

McNair Scholars Research Journal

Volume 11 | Issue 1

Article 6

2018

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Recommended Citation

Green-Warren, Robert A. (2018) "An Experimental Investigation of Torque Loss in a Die Cast Aluminum Threaded Fastener Joint," *McNair Scholars Research Journal*: Vol. 11 : Iss. 1 , Article 6.
Available at: <https://commons.emich.edu/mcnair/vol11/iss1/6>

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AN EXPERIMENTAL INVESTIGATION OF TORQUE LOSS IN A DIE CAST ALUMINUM THREADED FASTENER JOINT

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ABSTRACT

The intent of this study was to investigate experimentally the root cause of torque loss in a die-cast aluminum threaded fastener joint that is used to assemble an engine mount to the front cover of an automotive engine. The primary goal was to develop a practical experimental method to characterize threaded fastener joints in situ. Using installation torque measurements, hardness test results, and a DC nut runner, joint attributes were measured, and the data was evaluated. This investigation demonstrated that the incompatibility of the mechanical properties at the interface of the housing and fastener resulted in significant torque loss. Further investigations are planned to study cost effective design modifications.

INTRODUCTION

Threaded fasteners (i.e., bolts, screws, nuts, and studs) are common machine components that are used to connect structural members. Their primary function is to create and maintain a clamping force between joint members. In other words, threaded fastener joints behave like springs; the bolt stretches, and the joint members compress [1]. Although threaded fasteners are enablers for ease of assembly and allow access to machine components and structural members for maintenance, failure can be catastrophic.

Hundreds of variables affect the tightening process and in-service response of threaded fastener joints [1]. Because of this, design engineers are faced with a number of challenges when predicting the clamp load in a threaded fastener joint design. For example, in a typical automotive assembly plant, the clamp load is controlled by the tightening torque. However, published

experimental results have concluded that approximately 10% of the tightening torque is used to stretch the bolt [1]. Note that clamp load is established after a sufficient amount of bolt stretch or load is generated. The remaining 90% of the tightening torque is lost from processes such as, but not limited to: friction at the internal-external thread interface; friction at the bolt or nut flange-clamped member interface; shank (threaded and unthreaded regions) angle of twist; nut dilation. For a given threaded fastener design, quantifying “torque robbing” processes, such as those stated above, is challenging. For these reasons, there is a continued need for applied investigative studies to develop experimental and analytical methods to accurately predict the clamp load in threaded fastener joints.

Several researchers have previously studied the behavior of threaded fastener joints. The published works are diverse. For example, several research teams studied the temperature-dependent behavior of steel bolted joints [2], [3], [4], [5], [6]. Another group of investigators studied the temperature-dependent behavior of aluminum bolted joints [7]. Notably, a team of investigators experimentally studied the high temperature viscoelastic response of a bolted aluminum joint used in an engine cylinder head without the gasket [8]. They measured the bolt load loss in situ using strain gauges. They concluded, after one week at temperature, that the residual bolt load was zero at temperatures above 260°F.

Research that considered the fatigue behavior of bolted joints has also been published [9], [10], [11]. Remarkably, the effect of the bolt tightening torque on the fatigue life of an aluminum alloy bolted joint was investigated [12]. These researchers constructed a bolted joint comprised of an aluminum alloy (7075-T6) plate with a 5-mm bolt hole and an M5 bolt and nut assembly (tension type joint). The joint was then subjected to a single axis cyclic load to derive stress-life (S-N) curves experimentally as a function of clamp load. A steel washer was placed between the nut and plate to measure the clamp load. As expected, their results showed that fatigue life increases with clamp load.

To predict the clamp load in threaded fastener joints, computer aided engineering (CAE) modeling methods have also been previously published. Investigators used finite element analysis (FEA) to predict the stress and strain distribution in a

bolted aluminum joint subjected to the combined effects of clamp load and an axial force [12]. In another study, researchers developed an FEA modeling method to study rotational self-lessening in a bolted joint subjected to a transversely applied harmonic load [13]. In both studies, experimentally measured values were used to verify the accuracy of the CAE modeling results. In this paper, an experimental investigation was conducted to determine the root cause of torque loss in a die cast aluminum thread fastener joint.

METHODS AND MATERIALS

The initial challenge was to design and install the infrastructure to support the systematic investigation of threaded fastener joints. To this end, a state of the art threaded fastener testing lab was designed and commissioned (Figure 1, below). The primary feature of the threaded fastener lab was the DC electrical tool system. Auxiliary measuring equipment included

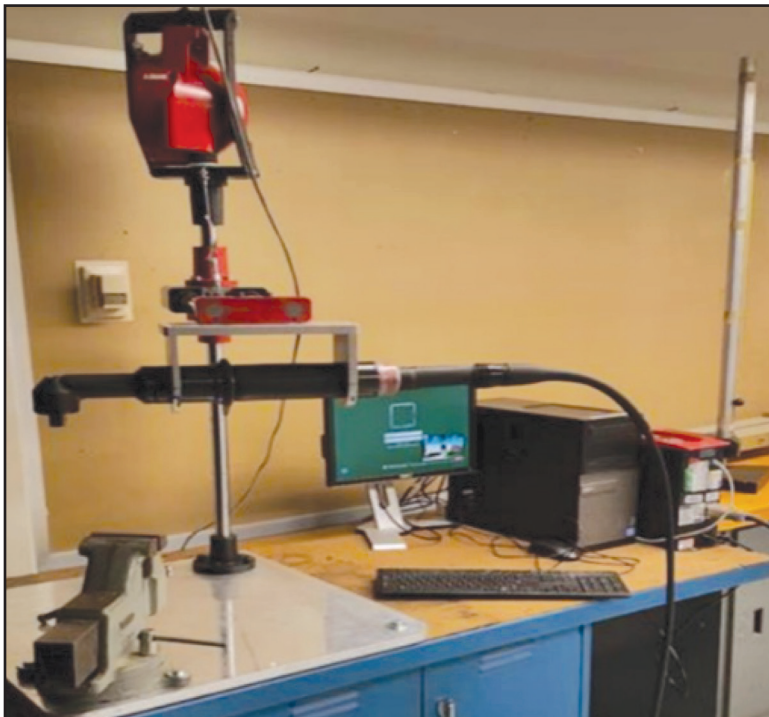


Figure 1. DC electrical tool system equipment and set up.

strain gauges for measuring bolt stretch and pressure paper for measuring pressure fields between contact surfaces. The controller allowed the operator to program a vast array of fastener tightening sequences, including up to 15 different phases powered by a CVIL II L Desoutter controller. These phases and sequences allowed the user to create multiple arrangements of cycles, depending on the required results. The CVIL II L also measured the corresponding torque versus time (angle) curve.

Referring to Figure 1, the electric wrench had a torque capacity of 280 Nm. For operator safety, the wrench was secured by a 300 Nm torque reaction arm. This prevented tool forces from being transferred to the operator. For additional operator safety, the sockets contained protective sleeves that rotated freely, which mitigated the potential for fingers being trapped on the socket. The controller allowed the operator to program a variety of fastener tightening strategies, including up to 15 different steps. It also measured the corresponding torque versus time (angle) curves and torque trace. This allowed us to replicate and evaluate real world joint behavior.

Experimental Method

Torque Loss Measurements.

The engine mount housing and engine front cover bracket were manufactured from die cast aluminum A380 and cast iron, respectively. The bolts' holes on the engine mount housing consisted of 1 – locating hole (10.2 mm diameter) and 3 – locating holes (10.5 mm diameter) (Figure 2). The threaded fastener joint was secured by tightening the four nuts to the nominal value of the torque specification, 62.5 ± 9.4 Nm, using a torque wrench. To minimize the effects of elastic interactions, several tightening passes were performed. Note that a similar tightening process was used during the prototype vehicle build. For production, however, the nuts were secured simultaneously, using an automated nut runner system. After one week, the residual torque of each nut-stud assembly was measured to determine the percentage of initial torque loss in each bolt. The percentage of initial torque loss was calculated using the following equation:

$$\% \text{ Difference} = \frac{|T_{\text{initial}} - T_{\text{measured}}|}{\frac{1}{2}(T_{\text{initial}} + T_{\text{measured}})} * 100\% \quad (3)$$



Figure 2. Die Cast Aluminum Joint: The image above represents the initial test set up



Figure 3. Die Cast Aluminum

Surface Hardness Measurements

Hardness is a means of quantifying a metal's resistance to permanent deformation. Hardness tests were performed on an analog Rockwell Hardness testing machine on the three left-most holes, as the machine could not access the fourth hole due to complex geometry. These tests were conducted to analyze discrepancies between the hardness of the aluminum housing and the steel bolt. A 1/16" indenter was utilized, as determined by the aluminum alloy. The required applied load for the Rockwell tester was 100 kg, and the results were to be read by the Rockwell B scale. The average hardness and standard deviation of the three holes were calculated in Table 1. A comparative analysis of the steel nut and the aluminum housing was then conducted to determine the relative contribution of the difference in each material's hardness. [16].

Torque Trace Measurements.

Evaluating the torque trace is critical to the design of threaded fastener joints. The torque trace is a measure of the bolts' strength and behavior within the joint. Typical design characteristics that can be identified from the torque trace include, but are not limited to: prevailing torque, hole bottoming, friction, stiffness, thread stripping, and surface hardness mismatch (soft clamped materials). The torque trace was measured on a current production engine mount housing. The test specimen consisted of as-manufactured design intent components, (Figure 2.), including:

- Engine mount housing
- Engine front cover bracket
- 4 – M10 studs
- 4 – M10 nuts

Theoretical Method

The following theoretical framework was used to draw conclusions from the experimental results. The short-form of the torque versus preload equation [1] was considered in the work.(1).

$$T_{in} = F_p(KD) + T_p \quad (1)$$

Where: T_{in} = input torque

F_p = achieved preload

D_p = nominal diameter

K = nut factor

T_p = prevailing torque

The Bosch Automotive Handbook gives an equation for predicting K [14],

$$K = \frac{\left(0.16 p + \mu_t 0.58 D_p + \mu_p \frac{D_h}{2}\right)}{D} \quad (2)$$

Where: p = thread pitch
 = thread friction coefficient
 D_h = under head friction coefficient
 D = nominal thread diameter
 D_p = pitch diameter

Using equation (2), the nut factor for the M10 stud and nut assembly was calculated. The coefficient of friction under the head of the nut (steel-aluminum) and between the threads (steel-steel) was 0.5 and 0.15, respectively. These values were taken from published engineering tables. The diameter of the contact area under the nut flange was 23 mm, and the hole diameter was 10.5 mm. From this, the calculated pitch diameter was,

$$D_p = D - 0.6495p = 10 - 0.6495 (1.5) \\ = 9.03 \text{ mm.}$$

The effective underhead diameter was,

$$D_h = \frac{(14.5 + 10.5)}{2} = 12.5 \text{ mm} = 0.2$$

RESULTS & DISCUSSION

Using the experimental methods, as stated above, the replication of the torque loss phenomenon was achieved. It was found that, on average, up to 59% of the initial torque load was lost. Upon investigation, brinelling, or plastic deformation on the surface of a part due to an applied load, was found on the die-cast aluminum joint between the nut and joint interface (Figure 3). The locations that indicated brinelling were approximately equal to those of the bottom surface area of the M10 nut used in the bolt-joint assembly (Figure 4). Through

torque measurements, the yield loads were graphically represented as between the range of 90 Nm to 105 Nm (Graph 1, below).

ROCKWELL HARDNESS TEST			
Test	Hole1	Hole 2	Hole 3
1	53.1	51.5	50.9
2	51.5	48.8	50.5
3	50.5	49.6	52.5
4	50.3	50	51.8
5	48.3	51	52.5
Average	50.73	50.18	51.64
Standard Deviation	1.761	1.083	0.915

Table 1. B Scale Rockwell hardness data of aluminum housing based on 100 kg load, and 1/16" indenter.

Hardness

Based on the testing experimental methods described above, the following table was developed: Mechanical properties for the M10 property class 10 nut and stud have been developed and published and can be found in Table 2:

Metric Property Class	Size Range	Core Rockwell Hardness		Min. Yield Strength Mpa
		Min	Max	
10	M5-M39	C32	C39	940

Table 2. Rockwell hardness data for property class 10 bolt per ISO 898-1[15].

The results found in Table 1 and Table 2 illustrate the contrast between the bolt and the housing composed of a proprietary aluminum alloy. The C scale is utilized for a material that tends to display a Rockwell hardness above 100 on the B scale. Materials such as steel and cast iron refer to the C scale when testing for hardness [16].

TORQUE.

The preloaded torque of 62.5 Nm resulted in an axial force along the vertical axis of the stud. Since the joint was aluminum and tested in static conditions, stress concentrations due to

rapid changes in geometry were negligible. The clamping force was assumed to be uniformly distributed along the bottom surface area of the nut. Given that the nut and stud had a higher stiffness constant, once the bolt was secure and is torqued to the specification, atoms in the face centered cubic (fcc) microstructure began to shift along the crystallographic slip planes in the crystal lattice of the joint. These dislocations along the slip planes resulted in yielding on the surface of the aluminum housing along the bottom surface of the nut, as seen in Figure 3. Further, it is suggested that the root cause of torque loss was due to an increase in gage length of the bolt, as the surface of the housing exemplified in elastic deformation through a reduction in initial clamp load. This increase in bolt length resulted in the decrease of the clamping force of the joint. By equation (1), a decrease in clamping force is directly proportional to applied (or residual) torque.

The effects of this analysis are illustrated in the following tables:

Preload Torque Loss		
Bolt	Torque (Nm)	Time
1	10%	3:28:00PM
2	44%	3:35:00 PM
3	59%	3:39:00 PM
4	51%	3:42:00 PM

Table 3. Preloaded torque.

Residual Torque Measurements		
Bolt	Torque (Nm)	Time
1	56.6	2:45:00PM
2	40	2:48:00 PM
3	34.2	2:50:00 PM
4	37.2	2:52:00 PM

Table 4. Residual torque after one week.

Initial Torque Measurements		
Bolt	Torque (Nm)	Time
1	62.5	3:28:00PM
2	62.7	3:35:00 PM
3	62.5	3:39:00 PM
4	62.5	3:42:00 PM

Table 5. Percentage of initial torque loss after one week.

Referring to Table 5, with the exception of bolt 1 due to the residual measuring tool being over-torqued, it is clear for the other three specimens that torque loss ranged from 44%-59%.

Torque Trace.

Five samples of the measured torque trace are shown in Graph 1. Regarding the root cause of torque loss, two observations are remarkable. First, the yield torque or proof load ranged from 90 Nm to 105 Nm. Note that significant yielding occurred at 62.5 Nm. Because of this, increasing the installed torque was not a viable design solution. Second, within the plastic deformation region of the torque traces, the slope was negative. This, along with the hardness data, suggests that the clamped member (die cast aluminum) housing was “soft.”

CONCLUSION

Based on the results of this experimental study, the root cause of torque loss was a result of brinelling at the surface of the die-cast aluminum housing around the bolt holes and localized yielding during installation. Inadvertently, this was caused by the hardness incompatibility between the housing and property class 10 steel nuts, respectively. Potentially, this design shortfall could be resolved by redesigning the joint with a property class 8 stud and nut assembly. The lower yield torque would allow the stud-nut assembly to be tightened to its proof load, which may lessen the degree of loosening. Other proposed design changes are planned for future investigations.

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