4B17 Multibody Dynamics: Main Assignment

Finn O'Connor — 22336740 — foconno1@tcd.ie

1 Introduction

The performance of an internal combustion engine depends strongly on its valvetrain dynamics, particularly the ability of the valve spring to keep the head of the pushrod in continuous contact with the cam profile at all operating speeds. Poor valve timing can alter the amount of air that can enter the cylinder before combustion occurs, which can lead to decreased engine performance at high RPM [1]. The research question for this assignment is therefore:

How does valve spring stiffness affect valvetrain dynamics and engine performance in an internal combustion engine at high RPM?

Understanding this relationship is relevant for both performance tuning and durability. Stiffer springs can reduce valve float at high RPM, but they may also increase cam–follower contact forces, friction losses, and wear. Multibody dynamics provides an appropriate framework for modelling this behaviour because the cam–pushrod–rocker system involves geometry-dependent contact and rapid changes in acceleration that are difficult to capture analytically. The purpose of this assignment is to model the contact between the cam and rocker and the resulting valve kinematics in an internal combustion engine.

In this model, the cam, pushrod, and rocker are treated as rigid bodies through the use of solid-body contacts in SolidWorks Motion Analysis. This enables the normal force required to maintain physical contact between the bodies to be calculated. This allows a penalty-based contact model of the system to be created, which simulates real-life rigid-body interactions that arise from penetration depth, relative velocity, material properties, and local geometry.

Through the use of a rotational motor applied to the camshaft in the model, the relationships between cam geometry, spring stiffness, friction, impact velocity, and other contact mechanics combine to produce valve kinematics. The SolidWorks simulation outputs data as .csv files, which are then post-processed in MATLAB for analysis.

2 Geometry and Model Setup

Using SolidWorks, a model of an internal combustion engine was created. This included a crankshaft, crankshaft sprocket, connecting rod, piston, cylinder, camshaft, camshaft sprocket, pushrods, rockers, springs, and valves. Some of the parts were imported and others were built from scratch. The system was assembled using a series of mates and could be rotated as a 3D model, simulating the four-stroke behaviour. For the motion analysis, some mates had to be replaced with solid-body contacts. These included the cam mate between the cam and the pushrod, and the tangential and parallel mates between the pushrod and the rocker.

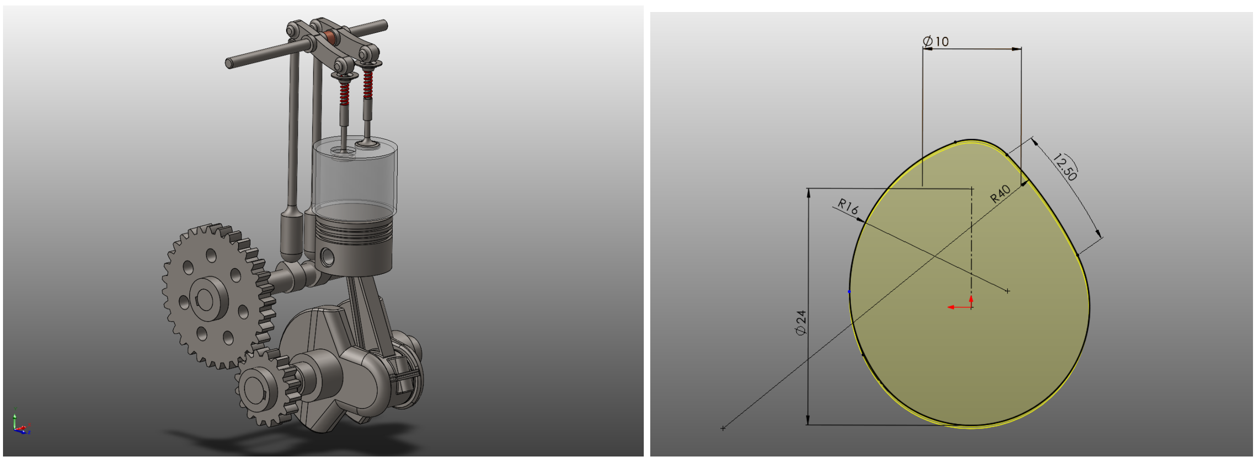

Figure 1: CAD assembly of the engine system and cam geometry used for the multibody analysis.

The cam had a base circle radius of 12 mm and a maximum lobe radius of 17 mm. For the sake of simplifying the analysis, some explicit assumptions were made. All parts were assumed to be rigid. There were no thermal or combustion loads accounted for— all loading came from the motor torque and spring stiffness. Friction was neglected in all contacts; however, friction would be very low in a real system due to lubricants anyway.

To maintain independence of the variables being studied in the system, the camshaft speed was set to 2000 RPM. This corresponds to a crankshaft speed of 4000 RPM (due to the gear ratio being 2:1), which is close to what a 1.6 L Volkswagen Golf TDI runs at when nearing a maximum power output of 104 hp [2].

$$\omega = \frac{2\pi}{60} \times \mathrm{RPM} = \frac{2\pi}{60} \times 2000 = 209.44\ \mathrm{rads}^{-1} \tag{1}$$

This corresponds to a single revolution taking $60 \div 2000 = 0.03$ s. The WSTIFF solver was used, with an FPS of 1000, and gravity was applied. The material of choice for all components was AISI 4340 annealed steel. It is a common type of steel used in combustion engines and has an elastic modulus and density of 205 GPa and 7850 kg/m^2. After evaluating the mass properties of the valve, rocker, and pushrod, their masses were 30.49, 106.34, and 173.62 grams respectively, and the combined mass of the system being pushed by the spring is approximately 0.310 kg.

3 Contact Model

To simulate contact between rigid bodies in the system, solid-body contact relations were used. As previously stated, these relations were configured without friction for the simulation. In an actual configuration, real-life cam–follower contact is nonlinear, transient, highly stiff, and includes energy loss/dissipation. However, this relationship can be modelled through the penalty-contact formulation. In SolidWorks, for a solid-body contact relation, the 'Elastic Properties' component allows configuration of this penalty contact relation. Parameters such as contact stiffness, force exponent, damping, and penetration depth can be adjusted to best simulate an actual rigid-body contact. It is important to model these parameters correctly to ensure accurate results. The force–penetration contact model has been covered previously in 4B17. From Newton's second law we know that:

$$\vec{F}_{\text{ave}} = \frac{m\,\Delta\vec{v}}{\Delta t} \qquad [\mathrm{N}] \tag{2}$$

However, this approach to contact force modelling does not take into account the effects of stiffness and damping. In 1882, Hertz derived that for elastic contact between two curved bodies [3]:

$$F = K\delta^{1.5} \qquad [\mathrm{N}] \tag{3}$$

The exponent $n = 1.5$, derived by Hertz, corresponds to Hertzian elastic contact between two curved bodies, which is appropriate for a cam and tappet surface. The classic force–penetration damping law was introduced by Hunt & Crossley in 1975 [4]. They related normal contact force to penetration rate, through stiffness and damping coefficients, which is the basis of how solid-body contact works in software such as SolidWorks or MADYMO.

$$F_n = k\delta^{1.5} + c\delta^{1.5} \qquad [\mathrm{N}] \tag{4}$$

This model does not take friction into account, which is consistent with our assumption of a frictionless system. The system was modelled as a steel-on-steel contact. The remaining parameter values were kept the same as those recommended by SolidWorks for this type of impact.

Stiffness, k (N/mm)	**Max Damping, c (Ns/mm)**	**Penetration, δ (mm)**
1×10^5	49.92	0.02

Table 1: Contact parameters used for solid-body relations in the penalty contact model.

These values can be used to model the peak contact force of the cam–rocker interaction, which ultimately yields a value that can be verified by the simulation for a stock spring system in Section 4.

$$F_n = (1 \times 10^5)(0.02)^{1.5} + (49.92)(0.02)^{1.5} \approx 283\ \mathrm{N} \tag{5}$$

4 Spring Model

The spring rate (otherwise known as the spring stiffness) is a measure of the extent to which an object resists deformation in response to an applied force. The spring stiffness can be calculated via:

$$k = \frac{(\text{Open Pressure } - \text{Seat Pressure})}{\text{Lift}} \qquad [\text{N/mm}] \tag{6}$$

A reference spring model was used in the analysis. Based on 'Upgraded Valve Springs for 1.6 / 2.0 16v & 2.7 / 3.0 24v TDI Common Rail Engines' on the Darkside Developments website [5], spring performance data were gathered for use in analysis.

Spring Type	**Seat Pressure (N)**	**Open Pressure (N)**	**Lift (mm)**	**Spring Stiffness k (N/mm)**
Stock Springs	195.81	378.27	8.5	21.47
Performance Springs	342.66	605.18	8.5	30.88

Table 2: Comparison of stock and performance valve spring characteristics.

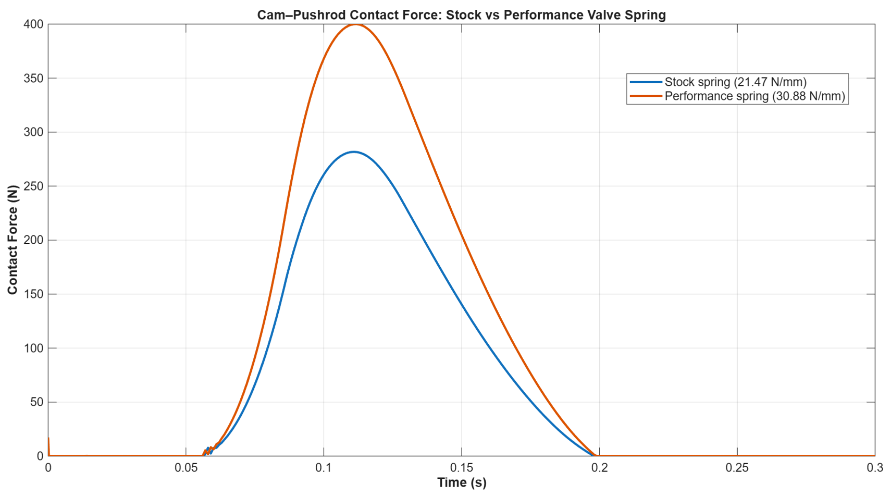

Figure 2: Cam–pushrod contact force for stock vs. performance valve springs over one engine cycle.

The peak force for both springs occurs when the cam nose (radius = 17 mm) is in contact with the bottom of the pushrod, or equivalently when the valve is at bottom dead centre. It is clear from the above data that by increasing the valve spring stiffness, the magnitude of the contact force also increases.

5 Motion Results

5.1 Rocker Angular Velocity

The rocker angular velocity was studied to determine whether the upgraded springs alter the timing or magnitude of the rocker motion. A peak rocker angular velocity of $\approx 384\ ^{\circ}/\text{s}$ occurs at a valve vertical displacement of ≈ 4 mm. The difference between the springs in terms of the angular velocity of the rocker is negligible; both deliver similar performance. There is a slight decrease in the magnitude of velocity, which is expected for springs with a higher stiffness value. Overall, the change in spring stiffness does not affect rocker angular velocity performance, but as shown in Figure 2, the contact force is greatly influenced.

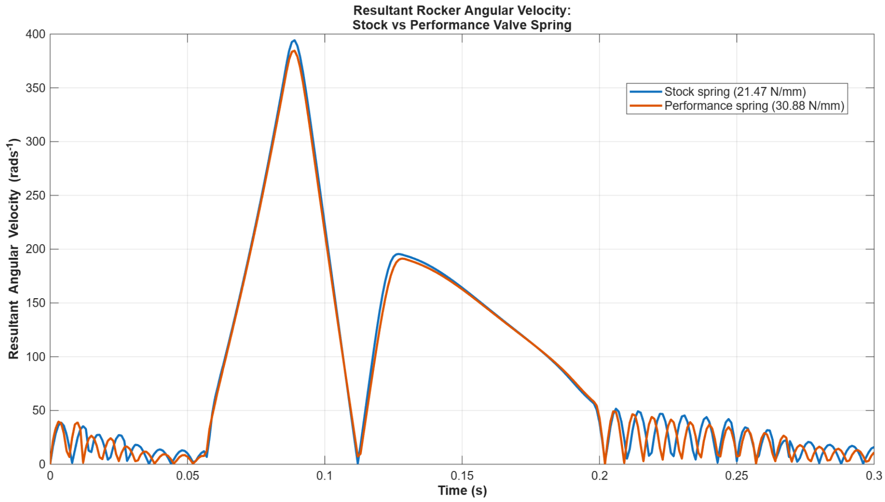

Figure 3: Rocker angular velocity over one camshaft revolution for stock and performance valve springs.

5.2 Valve Kinematics

The valve kinematics were also studied to determine whether upgraded springs have significant influence on them. The y-axis linear displacement, velocity, and acceleration were analysed. The maximum valve lift did not change appreciably, both springs giving a lift of ≈ 8 mm; however, the performance spring produced slightly less lift (≈ 0.2 mm). Similarly to the rocker angular velocity, this is expected due to the nature of having stiffer springs. Both springs had similar transient oscillation decay. The same applies to velocity and acceleration. The stiffer performance springs resulted in decreased overall velocity and acceleration, but they produced more stable transient oscillations after the valve movement. However, this is not a major factor in performance, as the majority of these oscillations can be reduced using a spring damper, as shown in Section 6.

Overall, for this particular simulation setup, at a camshaft speed of 2000 RPM, the performance springs increase force but do not enhance valve timing or lift in an impactful way.

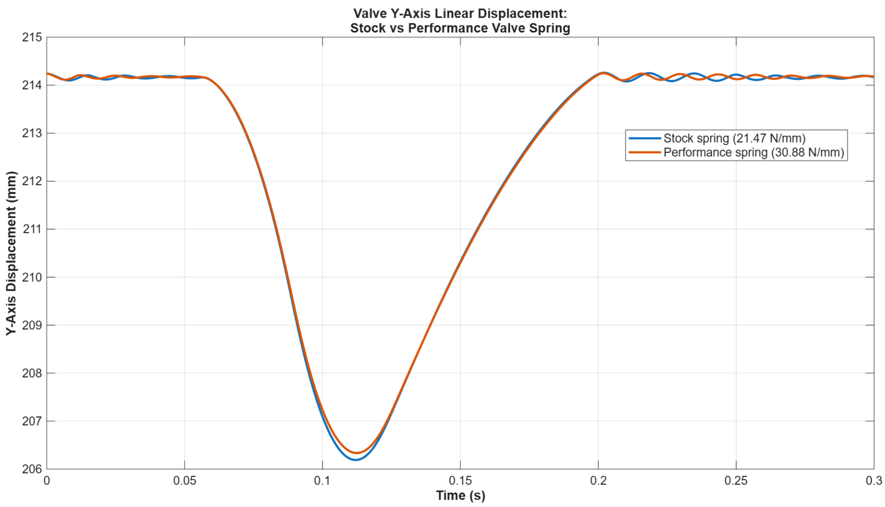

Figure 4: Valve vertical displacement for stock vs. performance springs.

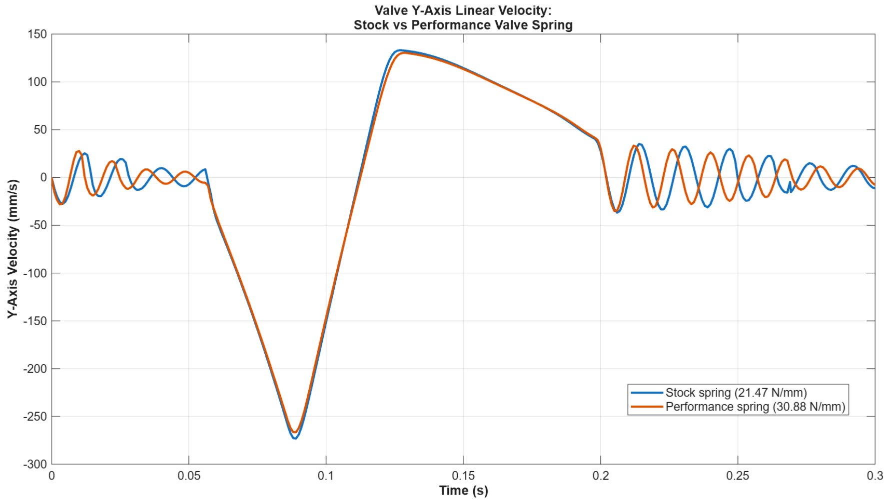

Figure 5: Valve vertical linear velocity for stock vs. performance springs.

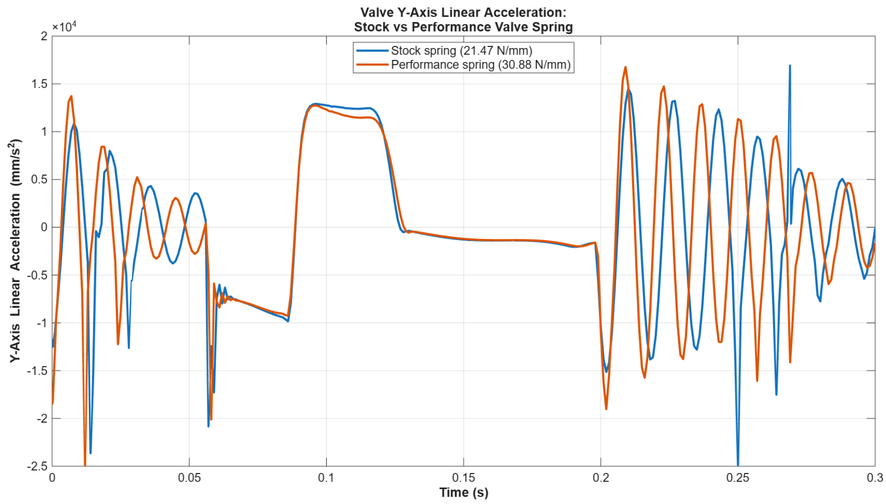

Figure 6: Valve vertical linear acceleration for stock vs. performance springs.

6 Damper Investigation

For each simulation, a damper was not used in either spring. However, it can be seen in each plot that, before and after the movement of the valve, there is some degree of transient oscillation. In an ICE valvetrain system this is not desirable and should ideally be addressed, as over time it can cause mechanical fatigue, noise and vibration, and disrupted combustion efficiency. To reduce this vibration, a damping coefficient can be used.

The valvetrain can be modelled using the standard second-order model for a single-DOF mass–spring–damper, where, for simplicity, the mass of the components is lumped into one:

$$m\ddot{x} + c\dot{x} + kx = 0 \tag{7}$$

To ensure that the chosen RPM was suitable for analysis, we check that the system is not oscillating at its natural frequency. This can induce resonance, which can lead to unpredictable behaviour and a risk of structural failure in real systems.

$$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{30880}{0.31045}} \approx 315 \text{ rads}^{-1} \qquad \rightarrow \qquad f_n = \frac{\omega_n}{2\pi} \approx \frac{315}{2\pi} = 50 \text{ Hz} \tag{8}$$

$$f_{\text{cam}} = \frac{2000}{60} = 33.33 \text{ Hz} \tag{9}$$

From this we can see that the valve spring would oscillate around its natural frequency of 50 Hz if it were underdamped ($c = 0$), but for a cam at 2000 RPM the excitation occurs at 33.3 Hz. Therefore, the critical damping for this system is:

$$c_{\text{crit}} = 2\sqrt{km} = 2\sqrt{30880(0.31045)} = 195.8 \text{ N/m} \approx 0.196 \text{ Ns/mm} \tag{10}$$

The damping ratio (ζ) is defined by:

$$c = \zeta\, c_{\text{crit}} = 0.196\, \zeta \tag{11}$$

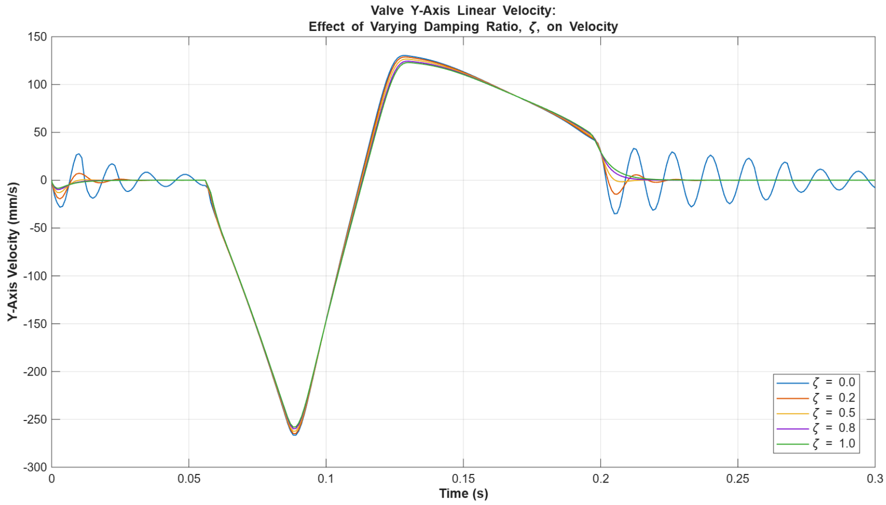

Figure 7: Effect of damping ratio ζ on cam–pushrod contact forces.

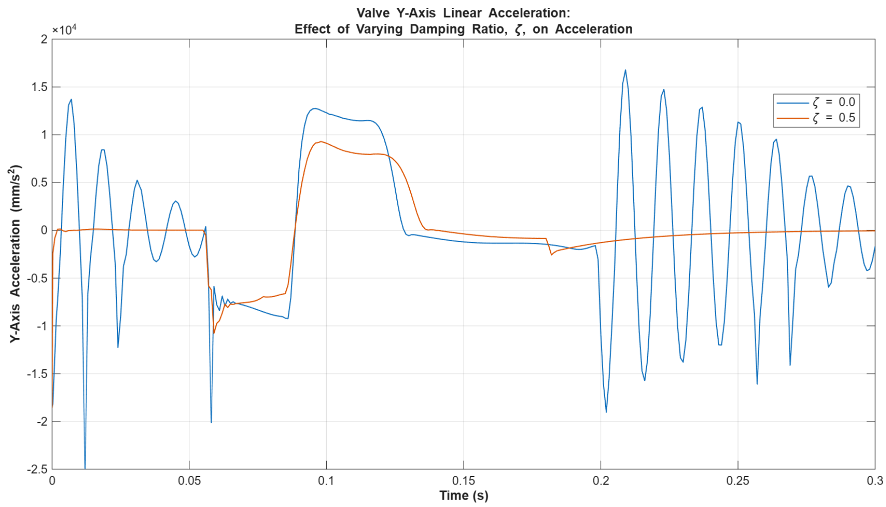

Figure 8: Valve acceleration response for different damping ratios illustrating oscillation suppression.

From the plots, we can clearly see the effects of varying the damping ratio and how it helps reduce transient oscillations. Looking at the contact force output, for $\zeta = 1$ the peak force was ≈ 412 N, compared to a peak force of ≈ 400 N for $\zeta = 0$. A value of $\zeta = 0.5$ appears to be the best, as it offers good damping performance without over-constraining the system. The use of $\zeta = 0.5$ massively improves stability and reduces transient oscillations for valve acceleration.

7 Conclusion

At a fixed camshaft RPM of 2000, when switching from stock ($k = 21.47$ N/mm) to performance ($k = 30.88$ N/mm) springs there is an increase in cam–pushrod contact force ($\approx 40\%$), but it has a negligible effect on valve lift, timing, or rocker angular velocity. Overall, stiffer springs do not massively improve engine performance at this operating point; however, they do offer greater stability against valve float. Many factors were neglected in arriving at this conclusion (rigid bodies, no friction, no combustion load, or energy transfer), but the results indicate that upgrading to performance springs yields only a negligible performance gain at 2000 RPM. If the simulations were to be carried out again, accuracy could be improved by including other factors that affect performance and by testing the engine over a wider range of camshaft speeds.

References

[1] Yanfei Qiang, Changwei Ji, Shuofeng Wang, Gu Xin, Chen Hong, Zhe Wang, and Jianpu Shen. Study on the effect of variable valve timing and spark timing on the performance of the hydrogen-fueled engine with passive pre-chamber ignition under partial load conditions. *Energy Conversion and Management*, 302:118104, February 2024.

[2] Volkswagen Golf Mk7, November 2025. Page Version ID: 1320190182.

[3] Heinrich Hertz. *Miscellaneous papers*. London: Macmillan, New York, Macmillan and co., 1896.

[4] K. H. Hunt and F. R. E. Crossley. Coefficient of Restitution Interpreted as Damping in Vibroimpact. *Journal of Applied Mechanics*, 42(2):440–445, June 1975.

[5] Upgraded Valve Springs for 1.6 / 2.0 16v & 2.7 / 3.0 24v TDI Common Rail Engines.

Acknowledgement of Generative AI and Plagiarism

The work produced in this assignment is 100% human. No form of generative AI use or plagiarism were conducted in the process.

Signed: Finn O'Connor