An Empirical Study on Torque Retention of High Pressure & High Temperature Joint due to Fastener Material

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Abstract - *All engineering parts are put together by joining one component to another. A vast majority among this is assembled with the fasteners or bolts. In the assembly process, the behavior of a bolted joint depends on a large number of variables that are difficult or impossible to predict and control. Obtaining the desired joint configuration and preload is subjected to a high degree of uncertainty. At high temperature, the joint clamping force may change (due to difference in coefficient of thermal expansion of adjacent materials in the joint), this can adversely affect the joint performance. It is therefore necessary to compensate for operating temperature and pressure conditions, when assembling the joint at room temperature. This article describes about loss of clamping force observed in joint, which is subjected to high temperature & pressure. The objective of this study is to suggest suitable modification in considered joint to avoid preload loss of fastener. The loss of joint clamping force can lead to degraded system performance. In high-temperature joint, adequate clamping force or preload must be maintained to compensate temperature induced dimensional change of the members. The reason of fasteners preload loss is investigated and the joint configuration is modified. The performance of existing and modified joint configurations are evaluated experimentally by thermal-cycling tests.*

Key Words: bolted joint, preload loss, clamping force, torque, gasket, thermal-cycling experiment, thermal load.

1. INTRODUCTION

The interconnecting lines of liquid rocket engine system must retain fluid at operating condition. These lines may have one or more leak proof mechanical joints. These joints are not usually subjected to aerodynamic loading however failure of these joints can be catastrophic. A vast majority among joints consist of three elements: flange, fasteners and gasket or O-ring. This article describes about loss of clamping force, observed in gasket flanged joint during its test. The reason of preload loss is investigated and the joint configuration is modified for the operating condition. The performance of existing and modified joint configurations are evaluated experimentally by thermal-cycling tests. The required preload of fasteners is normally achieved by applying predefined torque using torque wrench. Leak tightness of the joint depends on the gasket contact stress produced by the flanged joint at operating condition. The gasket is compressed due to fasteners preload. The loss of clamping force causes loss of compressive stress in gasket and finally loss of fluid [1]. Preload loss may result due to Vibration, Gasket Creep, temperature expansion differentials or a combination of two or more of these factors. Hence application of correct torque is essential. The stiffness of the gasket is much smaller than that of joint and bolt. It being of lower value dominates the elastic behavior of the joint assembly, hence play major role for finalizing the joint preload or torque [2]. Georgeta Urse consolidated recently published literature on the leak tightness and strength of flange joint [3]. Flange surface roughness influences the joint leak behavior however at relatively high axial force, it is found to be insignificant [4]. Flange joints with two concentric gaskets were investigated to determine the influence of gasket pressure and sealing performance of the gasket on the joint, double gasket joints can withstand higher internal pressure than those with a single gasket [5]. The axial stress variation induced by pre loading between the bolts of the same joints are lower in case of controlled tightening [6]. The tightening of bolts and order in which it is carried out affects both the uniformity of bolt loading as well as its leak tightness. Hence star assembly pattern is recommended by ASME PCC-1 [7]. The bolt load should be increased in three or more consecutive rounds to reach 100% of the prescribed preloading. Guruchannabasavaiah N G addressed the effect of temperature along with internal fluid pressure on gasket seal compression [8]. Differential temperature in the flange changes the initially applied bolt preload thereby changing the contact stress at the gasket. Analysis was carried out for three different bolt force to understand its effect on gasket seal load at working condition.

2. INVESTIGATION OF PRELOAD LOSS

A metallic braided corrugated flexible hose, interfaces with joint. The joint configuration is shown in figure 1, HSHC screws M6x1x20, 6 numbers are used for joint assembly with gasket and washers.

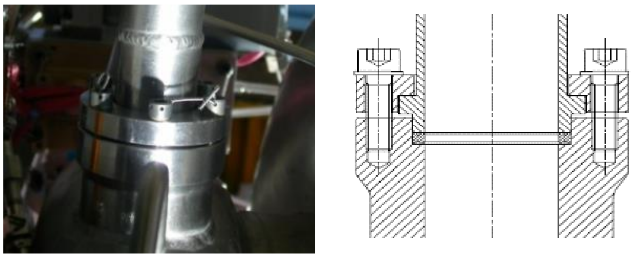

Fig -1: Interface with Gasket Joint

Hot gas at pressure 42 bar abs and temperature 350°C, flows through the joint. The torque of this interface fastener was checked at ambient temperature after the tests. Considering the various uncertainties in the applied torque, relaxation up to ≤ 20% of the applied torque is expected however it was found to be less torque in many instances as shown in table 1, which indicates design inadequacy or preload loss for the joint.

Table -1: Torque relaxation observation during the tests

Assembly Torque: 12 Nm	
Minimum Torque observed after test 1 (17 Sec)	8 Nm
Minimum Torque observed after test 2 (233 Sec)	5 Nm
Minimum Torque observed after test3 (153.7 Sec)	9 Nm

Tests performance were satisfactorily however the reason for torque or preload loss is investigated considering the operating condition of the joint. The joint material configuration is shown in figure 2.

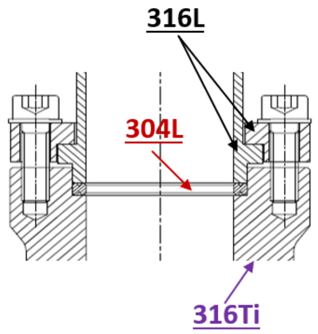

Fig -2: Material configuration of the Joint (with Z30C13 fasteners)

2.1. Minimum Joint Preload Requirement

The metallic gasket of various configurations is used for high temperature & pressure application. The gasket contacts stress must be large to give sufficient gasket deformation to overcome any flange surface imperfection and to overcome the internal pressure force. The ASME VIII gives a more conservative value for bolt load [10]. Considering the operating pressure condition an attempt is made to understand the adequacy of fasteners preload and gasket load.

Initial minimum bolt load required (Wm1) to seat the gasket regardless of internal pressure

$$Wm1 = \Pi bGy \quad (1)$$

Here, b is gasket width, G is effective diameter of the seal and y is seal pressure.

Minimum bolt load required (Wm2) for operating pressure condition

$$Wm2 = \Pi G^2 \frac{P}{4} + 2b\Pi GmP \quad (2)$$

$$2(m-1)^2 * 180 = y \quad (3)$$

Equations 1,2 and 3 are as per ASME VIII, which gives more conservative value for fastener preload. The factor 'm' and 'y' are the 'gasket factor' and 'initial seating stress' (psi) values respectively

y=10100 psi, for Grooved metal [10]
from equation (3), m=4.25
b=gasket width=1mm
G=35.2 mm
P=4.2 Mpa, Operating Pressure

From equation (2)

$$Wm2 = \Pi G^2 \frac{P}{4} + 2b\Pi GmP =$$

4087.18 N + 3947.85 N = 8035 N

Hence minimum fastener preload require is 8035 N (1339.2 N/fastener) at operating condition, including gasket seat load requirement. Thermal load due to operating condition is not considered in above calculation.

2.2. Z30C13 Fastener Nut Factor Evaluation

The fastener preload assessment requires nut factor. Many factors are influencing the torque-tension (load) relationship including material, size, plating, surface finish, thread lubricants, corrosion and wear of fasteners. It is an empirical value that linearly models the rate at which tension or force is developed within a fastener when torque is applied. The fastener preload can be estimated if nut factor is known. Hence nut factor was evaluated experimentally by simulating the joint material, surface finish, surface treatment and lubricant application. The evaluated nut factor is 0.298.

Table -2: Joint material details

Joint Parts	Material	Surface treatment
Hose along with Flange (top)	AISI 316L	Passivation
Adaptor (Bottom)	AISI 321Ti	Passivation
HSHC Screw and washer	Z30C13	Decontamination
Gasket (Plate washer)	AISI 304L	Passivation

2.3. Load Estimation for the Fastener

As per assembly procedure, 12 Nm torque was applied to the fasteners. With the knowledge of nut factor and applied torque, fasteners load and margin on tensile stress is calculated as shown in table 3.

Table -3: Fastener load and design margin on tensile stress

Torque considered for calculation (Nm)	12
Nut Factor (f)	0.298
Preload (N) = Torque/ (Nut factor x Diameter of fasteners)	6711.4
Pressure Load (N) = Pressure x Area $= 42 \times 10^5 \times \pi/4 \times (38/1000)^2/6$	793.5
Preload and Pressure load on each screw (N) = Preload + C x Pressure load Note: C= calculated (conservative) stiffness constant of the joint under study [22] =0.4	7028.8
Tensile stress (MPa) =Total load/ Tensile area (Tensile area=20.1 mm^2)	349.7
Margin on tensile stress (wrt YS@350°C) Yield strength of Z30C13 at 350°C is 686 N/mm^2	0.96

As per above calculation, sufficient margin exists for the fasteners even on yield strength at 350 °C. Hence preload application seems to be correct, however the effect of thermal load on joint during operating condition is not considered in table 3. If the magnitude of thermal load is high enough, it may affect the total load on fastener at operating condition. The coefficient of thermal expansion (α) for joint material is mentioned below for temperature range Room temperature (RT) to 350 °C

Coefficient of thermal expansion (α) for AISI 316L, AISI 316Ti and AISI 304L material: ~17.2e-6 mm/mm/°C

Coefficient of thermal expansion for Z30C13 material: ~11.7e-6 mm/mm/°C

The difference in coefficient of thermal expansion for the joint material configuration causes additional load on the fastener in terms of thermal load

Change in temperature (ΔT) : 350-25=325 °C
Height of flange (L) : 10 mm
Elongation (Δα*ΔT*L) : 0.017 mm
Young's Modulus E for Z30C13 : 218 GPa
Now force requires to elongate the fasteners by 0.017 mm

$$E^* (\Delta L/L) ^* A = 7452 \text{ N}$$

Considering the thermal load 7452 N at operating condition, total load on fastener and design margin is re-estimated as mentioned in table 4.

Table -4: Fastener load and design margin on tensile considering the thermal load at operating condition

Torque considered for calculation (Nm)	12
Nut Factor (f)	0.298
Preload (N) = Torque/ (Nut factor x Diameter of fasteners)	6711.4
Pressure Load (N) = Pressure x Area $= 42 \times 10^5 \times \pi/4x (38/1000)^2/6$	793.5
= Preload + C x Pressure load Note: C= calculated (conservative) stiffness constant of the joint under study [22] =0.4	7028.8
Thermal load (N)	7452
Total Load on each screw (N) =Pre-load and Pressure load + Thermal load	14480.8
Tensile stress (MPa) =Total load/ Tensile area (Tensile area=20.1 mm^2)	720.4
Margin on tensile stress (wrt YS@350°C) Yield strength of Z30C13 at 350°C is 686N/mm^2	-0.05

The flange is stiffer than the fastener hence additional thermal load 7452 N is experienced by the fastener in addition to the tensile load due to preload and pressure load. This causes yielding of the fastener. If the material settles, even just a few micrometers, the stretching of the bolt will lead to a loss of preload. Hence when the flange contracts after engine shutdown, the permanent set in the fastener will result in torque relaxation.

3. AMENDMENT IN JOINT CONFIGURATION

It can be inferred from the previous deliberation that the total load on Z30C13 fastener is more in operating condition. Fastener preload loss can be avoided by reducing torque application however it also calls for revisiting the compressive load requirement of gasket. This inadequacy in joint design can also be avoided by choosing joint material configuration with least difference in coefficient of thermal expansion. This may result in negligible thermal load at operating condition. The option of negligible thermal load at operating condition was further explored.

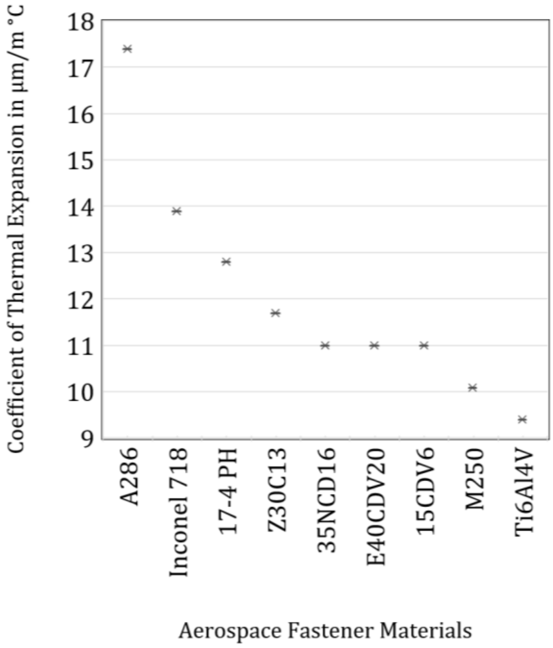

Chart -1: Fastener's coefficient of thermal expansion in operating temperature range of Joint

On literature study [23,24], it is found that A286 material is having similar coefficient of thermal (17.4e-6 mm/mm/°C) as the flange material, A286 fastener is also commercially available. Hence joint configuration amendment is proposed to use A286 fastener instead of Z30C13 fastener.

3.1 A286 Fastener Nut Factor evaluation

Assessment of proposed configuration joint requires nut factor with A286 fastener. The Nut factor was evaluated experimentally by simulating the joint material, surface finish, surface treatment and lubricant application. The surface treatment for A286 fastener is decontamination. The evaluated nut factor is 0.221.

3.2 A286 Fastener Joint Configuration Load Estimation

The thermal load on the fastener at operating condition is estimated. The Coefficient of thermal expansion (α) of A286 material is 17.4 e-6 mm/mm/°C. The difference in coefficient of thermal expansion for the joint material configuration causes additional load on the fastener in terms of thermal load

Change in temperature (ΔT) :350-25=325 °C
Height of flange (L) : 10 mm
Elongation (Δα*ΔT*L) : 0.0006 mm
Young's Modulus E for A286 : 208 GPa
Now force requires to elongate the fasteners by 0.0006 mm

$$E^* (\Delta L/L) ^*A = 251 \text{ N}$$

Hence fasteners will be subjected to additional thermal load 251 N at operating condition. Considering the thermal load, total load on the fastener is mentioned in table 5.

Table -5: Fastener load and design margin on tensile by considering the thermal load at working condition

Torque considered for calculation (Nm)	8
Nut Factor (f)	0.221
Preload (N) = Torque/ (Nut factor x Diameter of fasteners)	6033.2
Pressure Load (N) = Pressure x Area = 42 x 10^5 x π/4x $(38/1000)^2$/6	793.5
Preload and Pressure load on each screw (N) = Preload + C x Pressure load Note: C= calculated (conservative) stiffness constant of the joint under study [22] =0.4	6350.6
Thermal load (N)	251
Total Load on each screw (N) =Pre-load and Pressure load + Thermal load	6601.6
Tensile stress (MPa) =Total load/ Tensile area (Tensile area=20.1 mm^2)	328.4
Margin on tensile stress (wrt YS@350°C) Yield strength of A286 at 350°C is 620 N/mm^2	0.9

Hence positive margin is available on yield strength for proposed A286 fastener at operating condition for 8 Nm torque application. The torque application 8 Nm is chosen to retain approximately same fastener preload as that of used in in Z30C13 joint configuration.

4. THERMAL CYCLING EXPERIMENT

Obtaining the desired preload & leak proof joint at operating condition is subjected to an uncertainty. Hence to ensure the joint performance, thermal cycling experiment is carried out to validate the design. During engine operation (Duration: 143s), hot combustion product gas is flowing through the joint at 42 bar abs. The maximum temperature measured on screw head during the operation is 345°C approximately, which was measured during one of the ground acceptance hot test. Considering the difficulty associated with the thermal cycling test of joint in flow condition, the test is carried out in no flow condition. However, the required operating condition is achieved by placing the test article in furnace.

The required temperature during the test is maintained using the temperature controller. A set of test article simulating the interface is fabricated. The part no.1 represents the engine interface, part no. 2 & 3 represents the hose interface as shown in figure 3. The exploded view of parts is shown in figure 4.

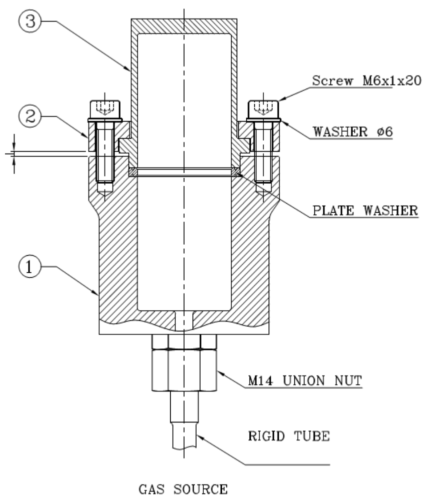

Fig -3: Test article simulating interface

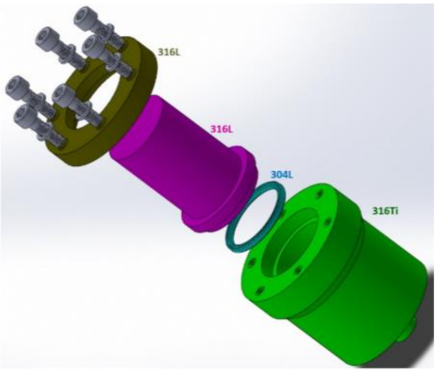

Fig -4: Exploded Joint details

4.1 Test setup description and measurements

The operating condition during the test is ensured by two pressure and three temperature measurements. Pressure is measured outside the furnace before the manual isolation valve as shown in figure 5. Temperature measurement thermocouples are directly attached to the test article.

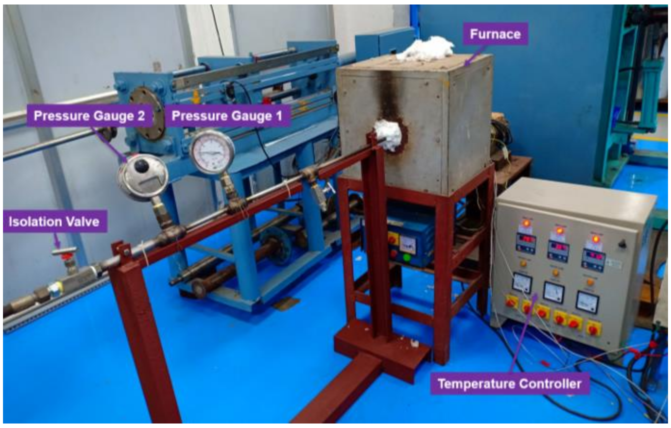

Fig -5: Thermal Cycling Test setup

Two temperature measurements are attached on the fasteners, diametrically opposite to each other and one temperature measurement on part no. 3 top surface as shown in figure 6. The required operating temperature is achieved by placing the test article in furnace. Using the temperature controller, furnace temperature is controlled, such that required operating temperature is achieved for the fasteners. The test article is initially kept at lesser pressure and isolated from the supply pressure source. Initial pressure requirement is finalized based on trails so that required operating pressure is achieved at operating temperature condition. The test setup is shown in figure 5.

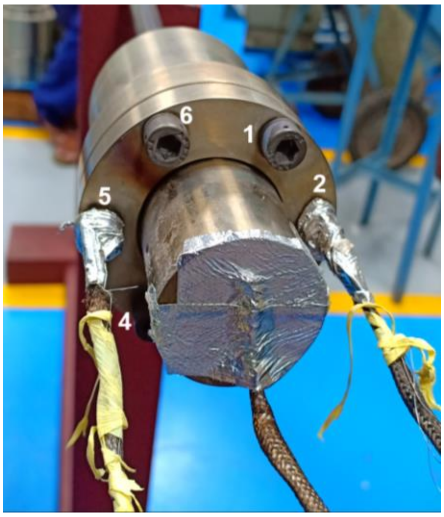

Fig -6: Temperature Measurement on test article

4.2 Thermal cycling test

The test is performed with existing Z30C13 fasteners and proposed A286 fasteners. The test is carried out for minimum 5 minutes duration after achieving the operating condition and stabilization.

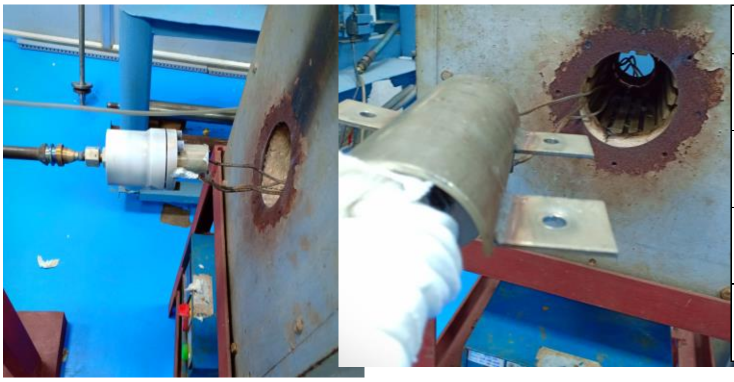

Fig -4: Test Article with Furnace

The joint leakage is assessed by monitoring the pressure gauges during the test. The torque relaxation is measured after bringing the test article to room temperature. The details of Thermal-cycling carried out is mentioned in table 6.

Table -6: Thermal-cycling test matrix

Thermo Cycling test configurations	Nos. of test carried out
A286 fasteners and test article-1	6
Z30C13 fasteners and test article-1	4
Z30C13 fasteners and test article-2	1

5. RESULTS AND DISCUSSION

As mentioned earlier the performance of joint configuration is assessed by following:

- ✓ By monitoring the test article pressure during the test. No pressure drop will ensure the leak tightness of the joint.
- ✓ Post thermal cycling test torque verification of the fasteners at room temperature. Retention of applied torque ensure that fasteners preload is retained in operating condition also.

The test article-1 assembly was done with A-286 fasteners and it was subjected to 6 series of thermal cycling test. The joint performance in terms of leak tightness and fasteners preload retention was satisfactory as mentioned in table 7.

Table -7: Thermal-cycling test on test article-1 with A286 fasteners

Tests	Test condition	Observation/Remarks
First	Pressure: 42 Bar abs Temperature: 350^{+5} °C Hold Duration: 5 minutes (after stabilization)	No Pressure Drop No Torque relaxation
Second		No Pressure Drop No Torque relaxation
Third		No Pressure Drop No Torque relaxation
Fourth		No Pressure Drop No Torque relaxation
Fifth		No Pressure Drop No Torque relaxation
Sixth		No Pressure Drop No Torque relaxation

Subsequently test article-1 parts were disassembled, visually inspected and cleaned. These parts are assembled with Z30C13 fasteners and it was subjected to 4 series of thermal cycling test. The joint performance in terms of leak tightness was satisfactory however few fasteners found rotating during torque verification at 12 Nm as mentioned in table 8.

Table -8: Thermal-cycling test on test article-1 with Z30C13 fasteners

Tests	Test condition	Observation/Remarks
First	Pressure: 42 Bar abs Temperature: 350^{+5} °C Hold Duration: 5 minutes (after stabilization)	No Pressure Drop No Torque relaxation during check up to 11 Nm. Fasteners 1,4 & 5 Found rotating while torque verification at 12 Nm
Second		No Pressure Drop No Torque relaxation during check up to 11 Nm. Fastener 5 Found rotating while torque verification at 12 Nm
Third		No Pressure Drop No Torque relaxation during check up to 11 Nm. Fastener 5 Found rotating while torque verification at 12 Nm
Fourth		No Pressure Drop No Torque relaxation during check up to 11 Nm. Fastener 5 Found rotating while torque verification at 12 Nm

The test article-2 assembly was done with fresh Z30C13 fasteners and it was subjected to 1 series of thermal cycling test. This is to ensure that torque relaxation observation is not hardware specific. The joint performance in terms of leak tightness was satisfactory however few fasteners found rotating at torque verification at 11 Nm as mentioned in table 9.

Table -9: Thermal-cycling test on test article-2 with fresh Z30C13 fasteners

Tests	Test condition	Observation/Remarks
First	Pressure: 42 Bar abs Temperature: 350^{+5} °C Hold Duration: 5 minutes (after stabilization)	No Pressure Drop No Torque relaxation during check up to 10 Nm. Fasteners 1 and 6 Found rotating while torque verification at 11 Nm

No pressure drop is observed during the Thermal Cycling test of Z30C13 & A-286 fasteners configuration hence applied pre-load on fastener or gasket is meeting the joint requirement.

The torque relaxation is observed for the few Z30C13 fasteners out of total six during Thermal Cycling test. Flange is the stiffer than the fastener hence additional thermal load is experienced by the fasteners in addition to the tensile load due to preload and pressure (conservative). This results in yielding of the fasteners. The Thermal load will get relieved once the fastener yields. When the flange contract after the test, the permanent set in the fastener will result in the torque relaxation.

6. CONCLUSIONS

The operating pressure and temperature condition is simulated during the Thermo Cycling test however it is not possible to simulate the vibration condition caused by assembled system during its operation. This may be probable reason for higher torque relaxation is observed during assembled system tests.

The magnitude of thermal load depends on joint material combination hence should be considered for selecting interface and fasteners material.

Usage of Z30C13 fastener causes additional thermal load 7452 N due to interface material combination at operating condition (for applied torque 12 Nm). This results reduction in available design margin.

No torque relaxation is observed during Thermo Cycling test of A286 fastener configuration. The resulting thermal load at operating condition is also negligible hence better option for joint interface under study. The coefficient of thermal expansion for A286 fastener is approximately same as interface material combination which results less thermal load 251 N.

Scope of the experiment done is limited to assessment on impact of A286 fastener induction for joint under study. More detailed experiments can be designed by observing the joint clamping force with the variation in the operating parameters like temperature, torque, pressure etc for understanding the sensitivity of each parameter and for better understanding of preload loss phenomenon of the joint.

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