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Jumper Connector Analysis

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Date Published
February 1994

To Be Presented at
1994 ASME PVP Conference
Minneapolis, Minnesota
June 19-23, 1994

Prepared for the U.S. Department of Energy
Office of Environmental Restoration
and Waste Management

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P.O. Box 1970
Richland, Washington 99352

Hanford Operations and Engineering Contractor for the
U.S. Department of Energy under Contract DE-AC06-87RL10930

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DISCLM-2.CHP (1-91)

JUMPER CONNECTOR ANALYSIS

2-INCH BY 2-INCH CONNECTOR

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M. R. Lindquist
Westinghouse Hanford Company

ABSTRACT

Jumper connectors are used for remotely connecting pipe lines containing transfer fluids ranging from hazardous chemicals to other nonhazardous liquids. The jumper connector assembly comprises hooks, hookpins, a block, a nozzle, an operating screw, and a nut. The hooks are tightened against the nozzle flanges by the operating screw that is tightened with a remotely connected torque wrench. Stress analysis for the jumper connector assembly (used extensively on the U.S. Department of Energy's Hanford Site, near Richland, Washington) is performed by using hand calculation and finite-element techniques to determine the stress levels resulting from operating and seismic loads on components of the assembly. The analysis addresses loading conditions such as prestress, seismic, operating, thermal, and leakage. The preload torque-generated forces at which each component reaches its stress limits are presented in a tabulated format. Allowable operating loads for the jumper assembly are provided to prevent leakage of the assembly during operating cycles.

INTRODUCTION

Jumper connectors are used for remotely connecting jumper pipe lines in radioactively contaminated zones. The jumper pipes transport radioactive fluids and hazardous chemicals. This analysis was made for a particular PUREX type connector used extensively in the Hanford Site, managed by the U.S. Department of Energy, Richland Operations Office (DOE-RL), Richland, Washington. The design analysis of the jumper connector is performed to achieve the following objectives.

- Determine the maximum amount of preload that can be applied to a 2- by 2-in. jumper connector at the time of installation. (Relate preload to operating screw torque.)
- Determine the maximum allowable load capacity for a 2- by 2-in. jumper connector.
- Investigate the possibility of leakage from the connector resulting from a seismic event and/or the application of the operating loads.

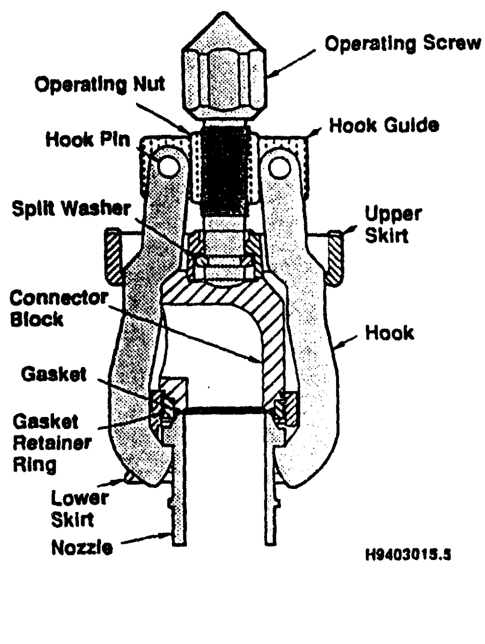
DESCRIPTION OF THE EQUIPMENT

A sketch of the jumper connector assembly is shown in Figure 1. The assembly mainly consists of three hooks, a block, an operating screw, a upper and a lower skirt, and the gasket retainer ring.

The installation and operation of the jumper is described below.

The jumper piping is welded to the block. The assembly is then lifted and placed so that the gasket is mounted over the nozzle and the hooks engage the flange of the nozzle. Once in position, the operating screw is torqued by use of a remotely operated torque wrench that is activated for a predetermined period of time. The torque wrench is rated for a specific maximum torque. The torque is balanced by a compression force on the block transmitted through the screw and by a tensile force in the hooks. These two equal and opposite forces hold the jumper connector assembly together. In other words, the higher the torque, the greater the clamping force.

Figure 1. Jumper Connector Assembly



DESIGN PARAMETERS AND ALLOWABLE

- Design pressure is 400 lbf/in^2.
- Design temperature is 240 °F.
- Torque versus force relationship is obtained from the test results.

The allowable stresses for the hooks and hook pins are as follows:

Allowable stresses in bending = 0.60 S_y
Allowable stresses in shear = 0.40 S_y

where S_y is the yield strength of the material.

These allowable stress criteria comply with the American Institute of Steel Construction (AISC), Ninth Edition (AISC 1989).

The allowable for primary membrane and bending stresses for the nozzle and the block material are as per American Society of Mechanical Engineers (ASME) Section III, Sub-Section NB-3221.3 design criteria (ASME 1989a).

The allowable material stresses for tension at the design temperature for the gasket retainer ring material are taken from ASME B31.3-1990 edition (ASME 1990). The shear stresses for the above materials are limited to 0.4 S_y.

MATERIAL PROPERTIES

The room temperature material properties of different component parts are shown in Table 1.

Table 1. Room Temperature Material Properties.

Component	Material	Yield strength (klbf/in^2)	Ultimate tensile strength (klbf/in^2)	% Elongation
Hook	ASTM A 747, Grades CB7Cu-1 or CB7Cu-2	115	145	9
Hook Pins	ASTM A 108 Grade 1045 (Cold drawn)	77	91	12
Nozzle	ASTM A 743 Grade CD-4MCu	70	100	16
Block	ASTM A 276 Grade 304L SST	30	75	40
Operating nut	ASTM A 148	60	90	20
Operating screw	ASTM A 193 Grade B7 C.S.	100	125	16
Gasket retainer ring	ASTM A 276 Grade 304L SST	30	75	40
Gasket	Teflon	-	-	-

LOAD CASES

Analyses for the jumper connector is performed to evaluate stresses for the following loading conditions.

<u>Preload</u>
Preload is because of loads generated when the operating screw is torqued. Several standard formulas are available to calculate the torque versus force relationship. The results vary widely, and arriving at a consensus is difficult. The load range considered for prestress analysis is 10 to 48 Kips pound force (klbf), which is approximately in the operating screw torque range of 200 to 1,000 foot pound force (ft·lbf).

<u>Seismic Loading</u>
When installed and connected to a nozzle, the assembled parts behave like a rigid mass. Two factors may influence the rigid behavior: (1) the reduction of preload torque from seismic motion and (2) the seismic lateral force generated at the joints where different parts contact each other (e.g., the tip of the hook and the nozzle flange), which should be less than the friction force acting at that joint. The fundamental frequency of the combined model (Figure 2) was determined to be 166 Hz, which indicates rigid behavior with no dynamic amplification. The connector will see insignificant seismic forces and will maintain its integrity during a seismic event.

<u>Operating Loads</u>
The operating loads are developed because of pressure, dead load, differential thermal expansion and seismic motions of the jumper piping. The piping analysis evaluates dead loads, thermal loads, and seismic loads; these loads constitute the reactions at the jumper connector point. A study of the load data sheet from jumper piping stress analysis, suggests that an enveloped load of 400 lbf in each direction and a moment of 350 ft·lbf in two directions is sufficient to cover the piping demands for a 2- by 2-in. jumper connector. These enveloped loads are considered the extreme operating loads for 2-in-diameter jumper pipe lines. The coefficient of friction between the Teflon[1] gasket and the steel is small; therefore, the block may rotate over the gasket surface because of the moment about the nozzle and block axis. Hence, the moment along the nozzle axis is released.

<u>Thermal Loads</u>
The design temperature is 240 °F. The room temperature material properties do not appreciably change at this low temperature. No time-dependent thermal analysis is performed on this jumper connector; instead, calculations are made with a temperature differential of 240 and 70 °F to determine the relative thermal displacement of the nozzle flange and top of the block. The resultant thermal movement is converted to an equivalent force that considers the stiffness of the assembly at the center point of the nut. This thermal-induced force is superimposed on the torque loads to find the combined stresses in the hook and the nozzle flange.

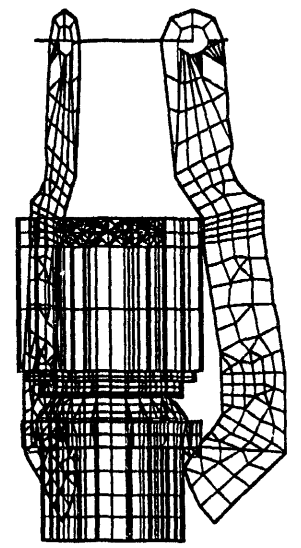
<u>Leakage</u>
The block is modelled with 400-lbf/in^2 internal pressure and with the gasket at the bottom. The torque preload is applied as pressure on the equivalent area at the top of the block and operating loads are applied at proper locations on the sides. The criteria for no leakage are (1) that the reactions at the sealing surface must be compressive and (2) that the net reactive forces normal to the compressive force should be less than the reactive friction forces, or a mechanism should exist to transfer these forces to the fixed nozzle base. The coefficient of friction between the steel block and the Teflon gasket is small (0.04) (Baumeister 1978). The friction force at the interface of the block and Teflon gasket is not enough to restrain the reactive forces. These forces are transferred to the fixed nozzle base through the lower skirt and the gasket retainer ring. The compressive stresses at the point of contact between the ring and the nozzle face are calculated and found acceptable. The bending stresses at the fixed end of the nozzle body are also acceptable. The torque loading that satisfies the above criteria is noted and shown in Figure 3.

[1]Teflon is a registered trade mark of E. I. du Pont de Nemours & Company, Wilmington, Delaware.

The minimum torque to stop leakage has a safety factor of 2.5 and is specified in the results of this analysis.

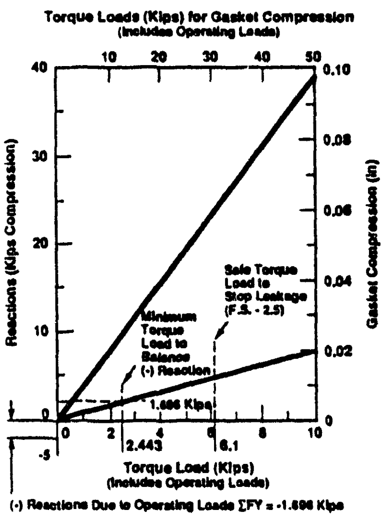
The leakage analysis is based on the assumption that forces are distributed uniformly and that moments are distributed proportionally and linearly on the sealing surface. The leakage analysis does not account for such things as the effects of fluid characteristics, uneven or discontinuous loading on the sealing surface, the condition of the sealing surface, or other variable factors that could affect sealing performance.

Figure 2. Combined Model



Solid Modelling of Nozzle
- Thermal Displacement Study

Figure 3. Torque vs. Compression of Gasket. (Leakage Study)



Gasket Retainer Ring (supplementary load path)

The gasket retainer ring is screwed to the lower skirt and contacts the block. The tip of the gasket retainer ring touches the nozzle face. The top of the retainer ring has a clearance of 0.033 in. from one edge of the block face. A load path may be established to distribute the compression load from the block to the nozzle face through the lower skirt and the retainer ring. A two-dimensional (2D) computer model is generated with the block and the retainer ring to evaluate the stresses in the retainer ring. The gasket is used as a spring element in this case.

DESIGN ASSUMPTIONS

The jumper connector presents a complex analytical problem having a multitude of boundary conditions. The following assumptions are made to evaluate the jumper connector.

- The connection between the hooks and the nozzle flange is fixed in translation. This fixity is based on the hook tension force generated by torquing the operating nut. The fixity in the other two directions is based on the friction force between the two surfaces.
- The hooks are connected to the hook pins by a rigid link to transmit the torque force and are guided in two other directions by their coupling with the operating nut. This assumption is valid because the tension of the hook is generated by the upward movement of the operating screw, which functions as a guide and a rigid body to transfer the tension forces.
- The operating screw itself is not modeled in the finite-element analysis; instead, the torque generated forces are represented by a tension force placed at the center of the operating nut, with the same compressive force or equivalent pressure on top of the block. This assumption has no effect on the results of the analysis.
- The connection between the Teflon gasket and the nozzle is assumed to be fixed in translation on the basis of the same arguments as stated in the first assumption above.
- The analysis assumes perfect fit-up conditions where each hook sees an equal amount of tension load. An imperfect fit-up is beyond the scope of this analysis.
- No thermal analysis is performed for the wetted surface. However, thermal expansion of the nozzle and block is considered, and stresses in the jumper connector components and the nozzle flange are increased to account for the relative thermal movements. Because the upper-bound operating temperature is below 400 °F, material properties do not change significantly. Thermal-induced stresses are self-limiting and have a high allowable.
- The nozzle base is considered fixed in all directions.

COMPUTER MODELING

The ANSYS version 4.4 computer code (DeSalvo 1987) is used to perform a finite-element analysis on the jumper connector assembly. The following models are made for the analysis.

- A combined model of all the components, (see Figure 2). The purpose of this model is to find the approximate stress levels in the various components that result from static prestress loads and also to make a dynamic run to find the natural frequency of this equipment. All hook analysis are performed based on the results of this model.

- A model for the nozzle. The contact area between hook and nozzle flange for one hook position is closely meshed for a 60-deg segment (see Figures 4 and 5). The hook tension loads are distributed uniformly on the contact nodes. The flange stresses obtained from this model are used for qualifying the nozzle flange.

- A model for the block (Figure 6). This model has Teflon gaskets attached to the block and is restrained at the base with translational restraint. This model is used to (1) perform a stress analysis of the block, (2) determine the critical combination of the operating loads, (3) perform the leakage study, and (4) determine the compressibility of the gasket.

- A 2D model of the block and the retainer ring. The gasket is modeled as a spring element with proper stiffness obtained from the three-dimensional (3D) model. The preload torque-induced compression loads were distributed as pressure at the top of the block. The retainer ring is coupled with the block at proper locations in translation only. The purpose of this model is to evaluate the stress level of the retainer ring should a load path be established through the lower skirt and the retainer ring.

- A 3D model of the block and nozzle. A spring element of equivalent stiffness to that of the operating nut and hooks is attached at the top of the block. This model is created to evaluate the relative thermal growth of the hook when free thermal displacement is restricted by the hooks.

ANALYSIS OF THE COMPONENTS

The stresses in the finite elements, except for the nozzle and block, are evaluated using the distortion-energy theory. The condition of yielding based on the distortion energy theory is

$$[\delta y = \frac{1}{\sqrt{2}} \sqrt{[(\delta 1 - \delta 2)^2 + (\delta 2 - \delta 3)^2 + (\delta 1 - \delta 3)^2]}$$

where

δ = Yield stress of the material, and

δ_1, δ_2 and δ_3 = Principal stresses in three orthogonal directions.

For the block and nozzle where ASME allowable are used, the stresses are compared to allowable stress intensity that is based on maximum shear stress theory, and is conservative compared to the distortion energy theory.

HOOK ANALYSIS

Description of the Model

All three hooks are modeled and attached to the nozzle flange by coupling translational degrees of freedom at the interfacing nodes. At the top, each hook is coupled to the operating nut in two directions normal to the hook to represent a guide and also connected with the hook pin by a rigid link (see Figure 7) to transfer tension forces. All hooks are modeled together with the nozzle and the block. Preload torque-generated tension forces are input at the center line of the operating nut and distributed to all three hooks by modeling a rigid link network. For the thermal displacement study, a computer run is made to find the stiffness of the assembly at the center of the operating nut. The relative thermal displacement is converted to preload force based on the stiffness. Analysis is performed with a combined loading of torque and thermal-generated preloads.

Analysis Results

The analysis results show stress concentrations at different parts of the hook because of curvature effects and also because of a small contact area. Stress contours for a preload torque force of 48 klbf is shown in Figures 8 and 9. The tip of the hooks, which contacts the nozzle flange, shows high concentration of stresses. This local concentration of stresses is partly because of the coupling effect between the hook and flange. At this location, only two nodes are coupled to transfer the forces, while the actual distribution is over a contact area. Because the stress concentrations are local and compressive in nature, they are not expected to have any adverse effects on the structural integrity of this component, and are not assessed further within the scope of this analysis. The next highly stressed area occurs where the profile takes an abrupt turn to form the hook. The average principal stresses at this region are considered as the design stresses the hook will see during the preload torquing process. The results are summarized in Figure 10.

Figure 4. Part Model of Nozzle

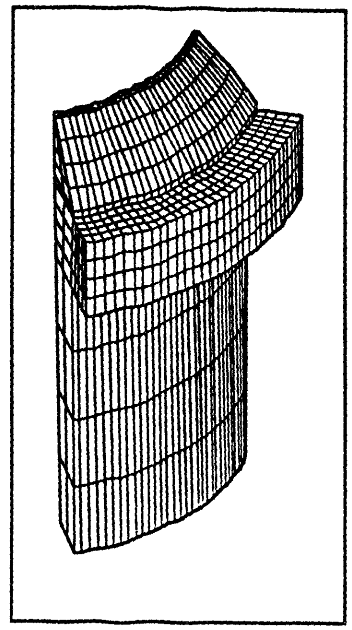


Figure 5. Hook Nozzle Flange Contact Area.

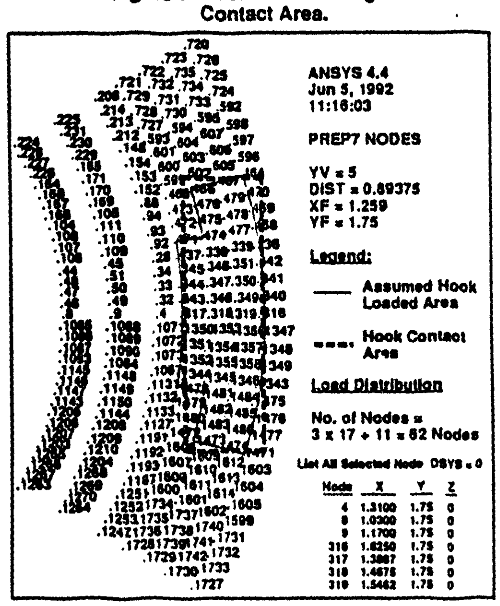


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Figure 6. Block Model

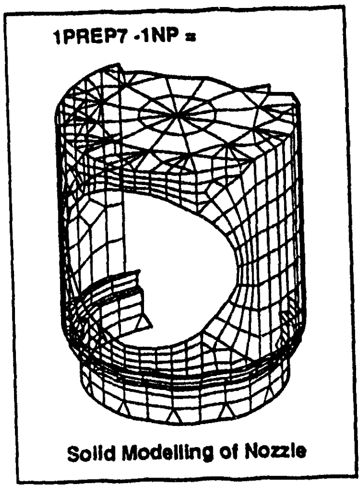

Figure 7. Hook Top Model

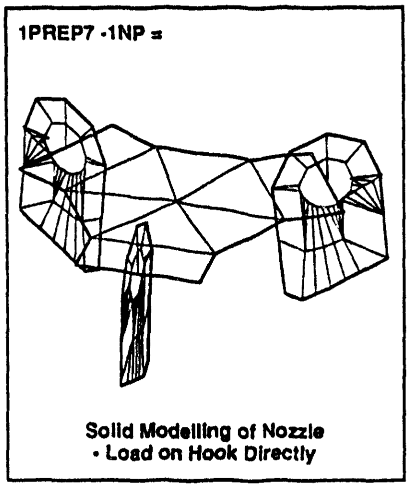

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Figure 8. Hook (Middle Portion) Stress Contour Plot. Torque Load 48 Kips.

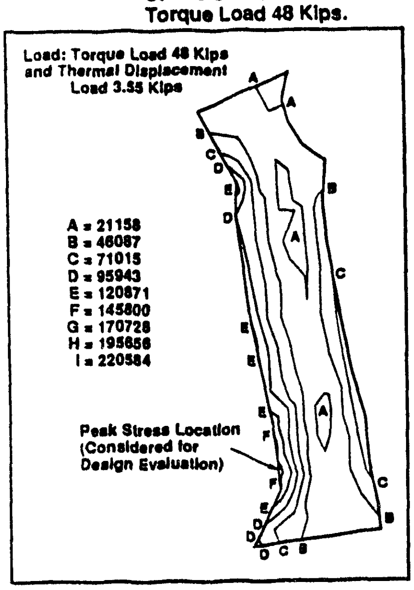


Figure 9. Hook (Bottom Portion) Stress Contour Plot. Torque Load 48 Kips.

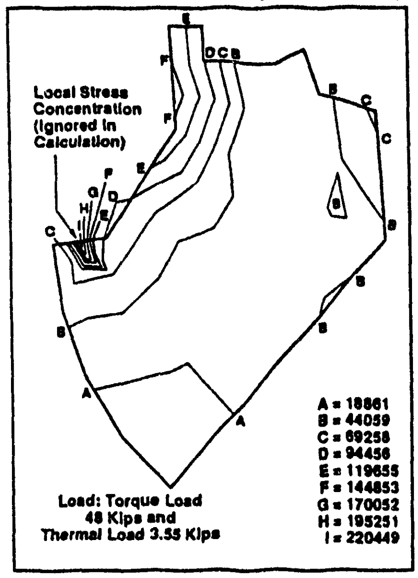
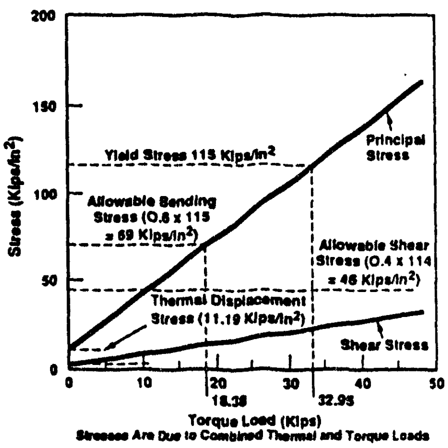


Figure 10. Hook Stress vs. Torque Material ASTM A 747 Grade CB7CU-1 Ultimate Tensile Strength = 145 Kips/in²



BLOCK ANALYSIS

Description of the Model

A detailed analysis of the block is made using a model consisting of shell elements (see Figure 6). The Teflon gasket is modeled with the block. A 2D model of the block and retainer ring is also made to evaluate the gasket retainer ring, should a load path be established through the lower skirt and the retainer ring that contacts the nozzle face at one end. A separate model of the block and nozzle is made to evaluate the thermal-induced stresses in the block. The block models are used to perform the following evaluation:

- Determine the stresses developed in the block
- Study the reactions at the joint between the block and nozzle and to determine the least torque load required to stop leakage under pressure and operating loads
- Find the possible worst-case combinations of applied forces and moments that produce the maximum stresses in the block material
- Determine the worst-case load combination to provide a leak path
- Determine the compressibility of the gasket
- Perform a complete analysis of the block with the combined preload torque-generated loads and the worst-case combination of the operating loads
- Compare the reactions at the assumed one-directional lateral supports with the friction forces and check for possible rotation of the block
- Find the axial thermal displacement of the block.

Analysis Results

Block Stresses. The worst stresses are at the top of the block where the operating screw seat sees compression while the bottom face sees maximum tension because of bending in two directions. The stress contour (Figure 11) shows some concentration around the location of the operating screw, but in general, the stress pattern is smooth and concentric. Stress concentrations developed around the hole (Figure 12) with or without operating loads. This part of the block is reinforced with full penetration welds for connecting jumper piping. The stress concentrations will partially dissipate once the piping is attached. Other than this concentration, all stresses in the block walls are within yield stress levels. In addition, stresses induced in the block because of restrained thermal growth are evaluated. The restrained thermal growth stresses are relatively low and are additive to the preload stresses. Although temperature dependent stresses resulting from thermal expansion are normally classified as secondary stresses, they are conservatively accounted for in this evaluation as additional preload generated stresses. Thermal peak stresses are not evaluated because the jumpers typically do not experience a large number of severe thermal transient conditions. For block torque versus stress intensity see Figure 13.

Operating Loads and Combinations. Operating loads produce maximum stresses on the elements located at both the edge of the opening and at the top of the block above the opening. The stresses around the opening for jumper piping will partially disappear after the jumper pipe is welded to the block. When operating loads are combined with the torque loads, the maximum stress occurs on the inside face of the elements located at the contact surface of the screw and the block. Stresses due to torque loads are critical. Hence, the controlling stresses for the block analysis are limited to those elements located below the operating screw. To find a load combination that produces critical stress in the block, an element that sees maximum stresses because of torque load is chosen. The combination of the operating loads that produce maximum stresses on the chosen element is considered critical. To find the worst combination, unidirectional loads are combined in all possible ways, starting with two variables and adding the third variable with the worst

Figure 11. Top of Block Principal Stress Contours Torque Load 48 Kips and Operating Loads.

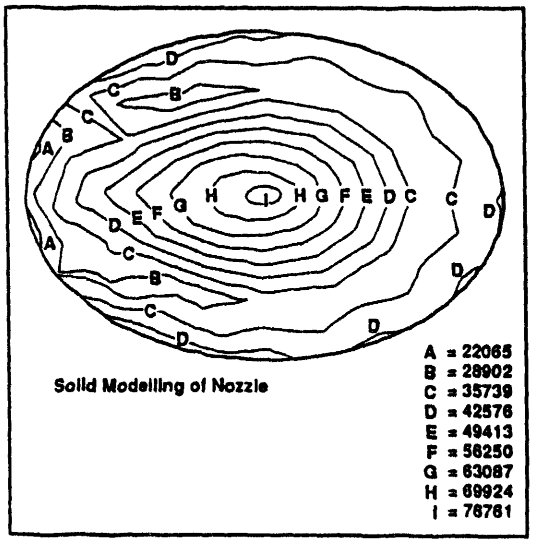


Figure 12. Principal Stress Contours Side of Block (Looking Toward Hole).

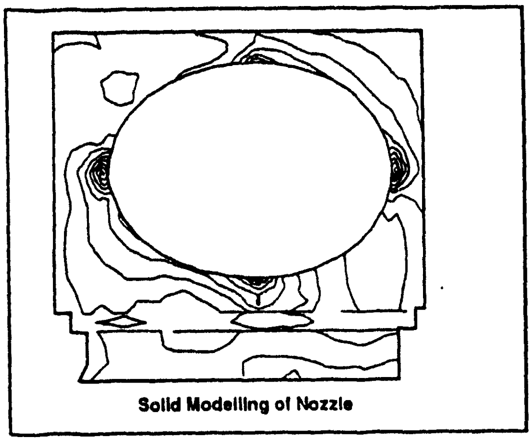
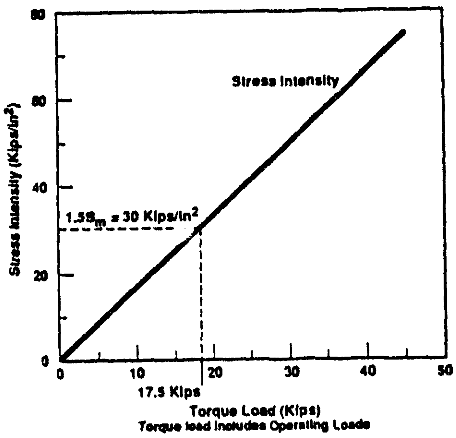


Figure 13. Block Torque vs. Stress Intensity Material ASTM A 240 GR 304L Ultimate Tensile Strength = 75 Kips/in² Yield Strength = 30 Kips/in²



combination of the first two variables until all forces and moments are enveloped.

For leakage consideration, the operating forces are combined so that maximum tension is developed at the gasket nozzle interface. The critical load combinations are as follows.

(1) Fx + Fy + Fz - Mx + Mz

(2) Fx + Fy - Fz - Mx + Mz

where

Fx = Fy = Fz = 400 lbf
Mx = Mz = 350 ft.lbf.

X direction is along the jumper pipe line and positive toward the jumper, Y direction is along the nozzle and block axis and positive toward the block top. All clockwise moments are positive.

The load combination (1) is critical for stresses, and load combination (2) is critical for leakage.

Load Path Determination. The preload torque generated by the operating screw acts on the top of the block and compresses the gasket. The retainer ring has a small area of contact with the nozzle face. A load path may be established through the lower skirt and gasket ring to the nozzle face. A 2D block model is prepared to study the behavior of load transfer through the gasket ring. The retainer ring becomes highly over-stressed because of its cantilever configuration and eccentric load path. As the retainer ring becomes overstressed, the compressive load shifts to bear on the gasket through the block/gasket interface. This concludes that neither the lower skirt nor the retainer ring participates in the direct load transfer mechanism.

Rotation of the Block. Because of the opening on one side of the block face, the compressive forces from the preload torque are not distributed uniformly over the gasket/block interface. One side of the gasket compresses more than the opposite side. This unequal distribution caused the block to rotate, releasing compression on one side and increasing compression on the other. The block also tends to rotate because of operating loads. The static friction on the contact surface between the operating screw head and the block top is assumed to act to prevent this rotation. Two supports, one in each direction perpendicular to the axis of the connector, are assumed at the top edge of the block to simulate the friction force effects. Thus, the loads are distributed uniformly. The reactions acting in a plane normal to the connector axis at the gasket level are primarily from pressure and the operating loads and remain practically constant throughout the torquing process. These reactions are transferred to the nozzle base through the gasket retainer ring and the lower skirt. The stresses developed on the retainer ring and nozzle for these loads are checked and found acceptable.

NOZZLE ANALYSIS

Description of the Model

A finer mesh model is prepared with a 60-deg segment (Figure 4) and the hook reaction was distributed to 62 closely spaced nodes covering one hook contact area (Figure 5). Proper boundary conditions are imposed to represent the continuity at the sides.

Analysis Results

The critical part of the nozzle is the flange. The pressure and operating loads had negligible effects on the nozzle stresses and are not addressed for the nozzle analysis. The 60-deg segmented model is used for the flange analysis. The maximum stress intensity for the elements at the bottom of the flange level for different torque loading with constant thermal preload are plotted in Figure 14. For principal stress plots refer to Figures 15 and 16.

Figure 14. Nozzle Flange Stress Intensity vs. Torque
Material: ASTM A 743 Grade CD4MCU
Ultimate Tensile Strength = 100 Kips/in²
Yield Strength = 115 Kips/in²

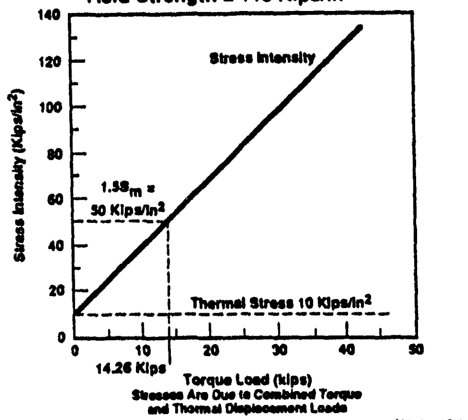
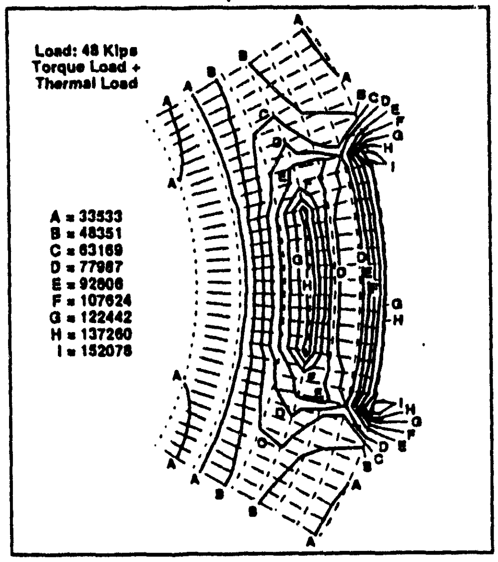


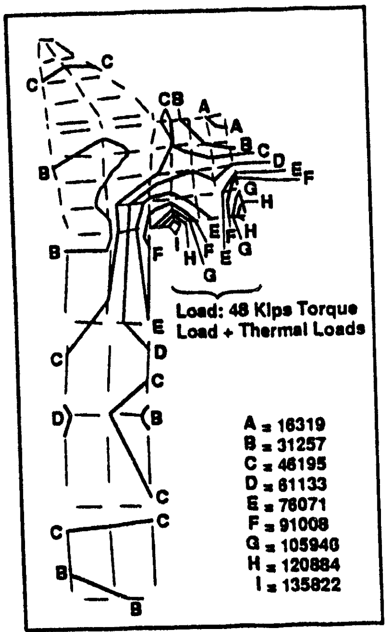
Figure 15. Plan of Nozzle Flange Hook Contact Surface Principal Stress Contours.



OPERATING BOLT AND NUT ANALYSIS

Hand calculations are made to qualify the operating bolt and nut. Internal and external threads are checked for shear stress per ANSI B1.1 (ANSI 1982). The threads for both the screw and nuts are found good for torque-generated loads to 48 klbf. The screw is checked for combined compression and shear loads. The operating nut is also checked for bending stresses for the overhung portion. Both of the parts are acceptable for a torque force of 48 klbf.

Figure 16. Nozzle Section Along X-Axis, Principal Stress Contour Plot.



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HOOK PIN ANALYSIS

The hook pin is considered as a simple beam spanning the two arms of the operating nut. The hook attachment is represented as a uniform loading at the center region of the pin. The pin is checked for bending and shear stresses. Bending stresses are found to control the design. The allowable bending stress is reached at a torque-generated force of 14.77 klbf at design temperature.

RESULTS

The results of the above analysis are tabulated in Table 2. The table indicates the preload torque-generated forces at which the component reaches the allowable stress limits. Thermal expansion of the block and nozzle causes a constant preload to the hooks, hook pins, and nozzle flanges. Two columns for preload torque-generated forces are shown on the table. One column shows the mechanical loads (i.e., loads from torquing of the screw only) at which the component reaches its allowable limits; the other column shows the allowable preload caused by torque considering thermal effects (i.e., thermal loads induced in the component from restricted thermal growth). In the case of the block, the preload torque-generated forces also include the operating forces.

DISCUSSION

The relative thermal displacement between the bottom face of the nozzle flange and the top of the block caused a constant preload of 3.55 klbf. This thermal load is added to the preload torquing forces and induced stresses on the above components. These components are evaluated on the basis of the combined thermal and mechanical loadings. Stresses resulting from operating loads are small compared to the preload torque stresses. Stresses generated by pressure loads are also small. The nozzle flange is stressed by the clamping forces of the hook. The highly stressed part of the flange do not experience the stresses from pressure or operating loads. The torque preload stresses in the flange are critical and governing; hence, pressure and operating loads are not considered in the nozzle flange analysis.

Table 2. Allowable Load Table

Item	Description of component/ item	Allowable stress ($klbf/in^2$)	Pre-load torque-generated forces (Mechanical) klbf	Allowable Pre-load torque-generated forces considering thermal effects klbf	Remarks
1	Hooks	69[1]	21.93	18.38[1]	Refer to Figure 10
2	Nozzle	50[2]	17.81	14.26[2]	Refer to Figure 14
3	Block	30[2]	17.5	17.5[2]	Refer to Figure 13
4	Gasket retainer ring	16.7[3]	see remarks	see remarks	Gasket retainer ring will yield and provide alternate load path
4	Operating screw	see remarks	see remarks	48[1]	Good for 48 klbf loading
5	Operating nut	see remarks	see remarks	48[1]	Good for 48 klbf loading
6	Hook pins	46.2[1]	18.32	14.77[1]	Bending stress is critical
7	Stop leakage (leakage study)		6.1	6.1	This force includes a factor of safety of 2.5 (see Figure 3)

Notes:

(1) These values are 60% of yield based on American Institute of Steel Construction (AISC 1989) design criteria.

(2) Based on the use of American Society of Mechanical Engineers Boiler and Pressure Vessel (ASME 1989) Code, Section III, Sub-Section NB 3221.3.

(3) Based on the use of ASME B&PV Code B31.3 temperature dependent allowable stresses (ASME 1990).

Friction between different mating surfaces plays an important role in stabilizing the block. The attached jumper pipe connection on one side of the block causes unequal distribution of compression forces on the nozzle face, resulting in possible rotation of the block. The block rotation may lead to unequal clamping forces on the hooks. This phenomenon is resisted by the friction force acting at the interface between the screw and the block. Hence, hook stresses are not significantly affected by the operating loads and are not considered in the hook analysis.

High stress concentration resulting from abrupt change of profile or concentrated loading is possible in the hooks, nozzle, and block. The hook tip that clamps with the nozzle flange experiences high local concentration of stresses. The operating loads produced stress concentration in the block around the opening where those loads are applied. The nozzle flange also has high stress concentration both at the hook engagement points and near the root of the flange. Peak stresses developed at the region of local concentration will redistribute to adjacent elements because of high ductile behavior of the material. These peak stresses are not considered as peak design stresses for this analysis.

The allowable torque load for the nozzle flange is 14.26 klbf, which is lower than the hook pin allowable of 14.77 klbf. The difference between these two allowable torque load values is approximately 500 lbf, which for all practical purposes is judged to be insignificant and is considered to be within the tolerance of engineering analysis accuracy limits. Also, it should be noted that the 14.77 klbf allowable torque load for the pin is based on a calculated maximum principal stress value, whereas the 14.26 klbf allowable torque load for the nozzle is based on a more conservative calculated stress intensity value. In addition, the nozzle material, CD4MCu, is slightly more ductile than the pin material, ASTM A108, Grade 1045 (16% versus 12%, see Table 1). Hence, because of calculational accuracy limitations, conservatism with respect to the calculated stress intensity value for the nozzle, and the slightly higher ductility of the nozzle material, the critical component in the jumper connector assembly is considered to be the hook pin that reaches its allowable bending stresses at a torque-generated load of 14.77 klbf. The allowable stresses for the block is considered as 1.5 S_m, or the yield strength and the corresponding torque load is 17.5 klbf, which is higher than the pin or the nozzle flange.

The maximum torque-generated force for the jumper assembly, including the operating loads, is rated as 14.77 klbf. Safety is already included in the allowable stresses and need not be considered further.

The minimum preload torque-generated force is 6.1 klbf (Figure 4) to prevent leakage resulting from internal pressure and operating loads. This load includes a safety factor of 2.5.

From the preliminary torque testing results, the minimum torque to prevent leakage at design temperature and pressure will be approximately 70 ft-lb. The maximum torque force of 14.77 klbf will be approximately 150 ft-lbf.

CONCLUSIONS AND RECOMMENDATIONS

The conclusions and recommendations of this analysis are summarized below:

- The maximum preload torque-generated forces should be limited to 14.77 klbf to keep the stress levels of all components within safe allowable limits. This torque force will correspond to approximately 150 ft-lbf of torque
- The minimum torque-generated forces should be 6.1 klbf to seal the gasket for an internal pressure of 400 psi and at 240 °F. The corresponding minimum torque value will be approximately 70 ft-lbf
- The allowable reactions from the operating loads should not exceed the enveloped absolute values of $F_x = F_y = F_z =$ 400 lbf and $M_x = M_z = 350$ ft-lbf, where x, y, and z are the three orthogonal directions and the x direction is axial for the jumper piping and y direction is parallel to the nozzle-block axis. The moment, M_y, is released at the connector point.
- The torque versus force relationship should be linear to fix the torque rating for this equipment. Friction force changes if galling occurs at high torque loads. The working preload torque range as specified should not cause any galling. More research is recommended on the materials or design to prevent galling effects.
- The lower skirt and gasket retainer ring should be assembled such that the top surface of the skirt is in contact with the block's stepped surface for transferring reactions normal to the torque load into the nozzle base.
- The tie bolts should be torqued snug tight to avoid unnecessary compressive stress on the block cross section.
- Because the jumper assembly is sensitive to preload torque, the remote control torque wrench should be controllable or calibrated and have the capability to exert measured amounts of torque.
- The gasket functions both as a load bearing and leakage sealing unit. The efficiency of the jumper assembly may be increased if the interface between the block and the nozzle is redesigned to provide clamping action at the interface as in the case of flanges.

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6 / 28 / 94

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