-motor driven automotive VVA systems, as described in the introduction [9]. The complexity is needed to facilitate robust operation at ultra-high boost pressures used in lean-burn marine engines.

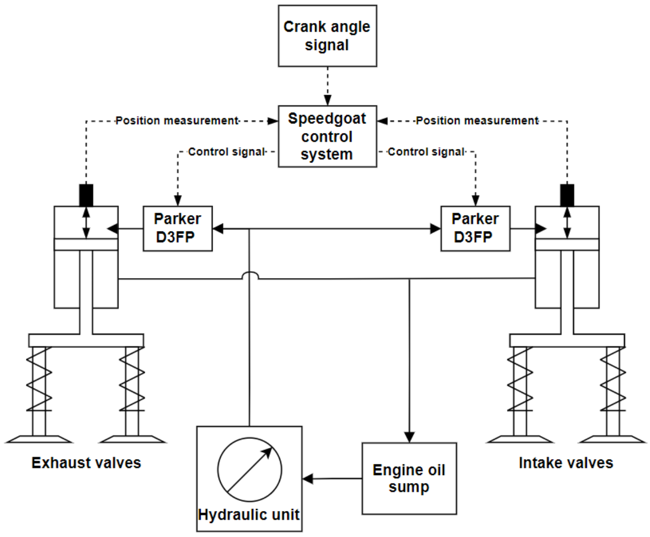

Figure 1. Operating principle of the electro-hydraulic valvetrain (EHVA) used in the experiment

2.2 NVH measurement setup and data acquisition system

Speedgoat requires both the valve position and CA references in order to control the valves. It uses these measurements for continuous calculation of the clearance between the gas exchange valves and the piston, activating immediate system shutdown if a valve comes too close to the piston. Positek P846 linear position sensors provide the positions of intake and exhaust valves, used as both the control signals and as reference quantities. The engine was not running during the test, thus making it possible to study noise and vibrations resulting explicitly from operating the gas exchange valves. Therefore, to actuate the valves, the engine-speed sensor was replaced with a signal generator to represent a signal from a conventional 120-1 gear tooth sensor. The data acquisition system used in the research is based on Dewesoft Sirius STG+ modules. Valve position and gear tooth simulator signal were sampled in parallel by both the Speedgoat-based control system and the Dewesoft system. Figure 2 depicts the experimental setup of the NVH measurements campaign.

The valve position and the CA encoder signal, together with signals from the accelerometers and microphone, were fed into the same Dewesoft measurement system. Kistler 8766A triaxial accelerometers, each covering three axes, were mounted on the engine using a magnetic base. They measured vibrations resulting from the impact of the intake and exhaust valves movement. A Gras 46AE ½-inch free-field microphone measured the resulting sound pressure levels. It was hung directly above the EHVA system, approximately one meter above the cylinder head, with the microphone axis directed in a perpendicular direction to the centreline of the engine, as shown in Figure 2.

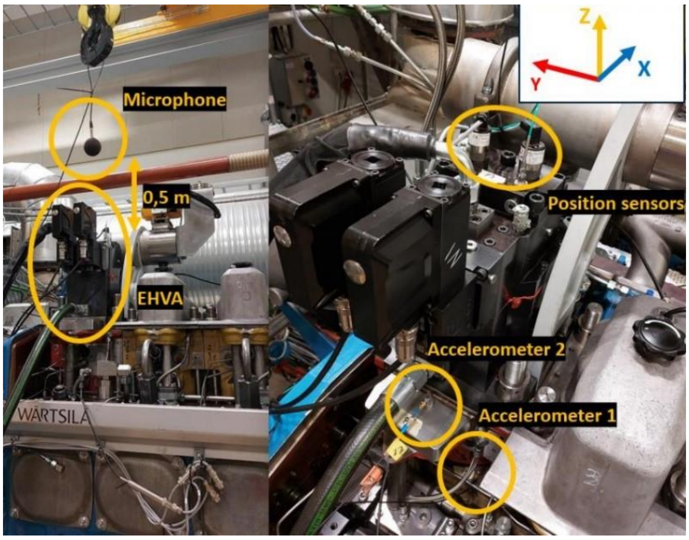

Figure 2. NVH measurement setup

2.3. Scope of the research

The research employs state-of-the-art NVH measurements to investigate the design of valve lift profiles and analyse noise and vibrations in electro-hydraulic engine valvetrains. The study´s reference valve lift profile, S0, aims to reproduce the 4L20 engine´s mechanical valve lift system. Using that as a reference baseline, three other different valve lift profiles were created by adjustment of the brake parameter in the EHVA control system. Each variant had large-scale differences in valve profile shapes, facilitating identification of vibroacoustic phenomena. The design of experiments also varied the speed with which valves would strike the seat. Measurements were made while emulating an engine speed of 1,000 rpm, representing the 4L20´s nominal operating condition.

3 RESULTS

The study´s aim is to obtain a fundamental understanding of excitations responsible for the excessive noise levels recorded during EHVA operations. This section provides analysis of the temporal and frequency aspects of the NVH measurements, together with an overall view of the results.

3.1. Temporal analysis

Figures 3–6 present the NVH results synchronised with CA-based valve lift signals for the cases S0 and S3. These are the profiles with the highest and the lowest measured overall sound pressure levels respectively. The analysis of temporal results indicates that valve closing is the origin of the main noise source. In CA domain, the valve closures correspond well to peaks in the signals for vibration and sound pressure. The delay between the peak of vibration and intake valve closure (IVC) is within 0.5 crank angle degree (CAD) for valve profile S0, as shown in Figure 3.

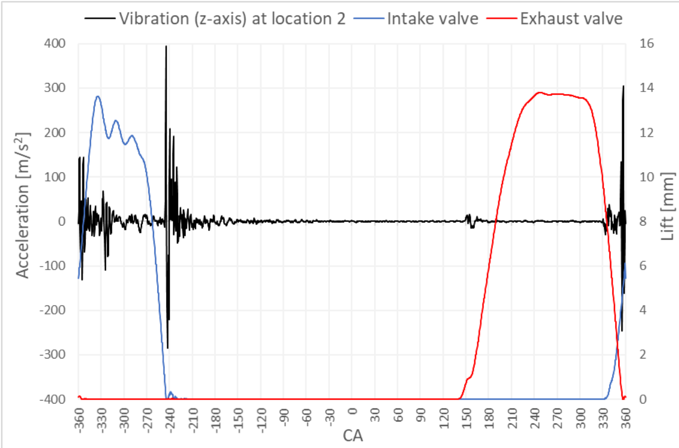

Figure 3. Valve profile S0, with CA-synchronised vibration data

The peak of the raw microphone signal has a short delay compared with the vibration signal peak. This is a consequence of the microphone being located 1 m away from the vibration sensor. The LAF_p signal is an A-weighted, post-processed product of the raw microphone signal by fast time weighting. The LAF_p signal responds more slowly to changes than the vibration signal and the raw microphone signal because it is created with a different signal processing technology. The sound pressure level signal LAF_p starts to rise almost immediately after both IVC and exhaust valve closure (EVC), as shown in Figures 4 and 6. The LAF_p signal peaks 89 CA after IVC for valve profile S0, as shown in Figure 4.

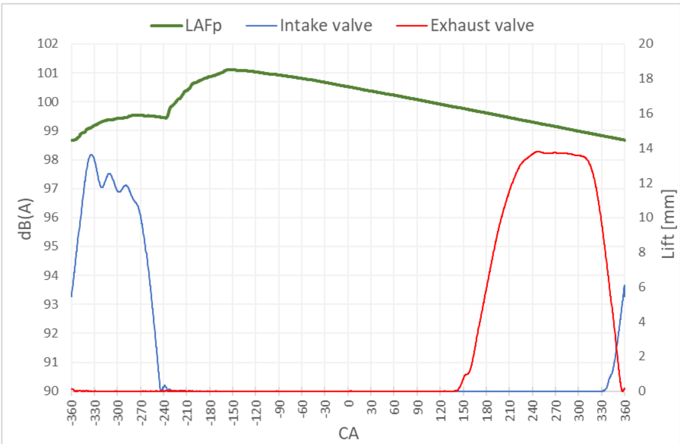

Figure 4. Valve profile S0, with CA-synchronised sound pressure level data

The noise and vibration amplitudes are determined by the speed with which the valve hits the seat. Due to a much smoother valve landing in profile S3 (Figure 6), the A-weighted sound pressure levels are significantly lower than those of valve profile S0 (Figure 4). The peak values of the LAF_p signal for valve profiles S0 and S3 are 101 dB and 93 dB respectively. Given the logarithmic scale of sound pressure level, the reduction in noise level is 84 %. The results for case S0, depicted in Figures 3 and 4, show that the closing of the intake valve is so severe that the impact generates significant valve backlash.

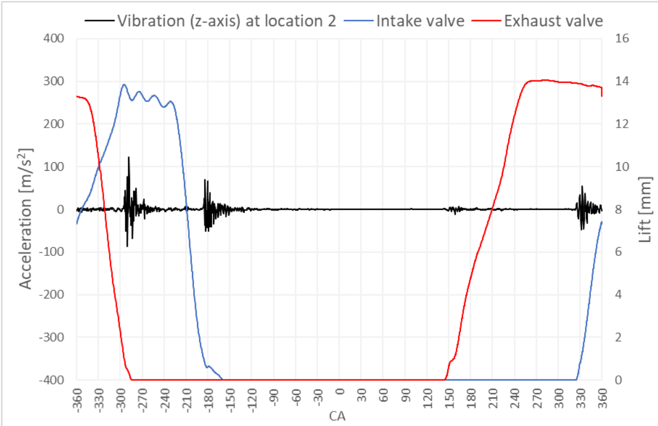

Figure 5. Valve profile S3, with CA-synchronised vibration data

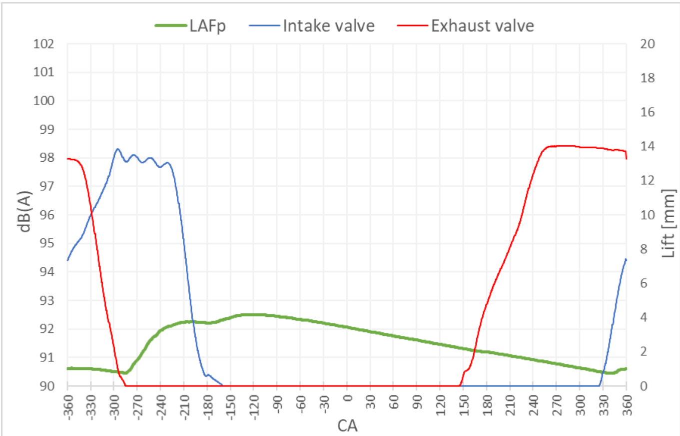

Figure 6. Valve profile S3, with CA-synchronised sound pressure level data

The vibration signals from accelerometer position 2 (z-axis) are displayed, together with valve profiles, in Figures 3 and 5. The accelerometer was located on the engine block near the cylinder head (Figure 2 shows accelerometer locations and axes). The vibration measurement results clearly show that the impact for the profile S3 is significantly lower than profile S0´s impact. The acceleration peak values recorded by the accelerometer at IVC for valve profiles S0 and S3 are 394 m/s^2 and 69 m/s^2, respectively. This represents an 82 % reduction in peak acceleration. Note that the discussion is narrowed down to profiles S0 and S3 because they have the most distinct difference. The results for the two other profiles are qualitatively the same and the trends correspond to the ones observed in Figures 3–6.

3.2. Frequency analysis

Measurement results indicate that overall vibration levels are significantly higher in the Z direction than in the X and Y directions, for all profiles. Therefore, only Z-direction measurements are analysed for each profile. Figure 7 presents a narrowband frequency analysis of the vibration measurements for profiles S0 and S3 at accelerometer positions 1 and 2 in the Z direction. It shows that acceleration levels are higher at position 2 than at position 1, for both profiles. Profile S0 generates higher acceleration levels than S3. Profile S0 has in position 1 a similar pattern but lower levels of acceleration compared with position 2. Profile S3 has in position 1 a flatter frequency response and lower acceleration levels compared with position 2. For valve profile S0 in position 2, two dominant ranges can be distinguished: the wider frequency range ranging from 1 to 2 kHz, and a narrower range at 2.5 kHz. For valve profile S3 in position 2, a narrow dominant range can be distinguished at 1.7 kHz.

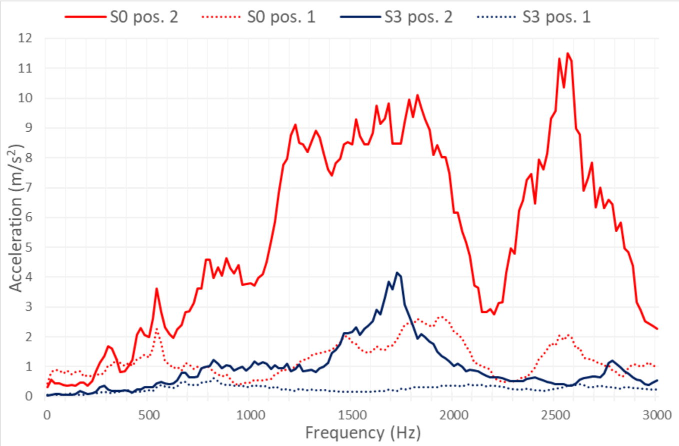

Figure 7. Overall vibration level at accelerometer positions 1 and 2, for tested valve profiles S0 and S3

Figure 8 depicts a 1/3-octave band analysis comparing all four valve lift profiles. It shows that the reduction in sound pressure levels from profile S0 to profile S3 is due to a significant reduction of amplitude levels for dominant frequency bands. Profile S3´s dominant frequency band is shifted to slightly higher frequencies compared with other profiles. The most dominant 1/3-octave frequency band is 2500 Hz for valve lift profiles S0, S1 and S2, whereas it is at 8000 Hz for profile S3.

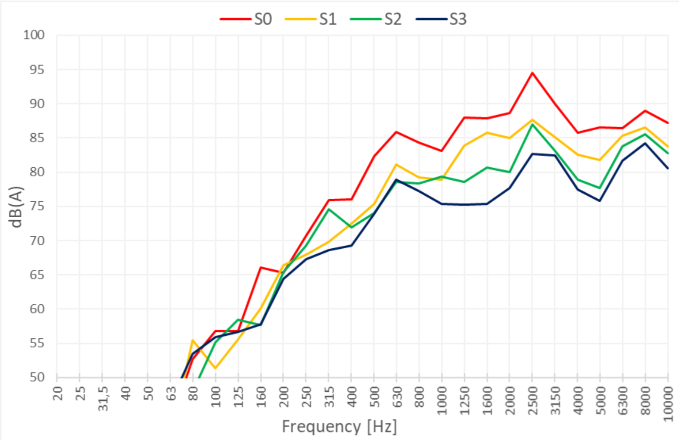

Figure 8. Sound pressure level analysis in 1/3-octave band

3.3. Discussion and outlook

The overall results indicate that profile S0 generates the highest sound pressure levels, and that valve lift profile S3 generates the lowest. The reduction in the A-weighted equivalent sound pressure levels (LA_{eq}) from valve lift profile S0 to profile S3 is 8 dB, as indicated in Figure 9. This is a huge margin over the currently used valve profile. Given the logarithmic scale of sound pressure level, the reduction in noise level is 84 %.

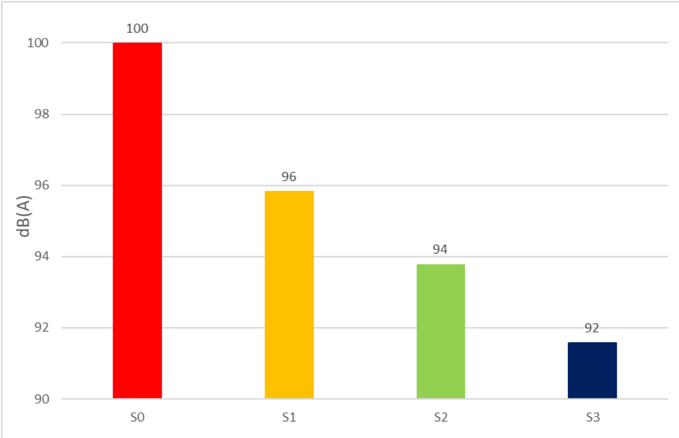

Figure 9. Overall noise level (LA_{eq}) comparison of different valve profiles

In the present study, the whole intake/exhaust valve profile timings were varied to allow separation/identification of the NVH phenomena related to EHVA. Still, the main takeaway is that, irrespective of the valve timing, a significant reduction in sound pressure levels can be achieved by designing smoother landing of the valves against the engine seats. EHVA, in principle, allows adjustment of the landing phase, without significantly affecting the other parameters of the profile. This gives potential to reduce the NVH footprint of the valvetrain, without affecting the volumetric efficiency of the engine or pumping losses. A more thorough study, to confirm this thesis, with the help of one-dimensional engine performance simulations, is planned for the next research phase of the Silent Engine Project [13].

4 CONCLUSIONS

- Analysis of NVH measurement results indicates that valve closing is the origin of the main noise source during EHVA operation.

- The results show that a significant reduction of the NVH footprint was achieved by improving the design of the valve lift profile.

- The differences between valve lift profiles are demonstrated by reduced levels of sound pressure and vibration.

- An overall difference in sound pressure levels of 8 dB was measured between the original valve lift profile and the profile with the smoothest valve landing.

- The reduction of vibration accelerations and sound pressure levels is due to a significant decrease in the amplitude levels of dominant frequencies.

- Vibration measurement results show an 82 % reduction in acceleration peak values at IVC between the original valve lift profile and the profile with the smoothest valve landing.

- The valve landing can be optimised without affecting the overall valve timing, providing a pathway for noise reduction without a negative trade-off.

ACKNOWLEDGEMENT

The results were achieved in the "Partnership model - Silent Engine" project that was co-funded by Business Finland (Next Generation EU funding).

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NOMENCLATURE

CA	crank angle
CAD	crank angle degrees
EHVA	electro-hydraulic valve actuation
EVC	exhaust valve closure
ICE	internal combustion engine
IVC	intake valve closure
NVH	noise, vibration and harshness
VVA	variable valve actuation