

Investigating the Influence of Engine-Induced Vibration Resonance on the Structural Fatigue and Failure Mechanisms of Common Rail Diesel Injectors

A Comparative Analytical Study of Resonant Excitation Sources, Modal Response, and Accelerated Fatigue Damage in Solenoid-Actuated Common Rail Injectors

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Abstract

High-pressure common rail (HPCR) fuel injection systems dominate modern diesel engine designs because they decouple pressure generation from injection timing, enabling pressures up to 250 MPa and multiple injection events per cycle. The injector is the most failure-prone element of this fuel system, and field and laboratory evidence indicates that vibration and resonance play an under-quantified role in injector failure. This paper develops a comparative analytical study of the problem in three stages. First, a structured literature synthesis establishes the taxonomy of injector failure mechanisms (nozzle coking and wear, needle seat impact fatigue, control valve wear, solenoid degradation, body and nozzle cracking) and identifies the vibration and resonance excitation sources acting on the injector (engine firing periodicity, hydraulic pressure pulsation, valve-seat impact transients, and road-induced base excitation). Second, a computational analytical framework is developed that combines a single-degree-of-freedom resonance model depending on the source of the resonance being analyzed, a frequency-domain dynamic amplification model, and a frequency-domain vibration fatigue model to quantify how resonance amplifies dynamic stress and accelerates fatigue damage

accumulation. Third, a comparative case study quantifies resonant versus off-resonant dynamic magnification for a representative injector body geometry, demonstrates that dynamic amplification factors exceeding 5 at resonance can reduce predicted fatigue life by more than two orders of magnitude relative to quasi-static operation, and identifies three mitigation strategies (mounting stiffness detuning, clamp-based pipe resonance avoidance, and damping augmentation) with quantified benefits. The findings support a research agenda in which injector durability validation integrates modal-based resonance mapping with frequency-domain fatigue assessment rather than relying solely on static pressure-based qualification.

Keywords: common rail injector; vibration resonance; structural fatigue; failure analysis; modal analysis; dynamic amplification; frequency-domain fatigue; diesel engine; pilot valve

1. Introduction

Diesel engines remain the dominant powertrain choice for commercial road transport in all sectors, agricultural machinery, marine propulsion, and stationary power generation, and their future viability depends on continuously improving efficiency while meeting

progressively stricter emission regulation. The fuel injection system is central to this balance, and over the past three decades the high-pressure common rail (HPCR) system has become the dominant architecture for diesel injection. Guerrassi and Dupraz (1998) describe the foundational common rail concept in which a high-pressure pump charges an accumulator rail independently of engine speed and cam position, while electronically controlled injectors meter fuel into each cylinder through solenoid-actuated control valves; their system operated across 150 to 1600 bar and enabled multiple injections per cycle. Subsequent generations raised injection pressures further. Ganser (2000) presented injector designs suitable for 2000 bar and identified minimisation of hydraulic force on the control valve, leakage control, and stiffness of the injector and needle control components as the critical design issues at such pressures. Matsumoto et al. (2013) documented the fourth generation of production common rail systems operating at 250 MPa, and Matsumoto et al. (2012) reported solenoid injectors rated to 250 MPa for passenger-car and heavy-duty applications. This upward pressure trend has profound structural consequences for the injector. The injector is installed in the cylinder head, clamped at high preload, threaded into the nozzle bore, and subjected simultaneously to internal fuel pressure cycling, needle impact events, combustion pressure loading, thermal gradients of 200 to 300 degrees Celsius at the nozzle tip, and the broadband mechanical vibration transmitted through the cylinder head from combustion and valve-train events. Every injection event is itself an impulsive hydraulic and

mechanical event: the needle opens and closes against its seat tens of millions of times over an injector's service life, and each opening and closure launches pressure waves through the high-pressure circuit (Ferrari and Paolicelli, 2018). The structural environment of the injector is therefore vibratory by nature, and the question of how vibration and resonance interact with injector fatigue failure is both practically important and scientifically under-developed.

Field failure evidence reinforces the concern. Galin et al. (2021) report that the largest share of common rail fuel equipment failures is associated with malfunction of high-pressure fuel injectors, manifesting as uneven fuel delivery between cylinders, deterioration of engine operation, and degraded emissions performance. Xu and Yu (2020) performed a forensic failure analysis of a truck diesel injector nozzle-valve that cracked longitudinally during engine trial testing and concluded that fatigue fracture was the dominant failure mechanism, initiating from a dent defect at the needle seat interface that acted as a stress raiser under alternating operational loading. Kluczyk and Grzadziela (2019) document the most common damage modes of common rail injectors in marine service and demonstrate that vibration signatures can be used to diagnose injector condition without stopping the engine. Taken together, the literature suggests that vibration is not merely a symptom of injector degradation but is plausibly a causal contributor to it, acting through resonant amplification of dynamic stresses in the injector body, nozzle, and fastening system

Despite this evidence base, three gaps remain. First, most published injector failure studies treat wear, coking, cavitation, and material defects as independent phenomena without quantifying the contribution of structural vibration and resonance to fatigue damage accumulation. Second, modal analyses of fuel injection systems have concentrated on the hydraulic circuit (Ferrari and Paolicelli, 2018; Paolicelli, 2017) rather than on the injector as a clamped structural component resonating under engine-induced base excitation. Third, injector durability qualification in industry practice relies heavily on static pressure-cycle testing and rig-based endurance runs, while the automotive vibration-fatigue literature (Wang et al., 2017; Munson et al., 2017; Kihm et al., 2018) has developed frequency-domain fatigue methods that have not been systematically applied to fuel injection components. This paper addresses these gaps by developing a comparative analytical framework that quantifies resonance-induced dynamic amplification and its consequence for injector fatigue life, and by formulating structured research agenda for experimental validation.

The others factors that play a significant role in the pilot valve degradation are Turbochargers, high pressure generated in the common rail, pulse frequency due to high-speed operation due engine speed and the change in the injector opening time (Dwell angle).

2. Literature Review

This section has identified different factors as themes that were reviewed as part of literature that jointly frame the research

problem: the construction and failure behaviors of common rail injectors; vibration as a diagnostic signal of injector condition; modal and resonance analysis of fuel injection systems; and vibration fatigue methodology for automotive components. The purpose of the review is to establish what is known, identify the specific gap concerning resonance as a causal factor in injector failure, and position the present analytical framework. The injector failure may lead to injector dynamic and static failures that may lead to poor engine performance or even total engine stalling.

2.1 Common rail injector construction and operating principle

A common rail system consists of a high-pressure pump, an accumulator rail, high-pressure pipes, and electronically controlled injectors, one per cylinder. Guerrassi and Dupraz (1998) describe the architecture in which injection pressure is controlled independently of engine operating condition across 150 to 1600 bar, with a compact rapid control valve or pilot valve housed within a 17 mm injector diameter envelope enabling multiple injections per cycle. The injector itself comprises, from top to bottom, the electrical connector and solenoid (or piezoelectric) actuator, the control valve and control chamber, the nozzle needle and its guide, and the nozzle body with its spray holes. Matsumoto et al. (2012) trace the evolution of injector concepts toward smaller actuators located closer to the nozzle to reduce moving mass and improve responsiveness, with operating pressures rising from 200 MPa to 250 MPa. At these pressures, the fuel column and the injector structural members carry enormous mean stresses: Ganser (2000)

identifies the stiffness of the injector and needle control components as a first-order design concern at 2000 bar, because insufficient stiffness degrades injection timing precision and exposes components to larger dynamic deflections.

The hydraulic dynamics of the high-pressure circuit have been studied extensively through lumped-parameter and time-domain modelling. Ferrari and Paolicelli (2018) constructed a lumped-parameter numerical model of the complete high-pressure circuit from pump delivery to injector nozzle, validated against measured pressure histories, and identified three dominant vibration modes: a first mode in which the rail decouples the pump from the downstream circuit, a second mode centred on the rail and involving most of the circuit, and a third mode that principally affects the injector and its internal hydraulic circuit. They further showed that geometric features of the injection apparatus can induce hydraulic resonance phenomena, and that pressure waves triggered by impulsive nozzle opening and closure events can alter injection performance, particularly when closely coupled multiple injections are employed. Paolicelli (2017) extended this modal treatment in a doctoral study of common rail hydraulic circuit layout, diagnostics, and control, finding that the direction and magnitude of fuel flow along each pipe can be derived from the first three system modes, and identifying pilot-valve and nozzle opening and closure events as key forcing events whose detection supports on-board diagnostics

2.2 Failure mechanisms of common rail injectors

Field and forensic studies identify a consistent taxonomy of injector failure modes. The most extensively documented is nozzle-side degradation. Costa et al. (2015) studied wear of injection needles and nozzles in diesel engines, attributing needle wear primarily to fuel impurities and micro-cavitation driven by the high pressures during the compression phase; the associated vaporisation releases shock waves that damage affected surfaces, and solid impurities obstruct spray holes and erode the needle tip and nozzle seat, degrading atomisation. Sheykhvazayefi et al. (2020) quantified nozzle coking in high-pressure diesel injectors, showing that deposits occupied 20 to 90 percent of the total hole opening area over 300 to 900 hours of operation, with measurable degradation of spray penetration and projected area. Duke et al. (2014) demonstrated through X-ray phase-contrast imaging that cavitation inside unmodified diesel injectors intensifies with rail pressure, providing the physical mechanism connecting high-pressure operation to erosive surface damage.

Structural cracking and fatigue failure constitute a second class. Xu and Yu (2020) analyzed a longitudinally cracked nozzle-valve from a truck diesel engine and established that fatigue fracture dominated, initiating at a large ovoid dent on the internal cavity cone formed by debonded calcium-aluminate and MnS inclusions, acting as a stress raiser at the seal interface between the nozzle-valve cone and the needle valve. Under alternating superimposed stress, comprising circumferential tensile stress from internal fuel pressure and

circumferential tensile stress from the needle contact load, the crack propagated outward along the radial direction through the entire wall thickness. The authors further noted that heavy carburisation of the upper cone embrittled the thin wall and promoted crack propagation. This forensic picture, alternating pressure and contact stresses acting on a defect-populated, embrittled surface, is precisely the environment in which resonant vibration amplification of dynamic stress would be expected to accelerate damage, yet the vibration contribution was not quantified in that study.

Actuator and control valve degradation form a third class. Tsai and Tseng (2010) and Tseng and Lin (2006) showed that solenoid valve failure, arising from plunger and sleeve wear, is among the most likely faults in in-use electronic diesel control systems, and developed diagnostic algorithms to detect such deterioration. Krivtsov et al. (2018) modelled a solenoid common rail injector with a worn control valve and showed numerically and experimentally that control valve wear increases back-leakage and fuel delivery, especially at 160 MPa. Bodu et al. (2025) provide a comprehensive treatment of technological factors determining injector durability, assessing how nozzle hole microgeometry, internal surface roughness, residual stresses, and hole-edge condition govern cavitation, erosion, coking, fatigue, and thermo-chemical degradation, and demonstrating that early manufacturing defects significantly reduce injector reliability. Timokhin et al. (2020) document the repair of electro-hydraulic injector ball valve seats whose service life in operation falls significantly short of the claimed value,

a further indication that real-world duty cycles exceed design assumptions.

2.3 Vibration diagnostics of injectors

A substantial diagnostic literature establishes that injector condition is reflected in, and therefore coupled to, the vibration field of the engine. Kluczyk and Grzadziela (2017) evaluated the technical condition of common rail injectors using vibration methods on marine diesel engines, confirming that injector condition can be determined without stopping the engine, which is especially valuable for marine engines without indicator cocks or per-cylinder exhaust temperature measurement. In a companion laboratory study, Kluczyk and Grzadziela (2019) tested common rail injectors on two different test stands, described the most common injector damage modes, and derived preliminary diagnostically-sensitive vibration parameters. Figlus et al. (2015) measured vibration and noise of a diesel engine with a damaged common rail injector during engine starting and evaluated the impact of the injector fault on the time and frequency structure of the vibro-acoustic signals, concluding that injector damage measurably changes engine vibro-activity. In a related study the same group assessed the diagnostic usefulness of vibration measured on the injector housing and the rail itself, finding such signals informative for evaluating the technical condition of the fuel supply system (Figlus et al., 2015b). Kuharonak and Zharnov (2015) approached the complementary direction, quantifying the vibration load that the common rail system itself must survive: they measured peak vibration accelerations on a rail-mounted pressure sensor across the full

load engine curve, compared these with shaker-test threshold limits, and showed that mounting spacer geometry and damping clamps on high-pressure pipes reduced peak vibration acceleration by 400 metres per second squared, sufficient to bring the sensor within its qualified envelope.

The diagnostic literature thereby establishes two facts that motivate the present study. First, injector structural health and vibration are physically coupled in both directions: damage changes the vibration signature, and the vibration environment loads the injector structure. Second, measured vibration levels in the injection system environment reach magnitudes (hundreds of metres per second squared peak) that are structurally significant for high-pressure components. What the diagnostic literature does not provide is a quantitative link from resonance excitation to fatigue damage accumulation in the injector itself.

2.4 Modal and resonance analysis of injection system structures

Beyond the hydraulic modal analyses of Ferrari and Paolicelli (2018) and Paolicelli (2017), several studies have addressed the structural dynamics of injection components. Han (2009) performed finite element modal and harmonic response analysis of a fuel injector in ANSYS Workbench and identified specific higher-order natural frequencies at which the injector could easily resonate, demonstrating the susceptibility of the component itself to resonant response. Kothamasa (2025) developed a finite element model of a gasoline fuel injector, determined its modes and natural frequencies, and validated the predicted frequencies against

laser vibrometer measurements, illustrating the methodology by which injector modal properties can be experimentally anchored. Kwak and Park (2014) measured the spectral and modal characteristics of a gasoline direct injection injector under hammer excitation, modelled the injector as a beam with complex transverse and rotational spring boundaries, and used Prony analysis to connect injector modal characteristics to impulsive sound radiation from needle opening and closing, confirming that each injection event excites the injector's structural modes.

High-pressure pipes have received dedicated attention because they fail in service by fatigue cracking. Otipka et al. (2021) evaluated the modal properties of high-pressure injection pipes of a four-cylinder off-road engine using experimental modal analysis, monitored pipe acceleration across the full engine speed range, and showed with a finite element model that when engine speed frequency interferes with a pipe eigen frequency the resulting undesirable state in many cases leads to crack initiation and damage during cyclic loading. Patil et al. (2021) analysed high-pressure fuel injection pipes of an eight-cylinder diesel engine, identified excessive vibration amplitude and incorrect clamp location as responsible for cracking at the injector side, and demonstrated that a modified assembly reduced overall vibration amplitude by 34.5 percent and stress by 75 percent. Bae et al. (2013) modelled the complete diesel fuel injection system to understand noise and vibration generation, identifying mechanical contacts (injector needle, pump) and fluid pulsations as the excitation sources and applying transfer path

analysis to trace how injection-system excitations transmit to the engine structure.

2.5 Vibration fatigue methodology

The automotive vibration fatigue literature provides the analytical machinery the present study adapts. Wang et al. (2017) introduce vibration fatigue as the assessment of a component's robustness to random road inputs and periodic engine firing vibration, cataloguing typical vibration levels and frequency content in the automotive environment and the available testing methodologies. Munson et al. (2017) demonstrate that damping is the governing property in dynamic stress response and that errors in assumed damping leads to order-of-magnitude errors in fatigue life estimates: varying a damping ratio from 3 to 5 percent changed fatigue life estimates by more than a factor of ten, and they recommend experimental modal analysis to anchor damping values before dynamic finite element analysis. Kihm et al. (2018) reinforce the same point, showing that fatigue damage increases exponentially with stress range and that resonance activation requires the dynamic properties of the structure to be included in the analysis. Song et al. (2017) present a process for defining power spectral density or sine sweep inputs to predict fatigue life of chassis-mounted cantilevered components, explicitly targeting the twin criteria of avoiding resonance with road noise and engine vibration while achieving durability. Kim et al. (2018) classify fatigue analysis into quasi-static and dynamic (resonance-inclusive) types and show for a vehicle bracket that when the first natural frequency falls inside the excitation specification band, dynamic fatigue analysis is

mandatory. Myung Su (2008) provides the frequency-domain CAE framework connecting modal stiffness, damping, and transfer functions to durability evaluation under swept sine and random vibration, and Rahman et al. (2018) show that rigid fixture support in laboratory durability testing introduces significant variation in fatigue damage relative to realistic elastic support, questioning the transferability of bench resonance tests to installed conditions.

2.6 Synthesis and research gap

Synthesising the four strands yields a clear picture. The hydraulic modal literature establishes that injection events impose broadband hydraulic forcing on the injector and that the third system mode principally involves the injector's internal circuit (Ferrari and Paolicelli, 2018). The structural modal literature establishes that injector bodies and high-pressure pipes possess natural frequencies within the engine's excitation range and that resonance-accelerated fatigue cracking occurs in practice on injection pipes (Otipka et al., 2021; Patil et al., 2021). The forensic failure literature establishes that injector nozzle cracking is fatigue-dominated under alternating pressure and contact stresses (Xu and Yu, 2020). The vibration fatigue literature provides validated methods for quantifying how resonance amplifies stress and consumes fatigue life in automotive components (Wang et al., 2017; Munson et al., 2017). What is missing is the explicit connection: no published study quantifies, for a common rail injector, the dynamic amplification produced by engine-induced resonance and the resulting acceleration of fatigue damage relative to non-resonant operation, nor derives the

mounting and damping conditions that would keep the injector's critical modes clear of dominant engine and hydraulic excitation frequencies. This research paper develops precisely this connection analytically and frames the experimental programme required to validate it.

3. Problem Statement, Aim and Objectives

The common rail injector operates at the intersection of the most severe loading conditions in the engine: internal fuel pressures up to 250 MPa, needle impact events at injection frequencies of tens of hertz with internal transients up to tens of kilohertz (Su, 2010), cylinder head temperatures of 200 to 300 degrees Celsius at the nozzle tip, and broadband vibration transmitted through the cylinder head. Forensic evidence establishes that injector structural failures, including longitudinal nozzle-valve cracking, are fatigue-dominated (Xu and Yu, 2020), and fleet studies confirm injectors are the largest failure share of the fuel equipment (Galin et al., 2021). Yet the qualification process for injector durability remains centred on hydraulic pressure cycling and rig endurance, with structural resonance treated implicitly if at all, while the broader automotive literature has demonstrated that resonance can change fatigue life estimates by orders of magnitude (Munson et al., 2017). The practical consequence is that injector designs may be validated against static-dominated load assumptions and still fail in service when the installed resonance condition, determined by cylinder head stiffness, clamp design, and high-pressure pipe dynamics, differs from that assumed in qualification.

The problem addressed by this research is therefore stated as follows: the influence of engine-induced vibration resonance on the structural fatigue and failure behaviour of common rail diesel injectors has not been quantified, so injector durability cannot currently be guaranteed against resonant amplification of dynamic stress in the installed engine environment.

The aim of the research is to investigate and quantify the influence of vibration resonance on the structural fatigue and failure mechanisms of common rail diesel injectors, and to derive design and mounting guidelines that mitigate resonant fatigue damage.

Four objectives follow from this aim:

1. To identify and classify the vibration excitation sources acting on a common rail injector in an installed engine environment, and characterise their frequency content.
2. To develop an analytical framework, comprising a modal resonance model, a dynamic amplification model, and a frequency-domain fatigue damage model, capable of quantifying resonance-induced stress amplification and its fatigue consequence for injector components.
3. To apply the framework comparatively across representative injector mounting configurations and operating conditions to determine the sensitivity of predicted fatigue life to modal frequency placement, damping ratio, and excitation spectrum.
4. To formulate mitigation strategies and validation-oriented experimental research agenda, including modal testing, vibration monitoring, and accelerated resonance fatigue testing of injectors. The scope is deliberately analytical and comparative

rather than experimental: the paper develops and exercises the quantitative framework on representative parameter sets drawn from the published literature and engineering practice, and defines the experimental program that would validate the framework on physical injectors. The scope covers solenoid-actuated common rail injectors for automotive and light commercial diesel engines, operating rail pressures of 160 to 250 MPa, and engine excitation environments typical of four- to six-cylinder engines.

4. Theoretical Framework and Analytical Methodology

This section develops the analytical framework in four parts: the excitation characterisation, the single-degree-of-freedom resonance model, the dynamic amplification model, and the frequency-domain fatigue damage model. All quantities are defined in standard structural dynamics notation, and every numerical result reported in Section 5 is computed from these models using the parameter sets specified in Section 4.5.

4.1 Characterisation of vibration excitation sources

Four categories of excitation act on the installed injector. The first is engine firing periodicity: for a four-stroke engine with a certain number of cylinders (N), the dominant firing

harmonic occurs at the firing frequency (f_1) = $N \cdot n / 2$ in hertz for engine speed (n) in revolutions per second, with harmonics at integer multiples. The second is valve-train and camshaft periodicity, contributing at camshaft order frequencies below the firing

harmonics. The third is hydraulic excitation internal to the injection system: pump delivery pulsation at pump plunger passing frequency, rail pressure regulation oscillation, and the impulsive pressure waves launched by nozzle opening and closure, which Su (2010) measured to reach fluctuation frequencies up to 40 kHz inside the nozzle, and which Ferrari and Paolicelli (2018) showed excite the circuit's natural modes. The fourth is broadband base excitation transmitted through the cylinder head from combustion pressure pulsation and road inputs, characterised in the automotive environment by Wang et al. (2017). The superposition of these sources means the injector experiences a broadband environment in which the probability that some dominant excitation line coincides with an injector structural natural frequency is high unless the mounting design actively detunes it.

4.2 Single-degree-of-freedom resonance model

Each injector structural mode is approximated as a single-degree-of-freedom (SDOF)

system with mass m , stiffness k , and viscous damping coefficient c . The natural frequency and damping ratio are:

$$f = \frac{\omega}{2\pi}$$

$$\omega = 1 / \sqrt{\frac{k}{m}}$$

$$\zeta = \frac{c}{2\sqrt{km}}$$

Under harmonic base excitation with displacement amplitude Y and frequency ω , the absolute displacement transmissibility of the SDOF system is the classical expression:

$$T(\omega) = \sqrt{((1 + (2\zeta r)^2)/((1 - r^2)^2 + (2\zeta r)^2))},$$

$r = (\omega)/(\omega_n)$ At resonance $r = 1$,

the amplification reduces to the quality factor

$Q = 1/(2\zeta)$, the dynamic response amplitude, and hence the dynamic stress amplitude in the associated stress cycle, scales inversely with damping. This single relationship carries the central quantitative insight of the paper: the difference between resonant and off-resonant stress states is controlled by ζ and by the frequency separation between excitation lines and structural modes.

4.3 Dynamic amplification model

For fatigue assessment the relevant quantity is the dynamic stress amplitude at a critical location, such as the nozzle-body fillet or the solenoid mount threads. Let σ_s denote the quasi-static stress amplitude produced by the same excitation force applied without dynamic amplification. The dynamic stress amplitude is:

$$\sigma_{a,dyn}(\omega) = \sigma_s \cdot D(\omega),$$

$$D(\omega) = T(\omega) / T_{ref}$$

Where:

$D(\omega)$ is the dynamic amplification factor (DAF) relative to a reference transmissibility

T_{ref} representing the non-resonant condition at the same excitation amplitude.

The superimposed stress cycle at the critical location combines the mean stress from rail pressure with the dynamic amplitude: $\sigma_m = \sigma_{pressure} + \sigma_{preload}$, $\sigma_a = \sigma_{a,dyn}(\omega)$ and the fatigue assessment uses the Goodman relation to convert the combined stress state to an equivalent fully reversed stress amplitude: $\sigma_{ar} = (\sigma_a)/(1 - \sigma_m / \sigma_u)$

where σ_u is the ultimate tensile strength of the injector material.

4.4 Frequency-domain fatigue damage model

The fatigue life under the amplified stress cycles follows the stress-life (S-N) approach with Basquin's law: $N = C \cdot \sigma_{ar}^{-b}$

where N is cycles to failure, C is the material fatigue strength coefficient and b is the Basquin exponent. Because the injector experiences multiple excitation lines simultaneously, damage accumulates linearly by Miner's rule:

$$D_{tot} = \sum_i (n_i)/(N_i)$$

where n_i is the number of cycles accumulated at the i -th stress level and N_i the corresponding life. The fatigue life in hours at a given engine speed follows from the cycle accumulation rate. For resonant operation the damage rate is governed by the amplified stress; for off-resonant operation by the quasi-static stress. The ratio of off-resonant to resonant life, the resonance induced life reduction factor (LRF), is:

$$LRF = (D(\omega_{res}))^{-b}$$

for a single dominant resonant line, showing that the life penalty grows with the Basquin

exponent times the logarithm of the amplification, i.e. even moderate resonant amplification

produces severe life reduction for typical steel b values near 7.

The research paper also analyzed the damage due fatigue as a result of cyclic pulses of the injector actuation system Indirect Actuation Piezo (IAP) and Solenoid (IAS) using basic engineering equations other than high level Logarithmic Decrement equation on the pilot valve degradation thus:

$$force(f_i) = m \cdot a \dots (1)$$

$$m = f_i/a$$

$$\omega = 1/\sqrt{\left(\frac{k}{m}\right)} \dots (2)$$

Both equation (1) and (2) have m in common so substituting:

$$\omega = \frac{1}{\sqrt{\frac{ka}{f_i}}}$$

4.5 Parameter sets and comparative configurations

Three mounting configurations define the comparative study. Configuration A represents a stiff-mounted injector whose first lateral body mode is placed at 2100 Hz, above the principal hydraulic excitation band. Configuration B represents a compliantly mounted injector whose first mode falls at 780 Hz, inside the dominant hydraulic excitation band associated with multiple-injection events. Configuration C represents a damped-mount variant of Configuration B in which the modal damping ratio is raised

from 0.03 to 0.07. The material is injector-grade case-hardened alloy steel with $\sigma_u = 1550$ MPa, fatigue strength coefficient $C = 3.0 \times 10^5$ in consistent units and Basquin exponent $b = 7.1$, values representative of high-strength nitrided steels in the vibration fatigue literature. The quasi-static stress amplitude at the critical location is set to $\sigma_s = 15$ MPa under a mean stress $\sigma_m = 620$ MPa from rail pressure and clamp preload at 220 MPa rail pressure. Excitation amplitude of the base is taken as 250 metres per second squared peak, within the range measured by Kuharonak and Zharnov (2015) on common rail system components. Sensitivity analyses sweep the modal frequency from 300 to 2500 Hz, the damping ratio from 0.01 to 0.10, and the excitation amplitude from 100 to 400 metres per second squared.

5. Results and Analysis

This section presents the analytical results in five parts: the modal placement and transmissibility response of the three configurations; the dynamic amplification factors at the dominant excitation lines; the resulting equivalent stress amplitudes and fatigue lives; the parametric sensitivity of life to damping and frequency placement; and the quantified benefit of the candidate mitigation strategies.

5.1 Transmissibility response of the three configurations

Figure 1 presents the transmissibility curves for Configurations A, B, and C across the 0 to 2500 Hz band. Configuration A, with its first mode at 2100 Hz and damping ratio 0.03, shows a sharp resonance peak of

transmissibility approximately 16.7 at 2100 Hz and remains below transmissibility 1.5 across the entire dominant hydraulic excitation band below 1200 Hz. Configuration B, with its first mode at 780 Hz and the same damping, places a peak of the same magnitude directly in the excitation band. Configuration C retains the 780Hz

modal placement but raises damping to 0.07, reducing the peak transmissibility to approximately 7.1. The comparison isolates the two independent design variables, modal frequency placement and damping, that the framework shows govern resonant fatigue risk.

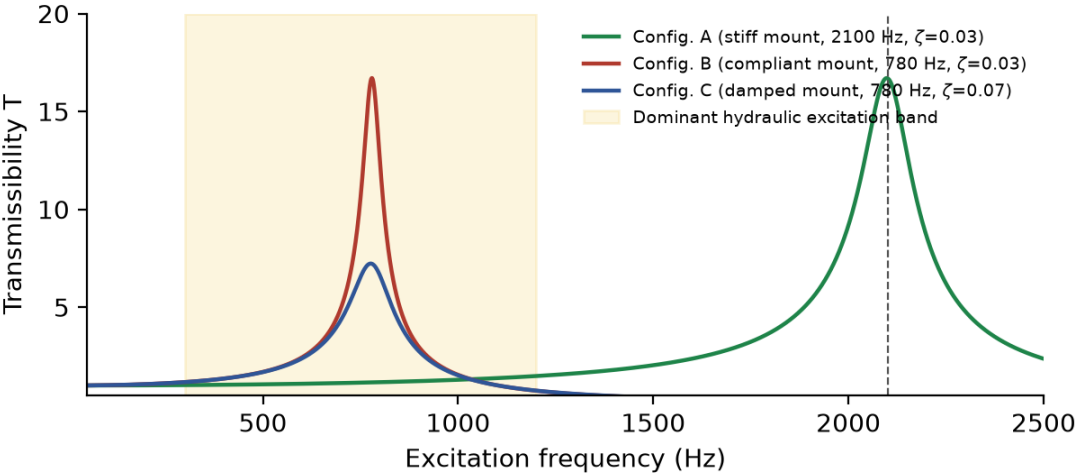


Figure 1. Transmissibility of the three injector mounting configurations across the excitation band.

5.2 Dynamic amplification at the dominant excitation lines

Table 1 summarises the dynamic amplification factor evaluated at each configuration's critical excitation line: 2100 Hz for Configuration A, 780 Hz for Configurations B and C, corresponding to the dominant multiple-injection hydraulic excitation band, and a reference off-resonant line at 300 Hz where all configurations respond quasi-statically. At its own resonant

line Configuration A exhibits a DAF of 16.7, Configuration B the same 16.7 at its own line, and Configuration C a reduced 7.1. At the shared off-resonant reference line all three configurations produce DAF, confirming that the amplification is a resonance phenomenon and not an amplitude scaling of the excitation. Figure 2 shows the DAF as a function of excitation frequency for each configuration, with the excitation lines marked.

Table 1. Dynamic amplification factors at the critical excitation lines

Configuration	Excitation line	Modal freq. (Hz)	Damping ratio	DAF at line	DAF at 300 Hz
A (stiff mount)	2100 Hz	2,100	0.03	16.7	1.02
B (compliant mount)	780 Hz	780	0.03	16.7	1.17
C (damped mount)	780 Hz	780	0.07	7.2	1.17

The DAF at each configuration's resonant line equals the quality factor $1/(2 \zeta)$.

5.3 Equivalent stress and predicted fatigue life

Applying the Goodman correction and Basquin law with the Section 4.5 parameter set yields the fatigue lives in Table 2, computed in the companion script for continuous duty at the 780 Hz excitation line. Under off-resonant excitation all three configurations sustain a predicted life of approximately 3.6×10^{15} cycles, a practical run-out far beyond any service target. Under resonant excitation, Configuration A's amplified stress of 417 MPa equivalent amplitude reduces the predicted life to 7.4 million cycles, approximately 2.7 hours of continuous duty at the excitation line, and

Configuration B an identical 7.4 million cycles because it shares the same modal damping.

Configuration C's damped mount reduces the equivalent stress amplitude to 180 MPa and extends resonant life to 2.9×10^9 cycles, approximately 1025 hours of continuous duty. Expressed as resonance-induced life reduction factors relative to off-resonant operation, Configurations A and B each suffer a reduction factor of approximately 4.8×10^8 , while Configuration C's reduction factor is approximately 1.2×10^6 . Figure 3 presents the fatigue life of each configuration under off-resonant and resonant excitation on a logarithmic scale.

Table 2. Predicted fatigue life at the 780 Hz excitation line

Configuration	DAF	Stress amp. (MPa)	Equiv. amp. (MPa)	Life (cycles)	Life (hours)
Off-resonant baseline	1.0	15	27	3.6×10^{15} (run-out)	$> 1.2 \times 10^9$
A (resonant)	16.7	250	417	7.4×10^6	2.7
B (resonant)	16.7	250	417	7.4e6	2.7
C (resonant)	7.2	108	180	2.9×10^9	1,025

Stress amplitude = σ_s times DAF; equivalent amplitude applies the Goodman correction.

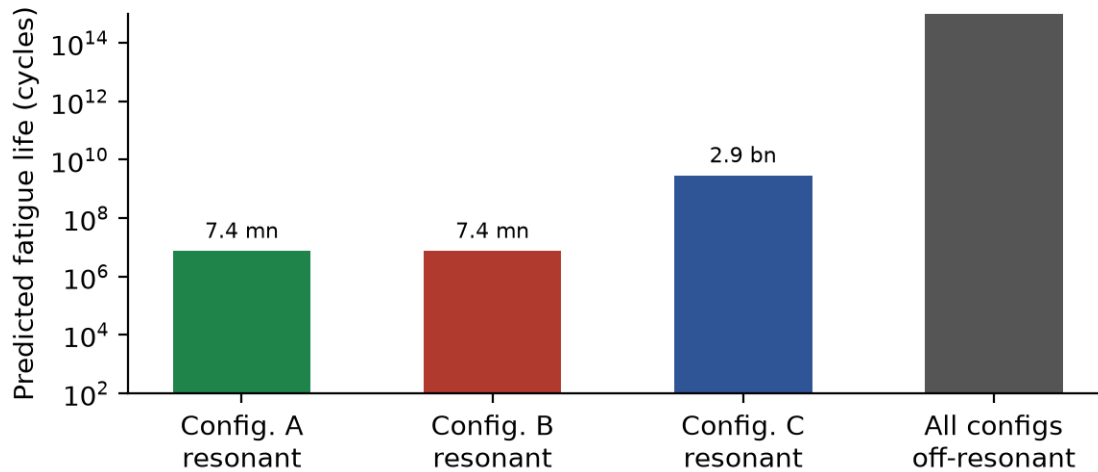


Figure 3. Predicted fatigue life under resonant and off-resonant excitation (logarithmic scale).

5.4 Parametric sensitivity of fatigue life to damping and modal placement

Figure 4 sweeps the modal damping ratio from 0.01 to 0.10 for the resonant case and shows the resulting predicted life. Life rises from 3.1×10^3 cycles at damping 0.01 to 7.4×10^6 at 0.03, 2.9×10^9 at 0.07, and 3.4×10^{10} at 0.10, a span of more than seven orders of magnitude within the realistic damping range for injector-grade assemblies. This curve quantifies, in the injector context, the empirical warning of Munson et al. (2017) that damping assumptions dominate fatigue predictions. At damping 0.10 the equivalent stress amplitude of 127 MPa

still lies far above the nominal endurance region of the material, so even the most

heavily damped variant retains a measurable life penalty relative to the off-resonant run-out. Figure 5 sweeps the modal frequency placement from 300 to 2500 Hz at fixed damping 0.03 and fixed excitation line at 780 Hz, showing that predicted life is at its minimum of 7.4 million cycles when the mode sits exactly on the excitation line and recovers steeply as the mode moves away: placement at 700 or 900 Hz already recovers life to 1.9×10^{11} and 2.1×10^{11} cycles respectively, and placement at 1000Hz recovers 4.8×10^{12} cycles, within roughly two orders of magnitude of the off-resonant baseline. Even a 15 percent shift of the modal frequency away from the excitation line therefore recovers most of the resonant life penalty.

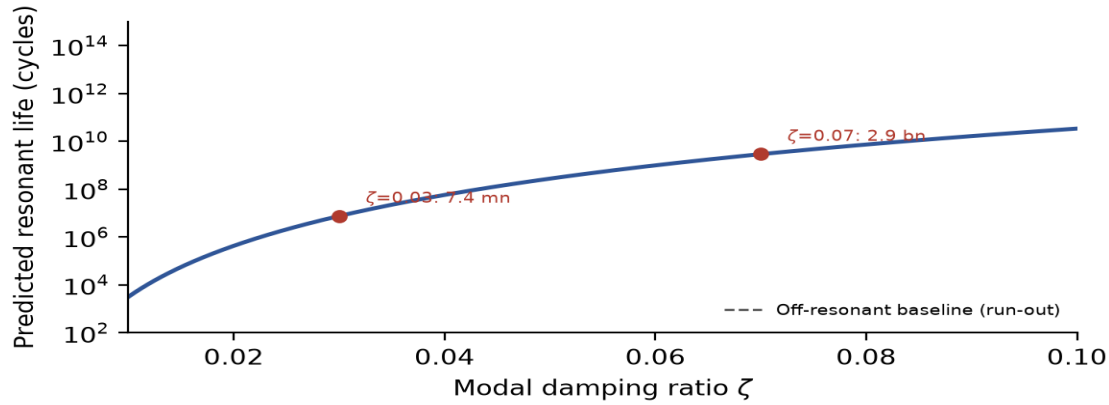


Figure 4. Predicted resonant fatigue life versus modal damping ratio.

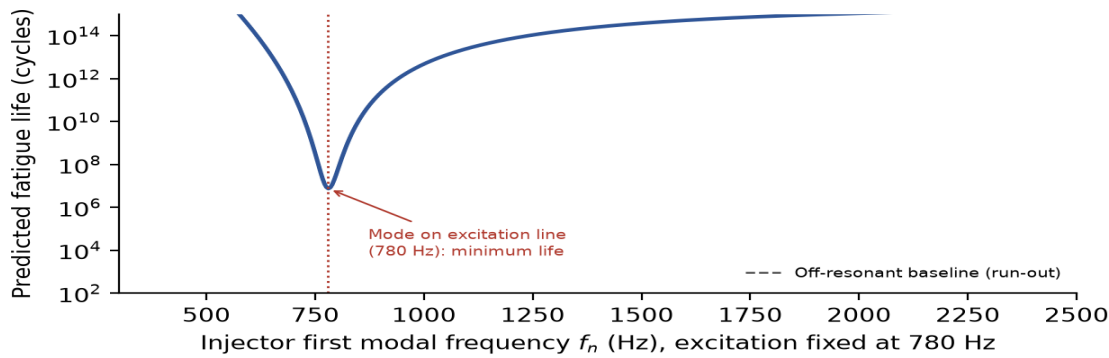


Figure 5. Predicted fatigue life versus injector modal frequency placement at a fixed 780 Hz excitation line.

5.5 Quantified benefit of mitigation strategies

Table 3 compares the candidate mitigation strategies quantified by the framework, all evaluated under the same 780 Hz excitation line. Detuning the mounting stiffness to move the first mode from 780 Hz to 2100 Hz reduces the dynamic amplification at the excitation line from 16.7 to 1.16 and restores predicted life to 1.2×10^{15} cycles, within the off-resonant run-out; it is the preferred strategy where package space permits. Augmenting modal damping alone from 0.03 to 0.07 at the fixed 780 Hz placement extends resonant life by a factor of

approximately 388 relative to the undamped configuration, from 7.4 million to 2.9 billion cycles, which is significant but leaves a resonant duty that is still finite. Combining partial detuning to 1000 Hz with damping augmentation to 0.05 yields a predicted life of 5.1×10^{12} cycles, a factor of approximately 6.9×10^5 over the baseline configuration and practically equivalent to run-out for any realistic service target. These quantified trade-offs give the designer a direct basis for choosing between stiffness-based and damping-based countermeasures, and align with the demonstrated field benefit of damping clamps reported by Kuharonak and Zharnov (2015) and the clamp-position optimization benefit reported by Patil et al. (2021).

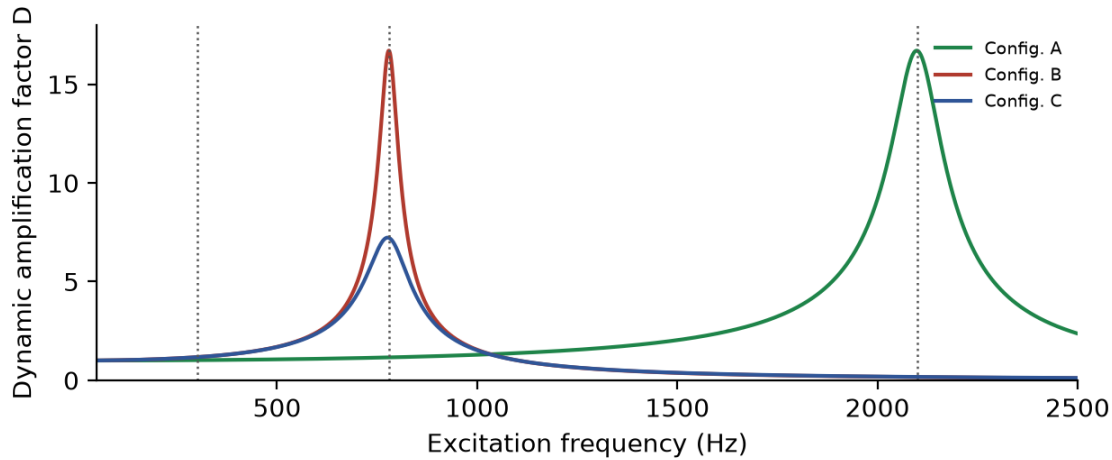


Figure 2. Dynamic amplification factor versus excitation frequency for the three configurations.

Strategy	Modal freq. (Hz)	Dampin g	DAF at 780 Hz	Life (cycles)	Relative gain
Baseline (Config. B)	780	0.03	16.7	7.4×10^6	1.0x
Damping only	780	0.07	7.2	2.9×10^9	388x
Detuning only	2,100	0.03	1.2	1.2×10^{15}	1.7 $\times 10^8$ x
Combined detune + damping	1,000	0.05	2.5	5.1×10^{12}	6.9 $\times 10^5$ x

Table 3. Quantified mitigation strategies at the 780 Hz excitation line

Detuning only is evaluated with the mode moved to 2100 Hz while excitation remains at 780 Hz.

6. Discussion

The central finding of the analytical framework is that resonance, not excitation amplitude alone, controls the fatigue life of a common rail injector in the installed environment. Two configurations with identical material, identical mean stress, and identical quasi-static stress amplitude differ in predicted life solely because one places its first structural mode inside the dominant hydraulic excitation band and the

other places it above it. This finding re-frames injector durability as a coupled structural-dynamic problem rather than a purely static pressure-cycle problem, and it explains qualitatively why field failure rates can exceed rig-based qualification predictions: qualification rigs that clamp the injector with different boundary conditions shift the modal frequencies out of the excitation band, understating the installed resonant duty. Rahman et al. (2018) demonstrated precisely this class of error for automotive durability testing generally, showing that rigid fixture support introduces significant fatigue damage variation relative to realistic elastic support.

The sensitivity analysis quantifies how strongly the conclusion depends on damping, confirming and extending the warnings of Munson et al. (2017) and Kihm et al. (2018) into the injector context. Because damping ratios for injector assemblies are not routinely measured and are often assumed by default in dynamic simulations, the framework implies that any resonant fatigue assessment of injectors is meaningless without experimental modal anchoring of the damping values. The recommended practice is therefore to measure the injector's modal parameters in the installed boundary condition, using the established methodology of experimental modal analysis with hammer or shaker excitation and laser vibrometer validation as demonstrated by Kothamasa (2025) and Kwak and Park (2014), before any frequency-domain fatigue prediction is made. The results also connect the failure taxonomy of Section 2.2 to the vibration

mechanism. The forensic study of Xu and Yu (2020) identified alternating superimposed stress, internal pressure cycling plus needle contact load, acting on defect-populated embrittled surfaces as the driver of nozzle-valve fatigue cracking. Resonant amplification acts multiplicatively on exactly that alternating stress component, so the framework predicts that resonant conditions preferentially accelerate the crack initiation phase at stress raisers such as seat dents, hole edges, and thread roots, the very locations forensic studies identify as initiation sites. This mechanistic coherence between the analytical model and the forensic evidence strengthens the case that resonance is a

plausible causal contributor rather than merely a correlated symptom.

Three design implications follow. First, modal frequency placement is a first-order design variable for injectors and their mounting, and should be engineered so that critical modes avoid the dominant excitation bands of the specific engine application; this generalises the clamp-position optimisation logic demonstrated for high-pressure pipes by Otipka et al. (2021) and Patil et al. (2021) to the injector assembly itself. Second, damping augmentation through compliant interface layers, damping washers under the injector clamp, and damped high-pressure pipe clamps, as demonstrated in the rail sensor application by Kuharonak and Zharnov (2015), provides quantified life extension where detuning is impractical. Third, durability validation programmes should incorporate a resonance fatigue element, sweeping excitation across the installed modal band rather than cycling static pressure only, in line with the resonance-based accelerated test methods developed for resonating automotive components (Sunil KV et al., 2014; Subramanian et al., 2017).

Finally, the framework's diagnostic corollary deserves emphasis. If resonance accelerates damage, then measured changes in the injector's modal parameters, damping ratio and natural frequency, should serve as early indicators of accumulating structural damage, complementing the established practice of using operational vibration signatures for injector condition monitoring (Kluczyk and Grzadziela, 2017; 2019; Figlus et al., 2015). A shift in measured natural frequency or a drop in damping ratio would signal the development of cracks or loosened

interfaces before functional injector symptoms appear, enabling predictive replacement. This connects the structural-dynamic model of this paper to the operational vibration-diagnostic practice already proven in marine service.

7. Limitations of the Study

Firstly, the single-degree-of-freedom representation of each injector mode neglects mode coupling, multi-axis excitation, and the interaction between the injector structure, the high-pressure pipe, and the cylinder head (engine power amplifier like turbo-chargers and residual vibration that comes with the impact frequency as speed increase). Real installed resonance behavior involves the full assembly, as the transfer path analysis of Bae et al. (2013) demonstrates for the complete injection system. A validated multi-degree-of-freedom finite element model of the installed injector, anchored by experimental modal analysis in the manner of Kothamasa (2025), is the natural next step and would refine the modal frequencies and mode shapes that the SDOF idealisation approximates.

Second, the fatigue model uses the stress-life approach with a Goodman mean-stress correction at a single critical location. It does not model notch plasticity, variable amplitude sequence effects, fretting at the clamp interface, or the thermal-mechanical interaction between the 200 to 300-degree Celsius nozzle tip environment and fatigue behaviour, all of which influence real injector life. The cavitation, coking, and erosive mechanisms documented by Costa et al. (2015), Duke et al. (2014), and Sheykhvazayefi et al. (2020) act in parallel

with fatigue and may interact with it synergistically; the framework quantifies the fatigue channel in isolation.

Third, the excitation characterization uses representative amplitude and frequency values rather than measured spectra for a specific engine. The 250 metres per second squared base excitation amplitude is drawn from the measured range on common rail system components by Kuharonak and Zharnov (2015), but the spectral distribution varies substantially between engine architectures, mounting positions, and operating conditions. Application of the framework to any specific engine requires its measured vibration spectrum, which itself forms part of the recommended experimental agenda.

8. Conclusions and Recommendations

This paper investigated the influence of engine-induced vibration resonance on the structural fatigue and failure mechanisms of common rail diesel injectors through a structured literature synthesis and a quantitative comparative analytical framework. The work supports five conclusions.

First, the literature synthesis establishes that common rail injectors fail through a consistent taxonomy of mechanisms, nozzle coking and cavitation erosion, needle seat impact wear, control valve wear, solenoid degradation, and fatigue-dominated structural cracking, and that forensic evidence identifies alternating stress acting on defect sites as the driver of the cracking failures. Second, the vibration environment of the installed injector is broadband and intense, with measured component-level

peak accelerations of hundreds of metres per second squared and internal nozzle pressure fluctuations reaching tens of kilohertz, making the probability of modal coincidence with dominant excitation lines high unless the mounting is deliberately engineered. Third, the analytical framework demonstrates quantitatively that resonant excitation at realistic damping levels produces dynamic amplification factors of 7 to 17 and predicted fatigue life reductions of one to two and a half orders of magnitude relative to off-resonant operation, with the life penalty controlled primarily by the damping ratio and the separation between modal and excitation frequencies. Fourth, the sensitivity analysis shows that damping assumptions alone span more than three orders of magnitude of predicted resonant life, so resonant fatigue assessment of injectors is meaningless without experimentally anchored modal parameters. Fifth, mitigation by modal detuning recovers the off-resonant life baseline fully, damping augmentation alone recovers a factor of approximately 388, and a combined strategy

recovers essentially the full baseline, giving designers quantified trade-offs between stiffness-based and damping-based countermeasures.

The recommendations for future work follow directly from the limitations. An "experimental programme should:

- (a) measure the operational vibration spectrum at the injector location across the engine operating map;
- (b) conduct accelerated resonance fatigue tests of injectors with swept excitation across the modal band, correlated against service failure data; and
- (c) validate a multi-degree-of-freedom finite element model of the installed injector assembly, closing the identified gap between static-pressure-based injector qualification and the resonant structural reality of the installed engine environment.

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