NRC/ANL INDUSTRY MEETING

Dated: November 6, 2008

H* Results Study

3D Analysis Results
Radius Dependent Tubesheet Stiffness Results
Combined Applied Loading
Axisymmetric vs. 3D Results

Background

- Reference model for H* calculations was the Axisymmetric Linear Superpositioning (ALSP) model
  - Unit loads calculated separately, then combined by linear scaling and superpositioning
  - The divider plate is not an axisymmetric component
  - Approach benchmarked against 3D Finite Element Analysis (FEA) and shown to be very conservative
- Current preferred approach is the 3D FEA model
  - Integrated loading
  - Responds to staff questions

H* Contact Pressure Theory Basis

- Based on Initial work by Goodier (1943)
- Includes material effects, operational effects, etc. that affect the tube-to-tubesheet contact pressure distribution from the top of the tubesheet to the elevation of a flaw within the tubesheet.
- Goodier model is refined and bench-marked by test data.

Factors in H* Contact Pressure Solution

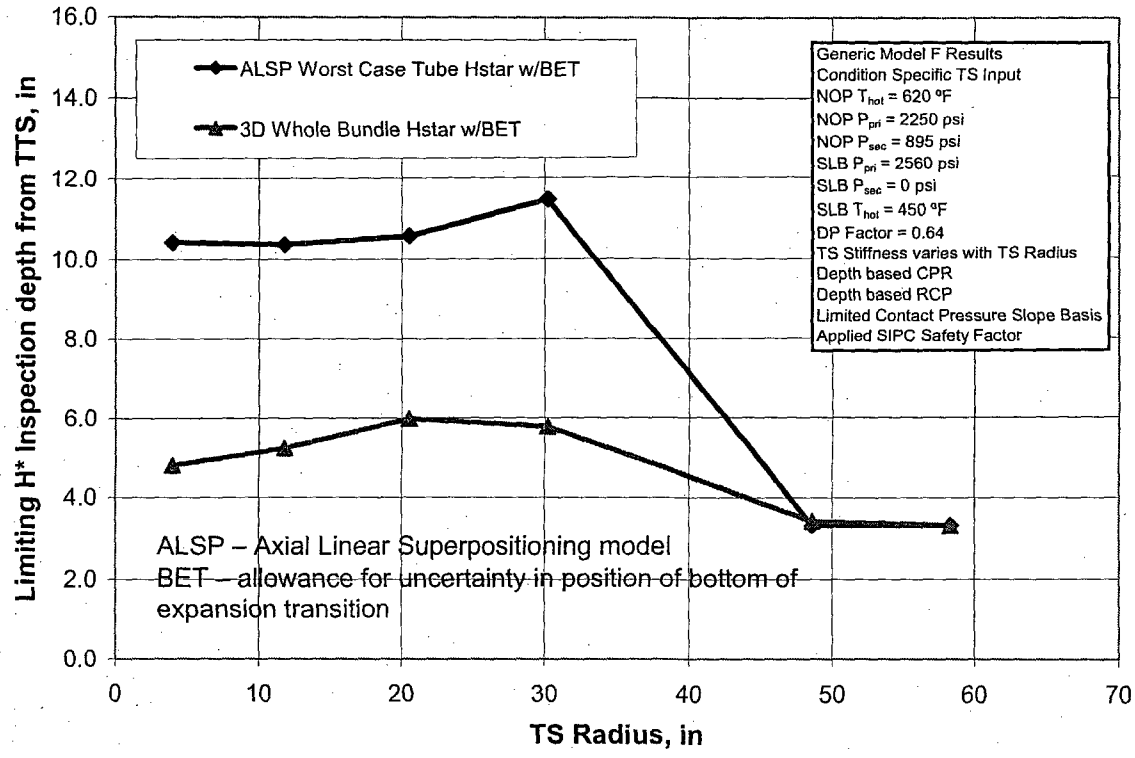
- Crevice Pressure Distribution
  - Determined using Test Data and Analysis
- Tubesheet Rotation & Displacement
  - Determined using 3D and Axisymmetric FEA
- Tube & Tubesheet Bore dilation and contraction
  - Determined using 2D FEA
- Residual Contact Pressure
  - Determined using 2D FEA
  - Test data simulates real SG manufacturing conditions
- Variation in Material Properties
  - Thermal growth of Tube & Tubesheet Bore
  - Determined using Test Data

H* Contact Pressure Evaluation

$$P_{contact} = \frac{c\alpha_t(T_t - 70) + \frac{P_{pri}c}{E_t}\left[\frac{(2-\nu)b^2}{c^2-b^2}\right] - \frac{P_{cp}c}{E_t}\left[\frac{(1-2\nu)c^2+(1+\nu)b^2}{c^2-b^2}\right] - \frac{P_{cp}c}{E_{TS}}\left[\frac{d^2+c^2}{d^2-c^2}+\nu\right] - \Delta R_{ROT}}{\frac{c}{E_t}\left[\frac{c^2+b^2}{c^2-b^2}-\nu\right] + \frac{c}{E_{TS}}\left[\frac{d^2+c^2}{d^2-c^2}+\nu\right]}$$

- P_i = Internal primary side pressure, P_{pri} psi
- P_o = External Pressure = Crevice pressure, P_{cp} psi
- b = Inside radius of tube
- c = Outside radius of tube
- d = Outside radius of equivalent tubesheet collar
- α_t = Coefficient of thermal expansion of tube, in/in/°F
- E_t = Modulus of Elasticity of tube, psi
- E_{TS} =Modulus of Elasticity of tubesheet collar, psi
- T_t = Temperature of tube, °F
- ν = Poisson's Ratio of the material

Current H* Results



- Contact pressure distributions for all operating cases compared and worst case results at each radius used for H*.
- Worst case tube H* defined using SIPC, limiting TS sector, worst condition result (ALSP).
- Whole bundle (95/50) analysis based on 5500 tubes in Model F bundle.

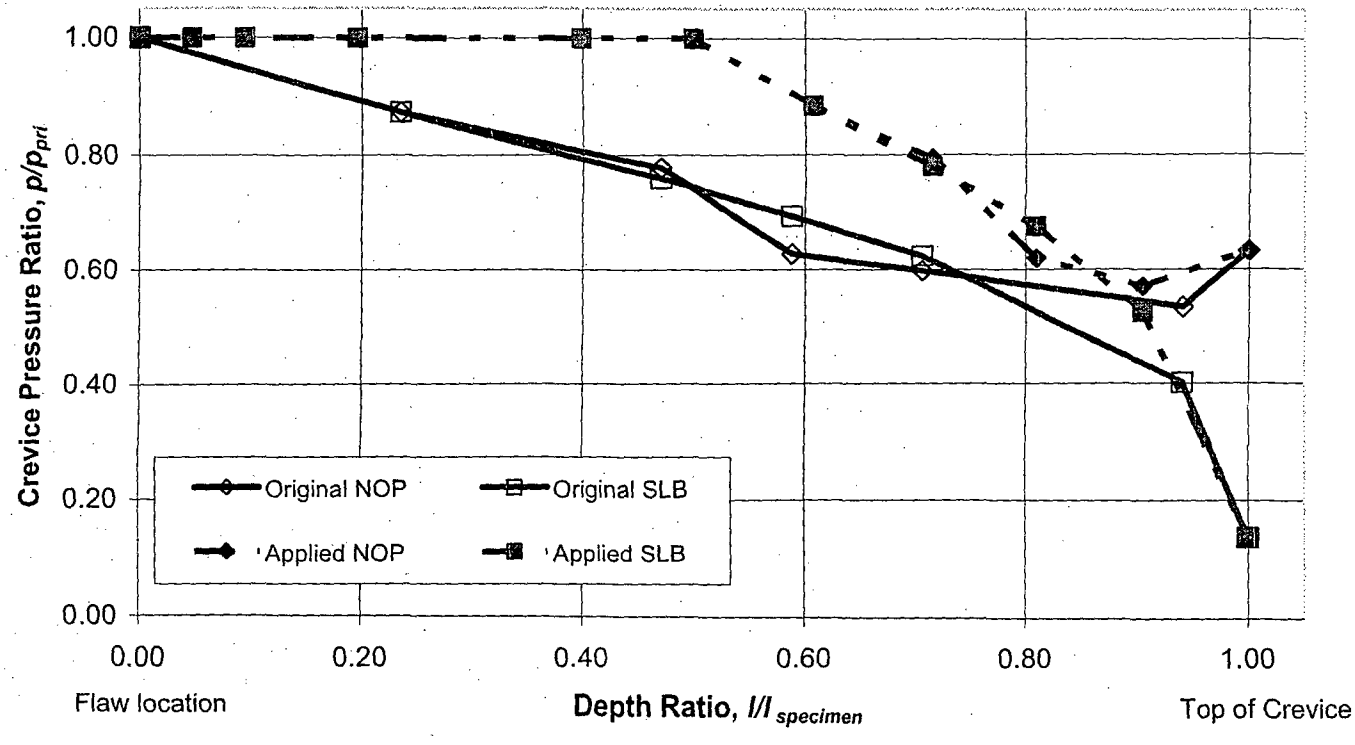
Crevice Pressure Distribution

- Based on test data from pressure taps with axial offset. (LTR-CDME-07-198)
- Scaled to apply test specimen data to real tubesheet crevice.
- Initially applied to penalize structural and leakage analysis using limiting median value.
- NRC RAI questioned use of limiting median value and identified an alternate ("depth based") approach (December 2007).
  - The depth based approach utilizes average of crevice pressure test values
  - References the crevice pressure distribution to the predicted value of H*

Crevice Pressure Distribution

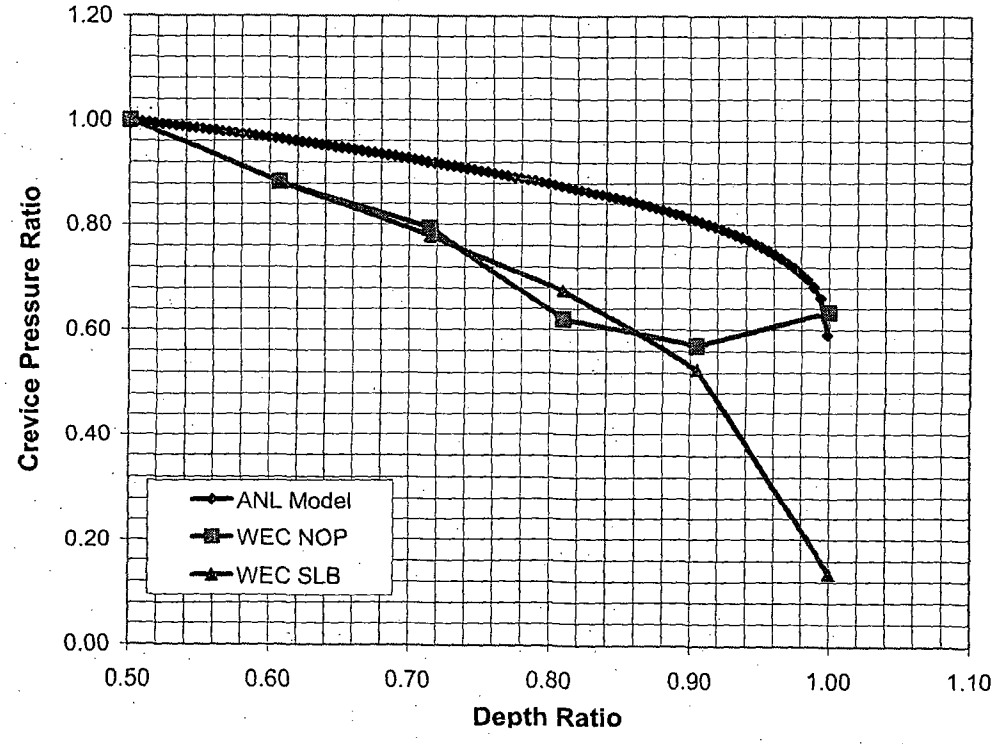
- Major effects on H*/Tube Integrity:
  - Defines pressure expansion of tube and TS
  - Defines driving potential for leakage in crevice
  - Determines final length of tube required and value of H*
- Requires iterative H* solution
  - Scale crevice pressure length to first prediction of H*
  - Recalculate H*

Crevice Pressure Distribution



Crevice pressure model applied as ratio to scale results to final applied tube length.

Crevice Pressure Distribution



- ANL and WEC crevice pressure distributions are similar.
- WEC test data shows larger pressure difference near TTS.
- WEC test results for SLB and NOP are very different.

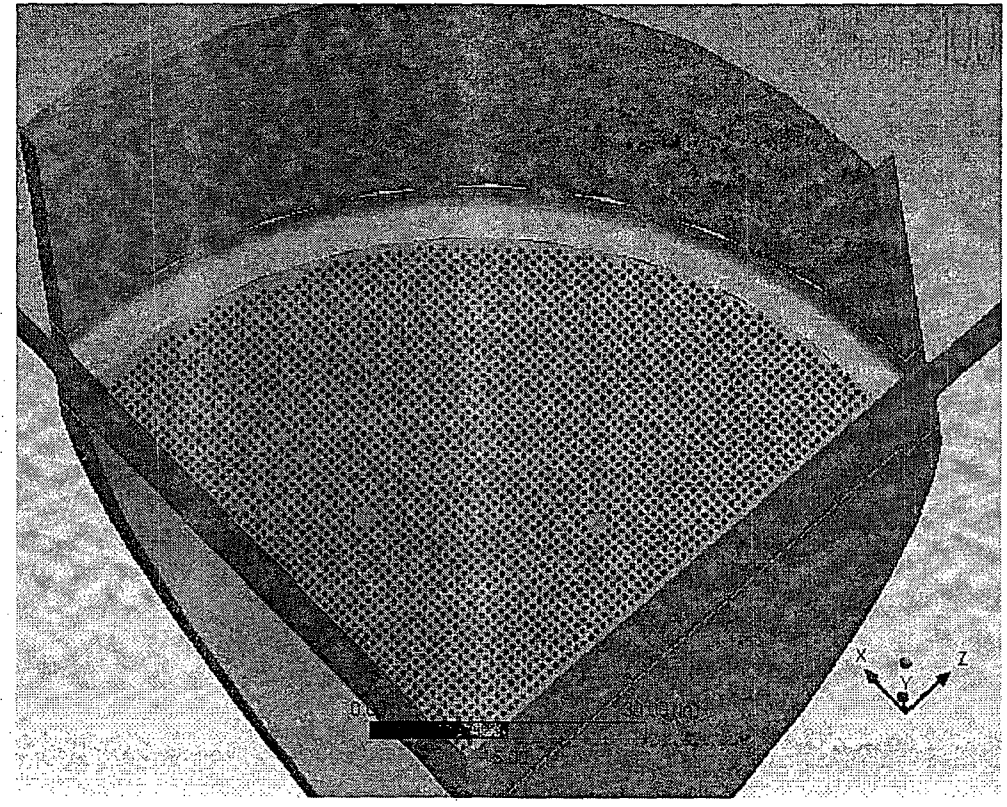
Tubesheet Rotation & Displacement

- Two different models and approaches used in past:
  - 3D applied loads and displacements
  - Axisymmetric Linear Superposition (ALSP)
- 3D finite element analysis preferred
- NRC RAI requested 3D model to benchmark ALSP

3D Tubesheet Displacements

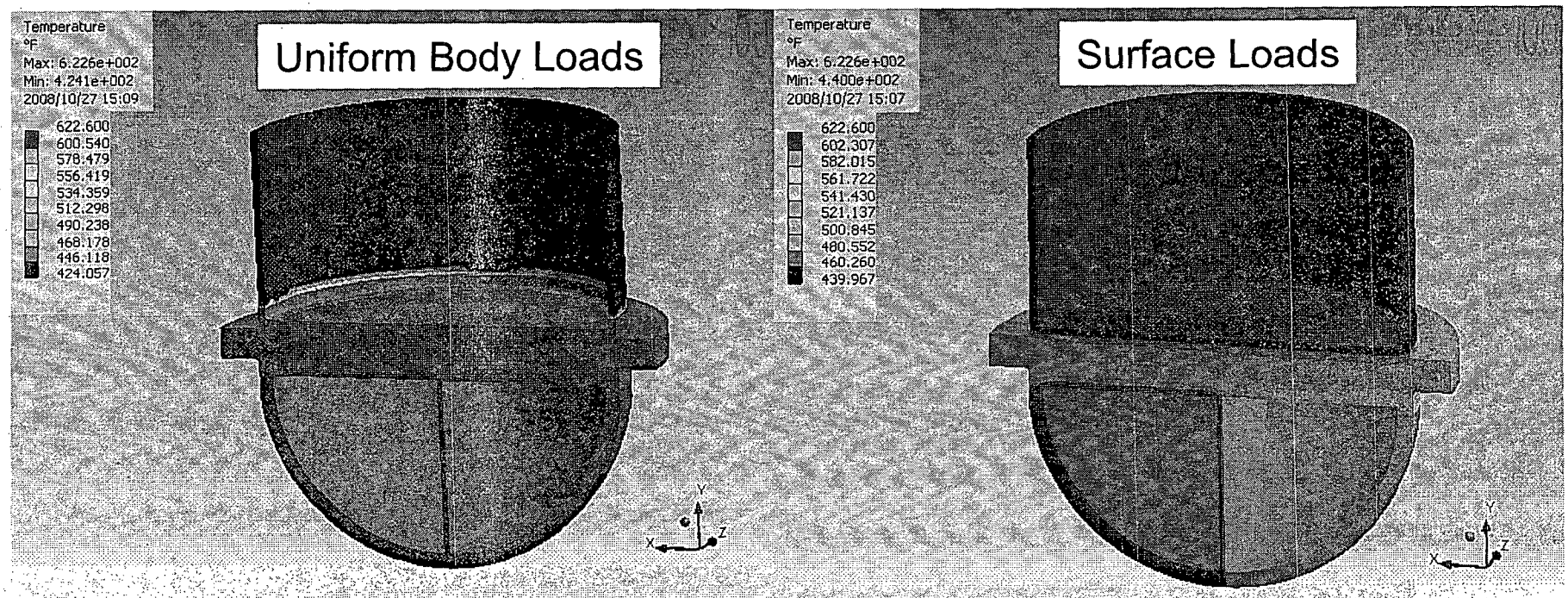
- TS displacements vary with respect to radius and angle
- Maximum TS displacements due to pressure loading occur perpendicular to DP
- Minimum TS displacements due to pressure loading occur parallel to DP
- TS displacements due to thermal loads are similar throughout the TS
- Reduced TS displacements (compared to ALSP model) lead to increased T-TS Contact Pressure

ANSYS WB Model of SG



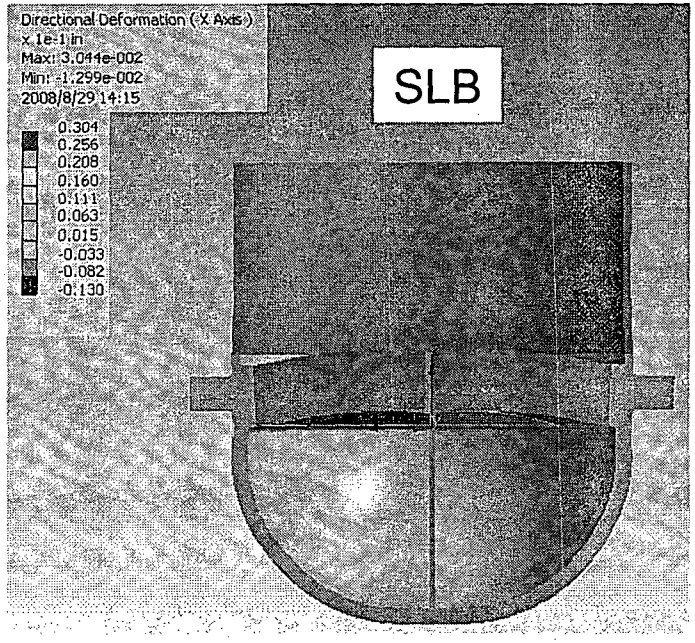
- ½ symmetry and ¼ symmetry FEA models used.
- Tube perforations included as separate model layer.
- Anisotropic TS material model (Ref. Slot) used for global TS displacement analysis.
- Perforated local TS models used for areas of concern.

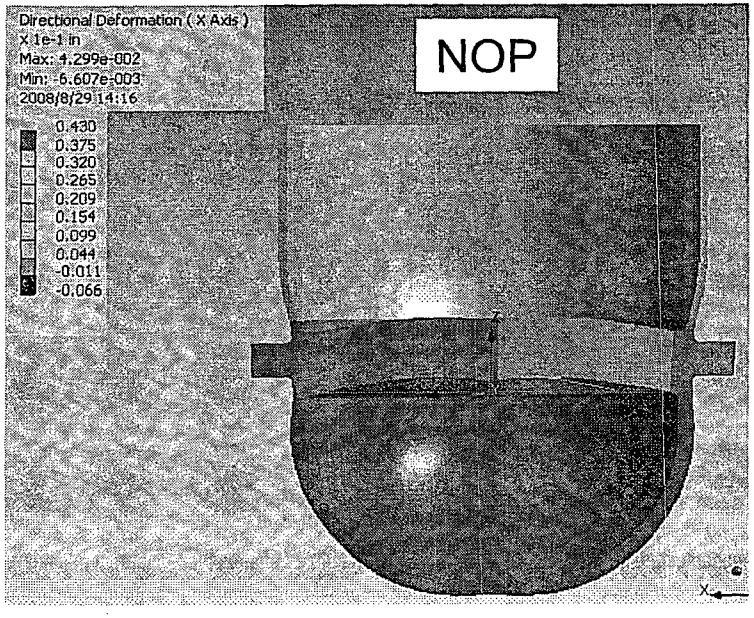
Application of Thermal Loads



- Typical NOP Temperature Distributions
- Uniform Body Loads produce Greater TS Displacements
- ALSP uses Uniform Body Loads
- 3D FEA model uses more realistic surface loads

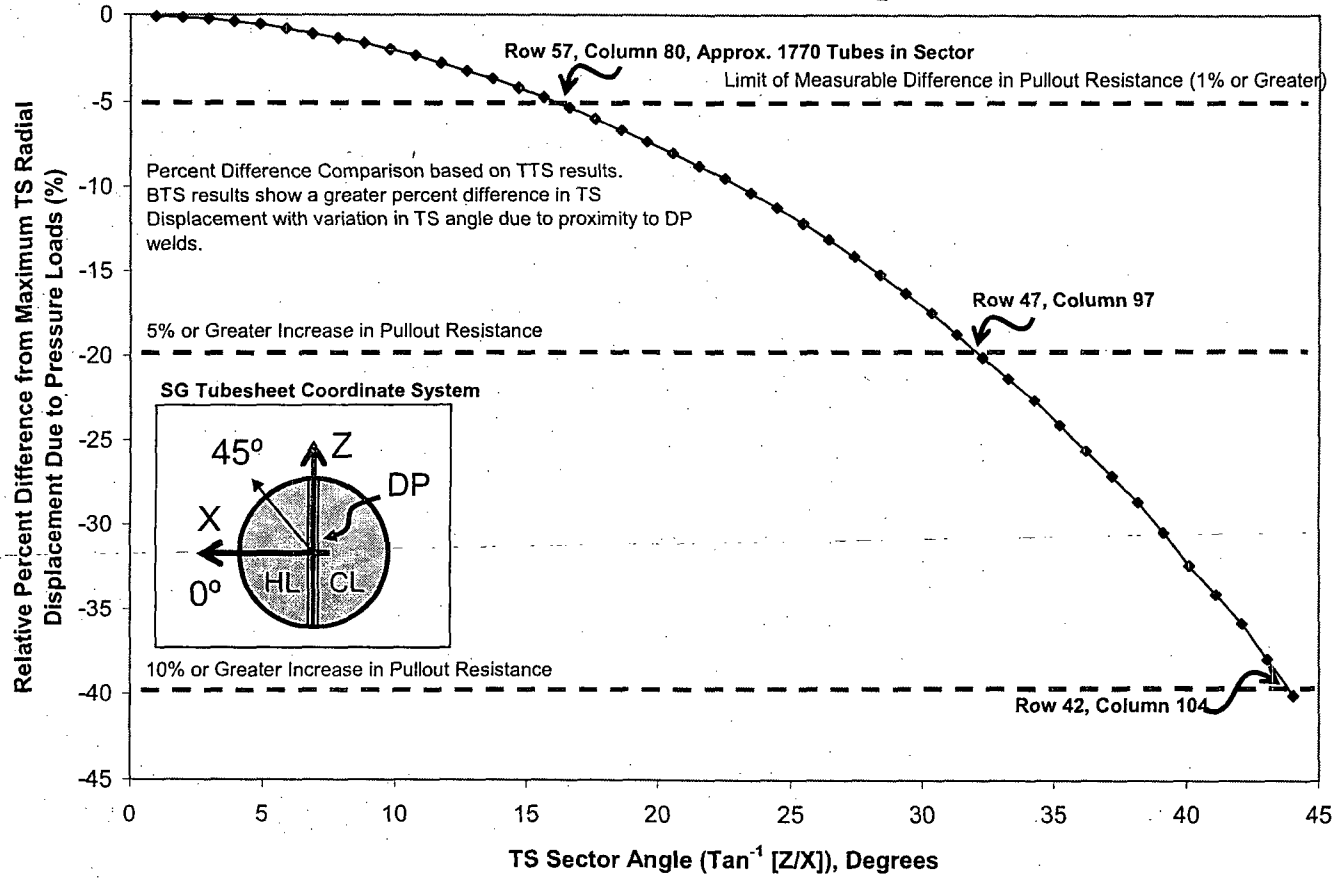
Application of Pressure Loads





- 3D boundary conditions vary with SG model due to geometry differences.
- 3D Applied Load Cases prove that axisymmetric loading is conservative.
- ALSP approach does not capture 3D mode shape of TS.
- 3D Applied Load TS Disp. << Axisymmetric Unit Load TS Disp.

3D Tubesheet Displacements



- Primary and Secondary Pressure loads applied simultaneously
- Reference case is maximum TS Radial Displacement at 0° (90° away from DP)
- H* analysis is based on worst case TS azimuth (90° away from DP)

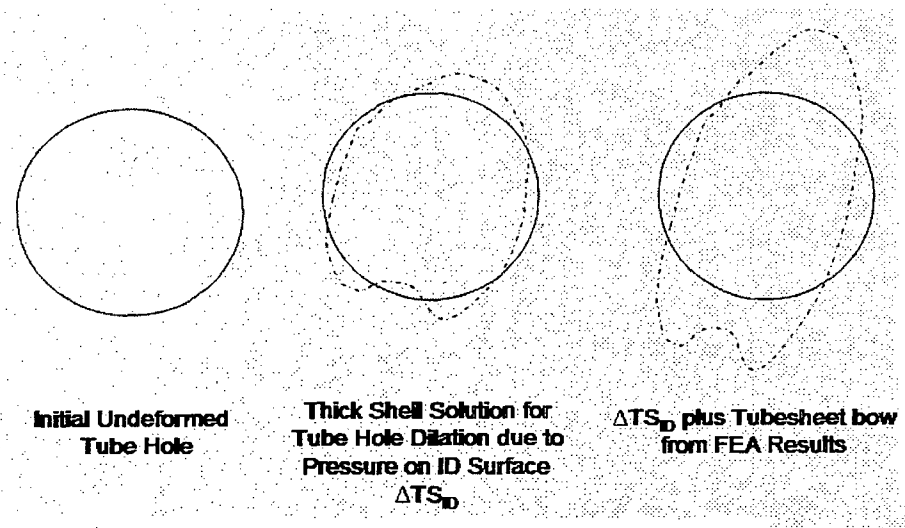
Tube & Tube Bore Effects

- Goodier analysis assumes tube and tubesheet always in contact around circumference.
  - Elastic restoring forces cause surfaces to follow each other through deformations.
- WEC analysis decomposes contact pressure into tube effects and tubesheet effects with T-TS interaction coefficients to maintain compatibility.

Tube & Tube Bore Effects

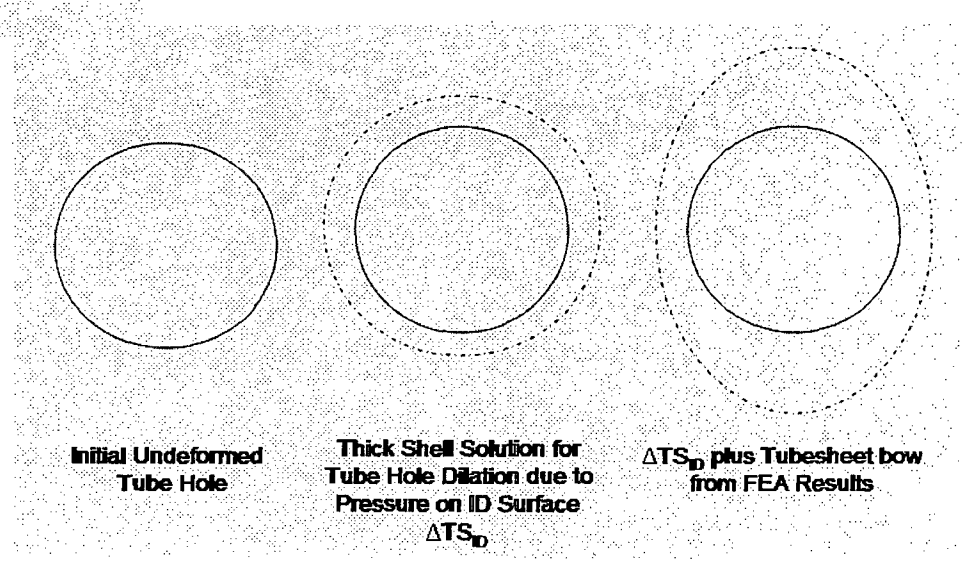
- W analysis conservatively allows for tube bore to dilate away from tube such that no benefit to contact pressure is gained.
  - No "pinching," just major axis ovalization.
  - Thermal growth and pressure dilation act on major and minor axis of tube bore.
  - Tube effects then compared to see if tube OD "catches up" to tube bore deformation.
  - Predicted value of dilation, ~10^{-5} in., <<bore surface finish; no effect on leakage analysis
- Tube & Tube Bore dilation effects in a real tubesheet quantified in tubesheet stiffness study.

Tubesheet Bore Deformation in H*



- Non-uniform deformations may occur in real TS.
- The tube follows the tubesheet.

- H* analysis requires uniform deformation.
- Decomposes contributions to tube and tubesheet deformation.



Original Tubesheet Stiffness Study

- Original TS Collar (unit cell) wall thickness determined by pressurizing single tube in square pitch bundle.
- No other TS bore pressurized.
- Minimum material properties used.
- Appropriate for calculation of TS displacements around a single hole.
- Underestimates TS bore stiffness in bundle because it ignores restoring effects from adjacent pressurized tubes.

Tubesheet Stiffness Study

- Restoring effect of adjacent pressurized tubes was determined in a separate study.
- Objective was to improve accuracy of modeling a specific tube at a given TS radius.
- Tubesheet stiffness coefficient (TS OD, d) can now vary with radius based on constraint from boundary conditions.
- Result rationalizes ALSP model response to uncertainties; no significant effect on 3D FEA solution

Tubesheet Stiffness Study

- Two Plane Stress Finite Element Models used:
  - Model 1: Represents limiting region in bundle, perpendicular to DP
  - Model 2: Represents stiffest region in bundle, parallel to DP.
- Two cases considered:
  - All tubes pressurized.
  - Three "tubes of interest" pressurized.
  - Shown to bound all other conditions
- Two pressure cases used on tube bore:
  - SLB: 2560 psi, P_{sat} = 358 psi
  - NOP: 2250 psi, P_{sat} = 1418 psi

Tubesheet Stiffness Study

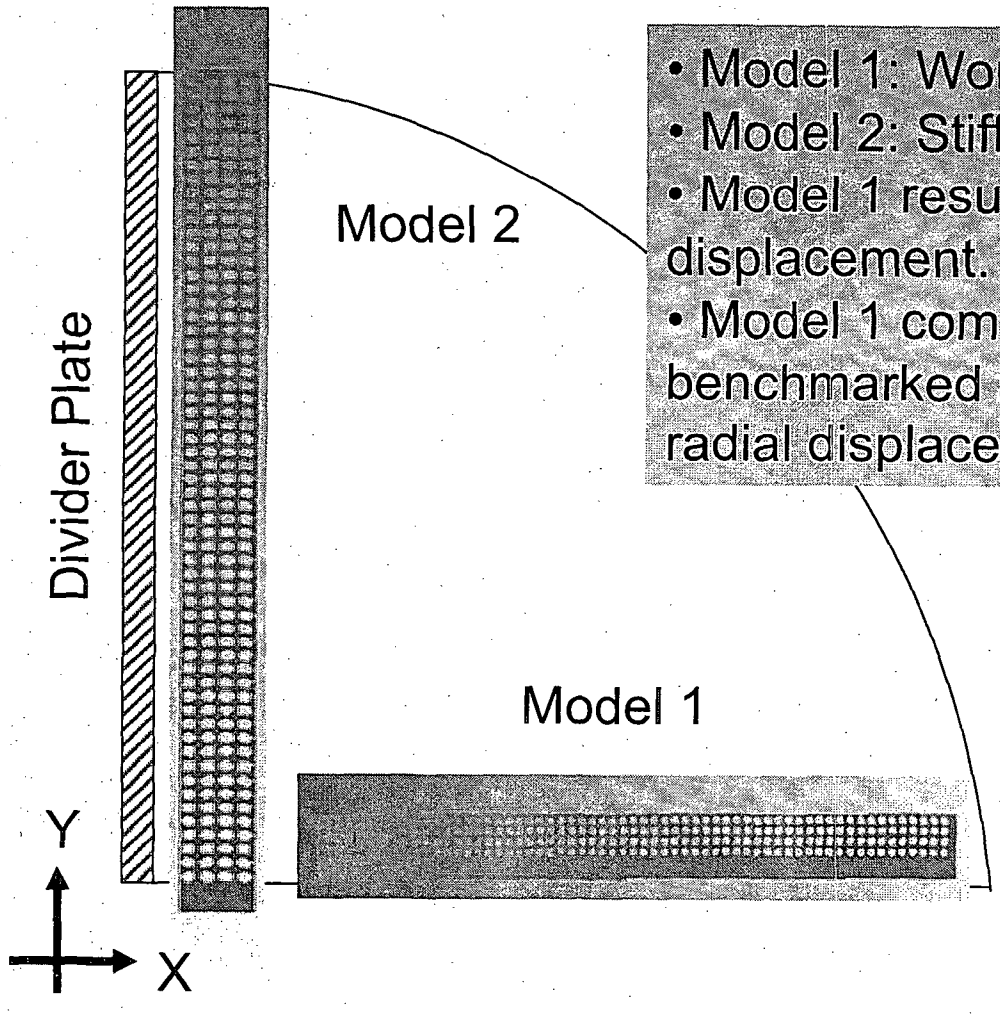
$$\left(\Delta TS^{ID}_{\Pr essure}\right)_{TS} = \frac{cP_{CREV}}{E_{TS}}\left[\frac{d^2+c^2}{d^2-c^2}+\nu\right]$$

$$\left(\Delta TS^{ID}_{\Pr essure}\right)_{FEA} = \int_0^{2\pi} A_{FEA}\,d\theta = \sum_0^{360}\frac{1}{2}(a+b)d\theta$$

$$\Delta FEA = \frac{\left(\Delta TS^{ID}_{\Pr essure}\right)_{FEA}}{360} - \left(\Delta TS^{ID}_{\Pr essure}\right)_{TS}$$

- ΔFEA represents net difference between FEA and TS Results.
- Net difference is required because real tube bore dilation is not uniform.
- If ΔFEA>0, FEA ΔTS_{ID} predicts greater tube bore dilation than Thick Shell Equations.
- If ΔFEA<0, FEA ΔTS_{ID} predicts less tube bore dilation than Thick Shell Equations.

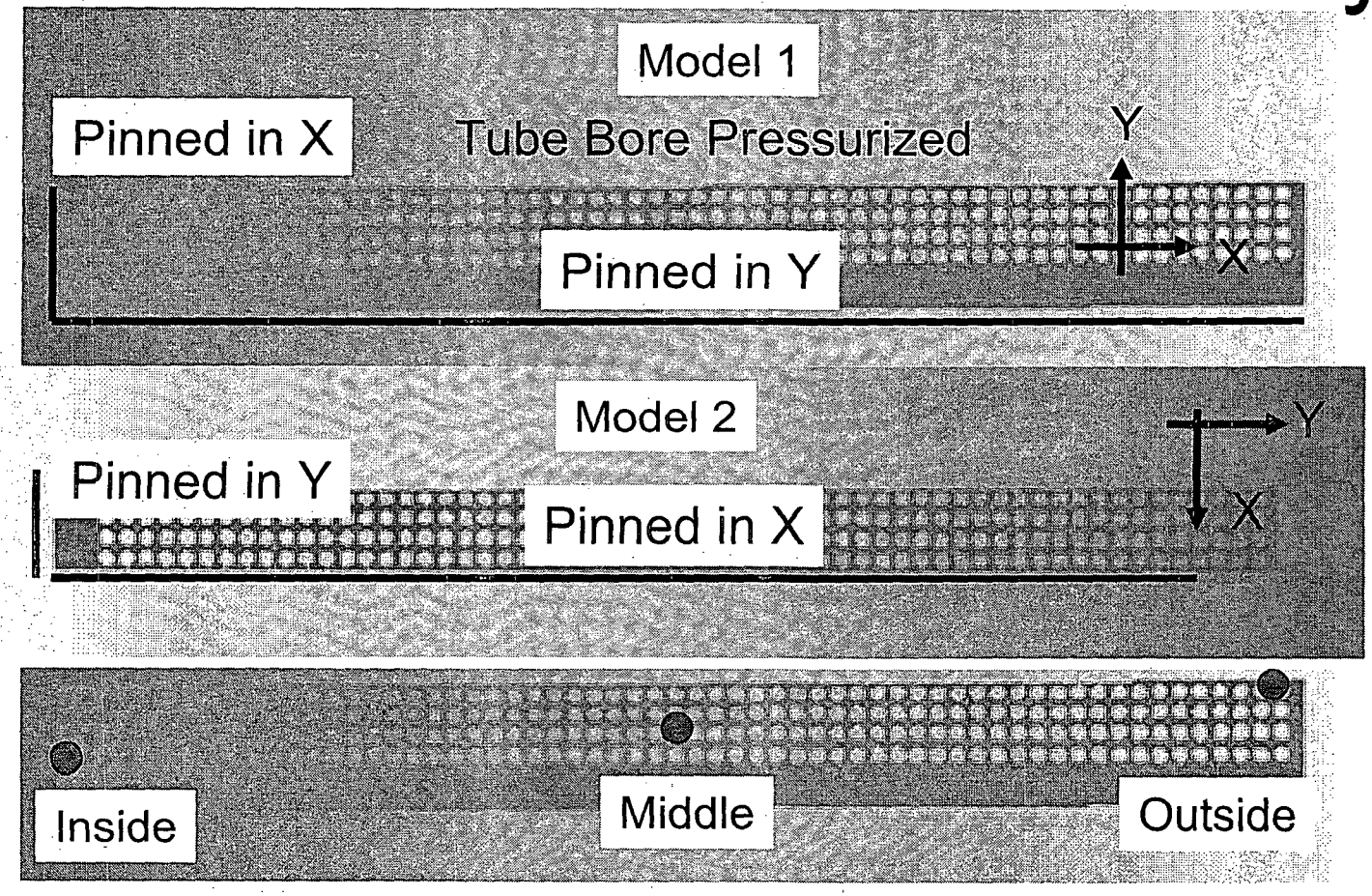
Tubesheet Stiffness Study



- Model 1: Worst Case TS Displacements
- Model 2: Stiffest TS Sector
- Model 1 results show greatest displacement.
- Model 1 combined load case benchmarked against 3D TS within 1% of radial displacement.

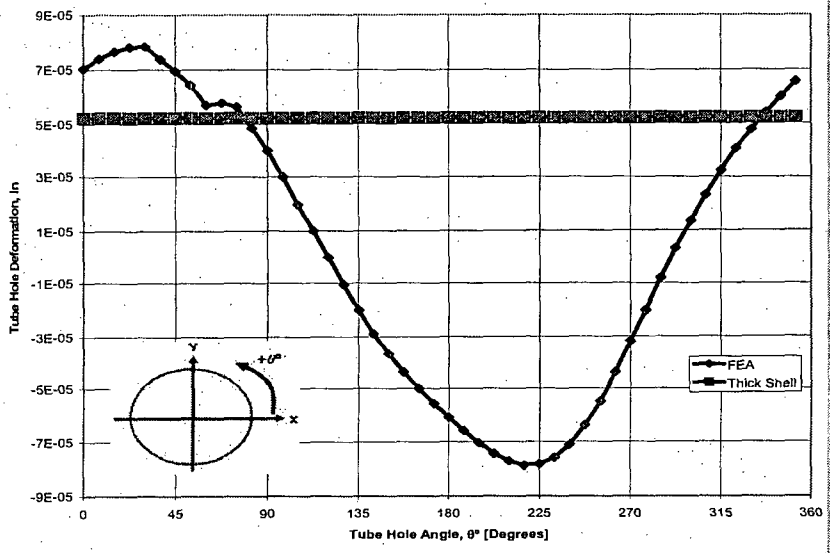
Additional edge material added for conservatism and to separate tube bore from BCs.

Tubesheet Stiffness Study

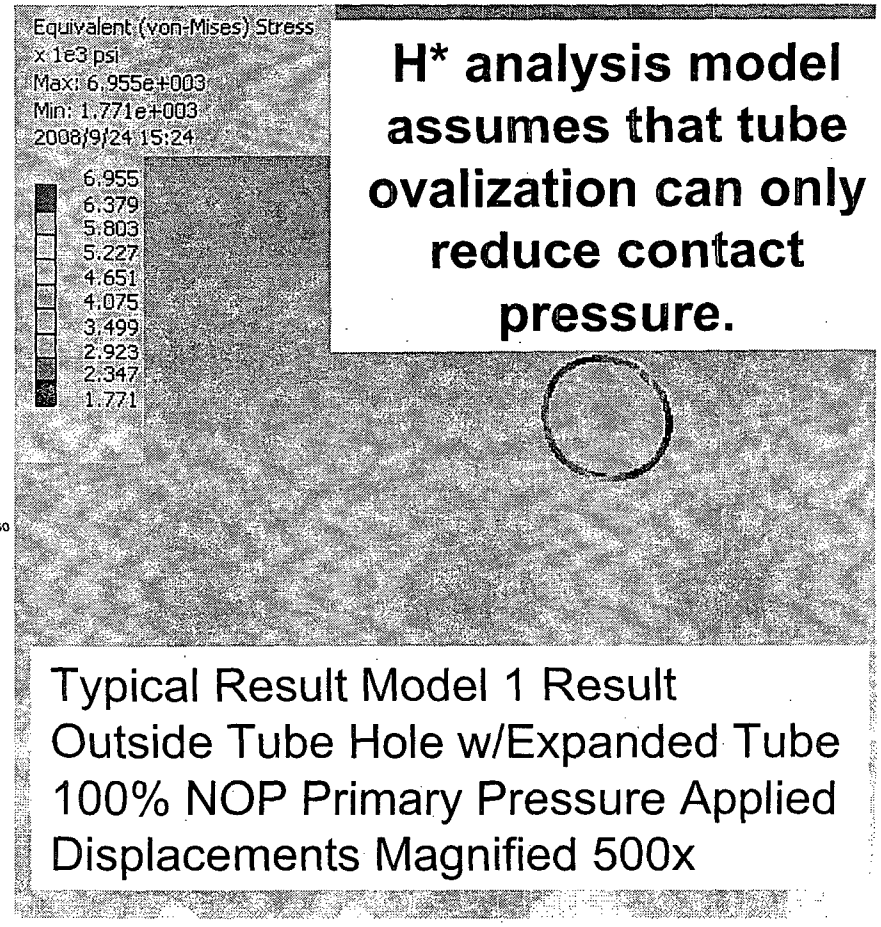


- 3 tube bores of interest constant in each model.
- Analysis also included expanded tubes in tubesheet bores of interest.

Tubesheet Stiffness Study



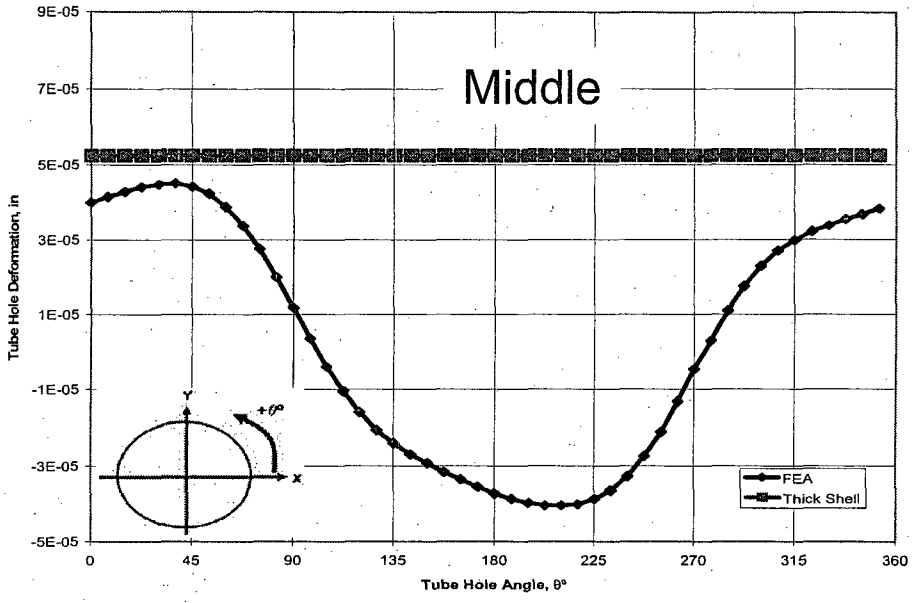
Typical peripheral ("outside") tubesheet bore deformation for a pressurized tubesheet.

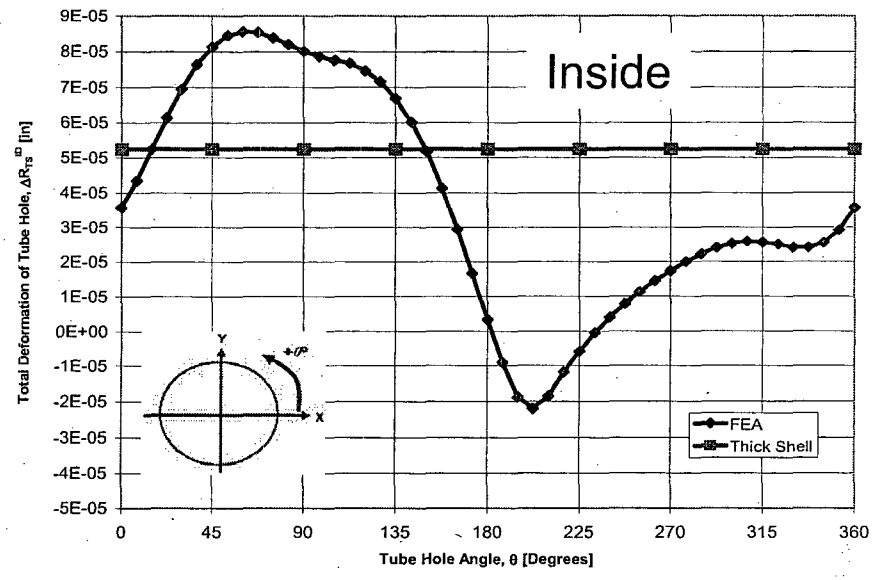


H* analysis model assumes that tube ovalization can only reduce contact pressure.

Typical Result Model 1 Result
Outside Tube Hole w/Expanded Tube
100% NOP Primary Pressure Applied
Displacements Magnified 500x

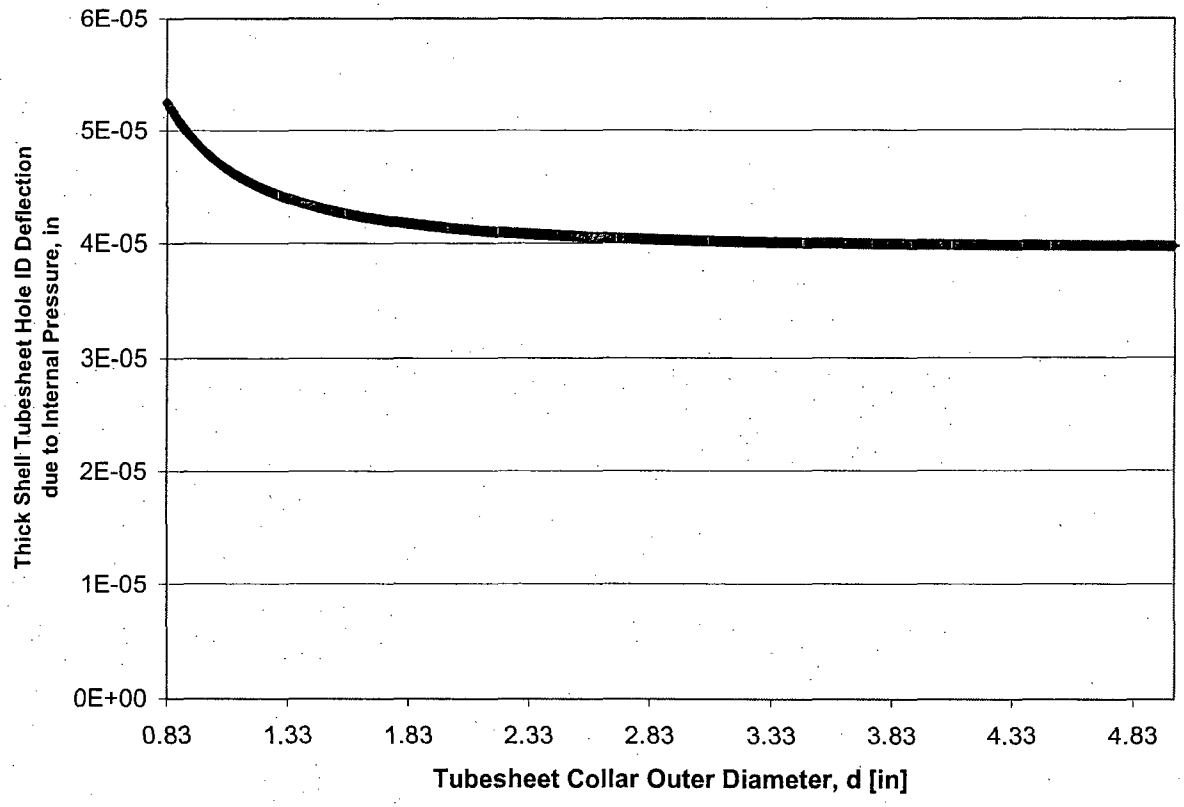
Tubesheet Stiffness Study





- Mode shapes of tube bore ovalizations are "constant."
- Scale of deformation changes based on tubesheet bore pressure.
- All tubesheet bore locations show less average deformation than Thick Shell equations.

Tubesheet Stiffness Study



Thick Shell tube bore ID dilation is asymptotic with respect to ***d***.
H* calculation model uses dilations not predicted to be theoretically possible in a real TS;
No significant effect in most of bundle.

Tubesheet Stiffness Study

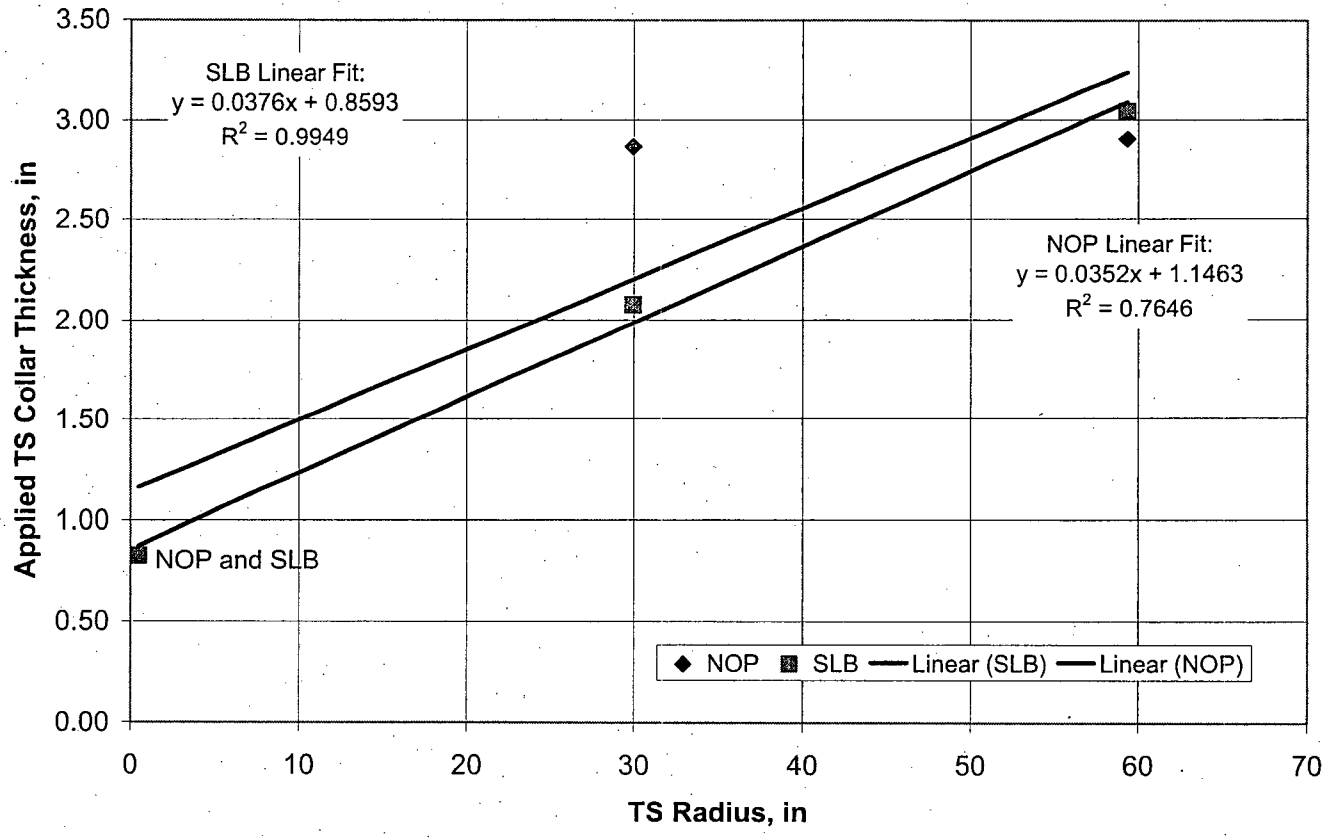
Condition	Location	Thick Shell Deflection	Avg. FEA Deflection	% Diff
-	-	in	in	-
SLB	Inside	5.72E-05	3.43E-05	-40.1
SLB	Middle	5.72E-05	-7.53E-08	-100.1
SLB	Outside	5.72E-05	-2.68E-05	-146.8
NOP	Inside	5.24E-05	3.77E-05	-28.0
NOP	Middle	5.24E-05	1.79E-06	-96.6
NOP	Outside	5.24E-05	1.10E-06	-97.9

- FEA Results predict a net **reduction** in tube bore ID compared to Thick Shell equations.
- Tube bore deflection calculation in H* model must be positive and uniform, but best representation of TS bore dilation indicates non-uniform deformation.
- Non-uniform deformation increases contact pressure.

Tubesheet Stiffness Study

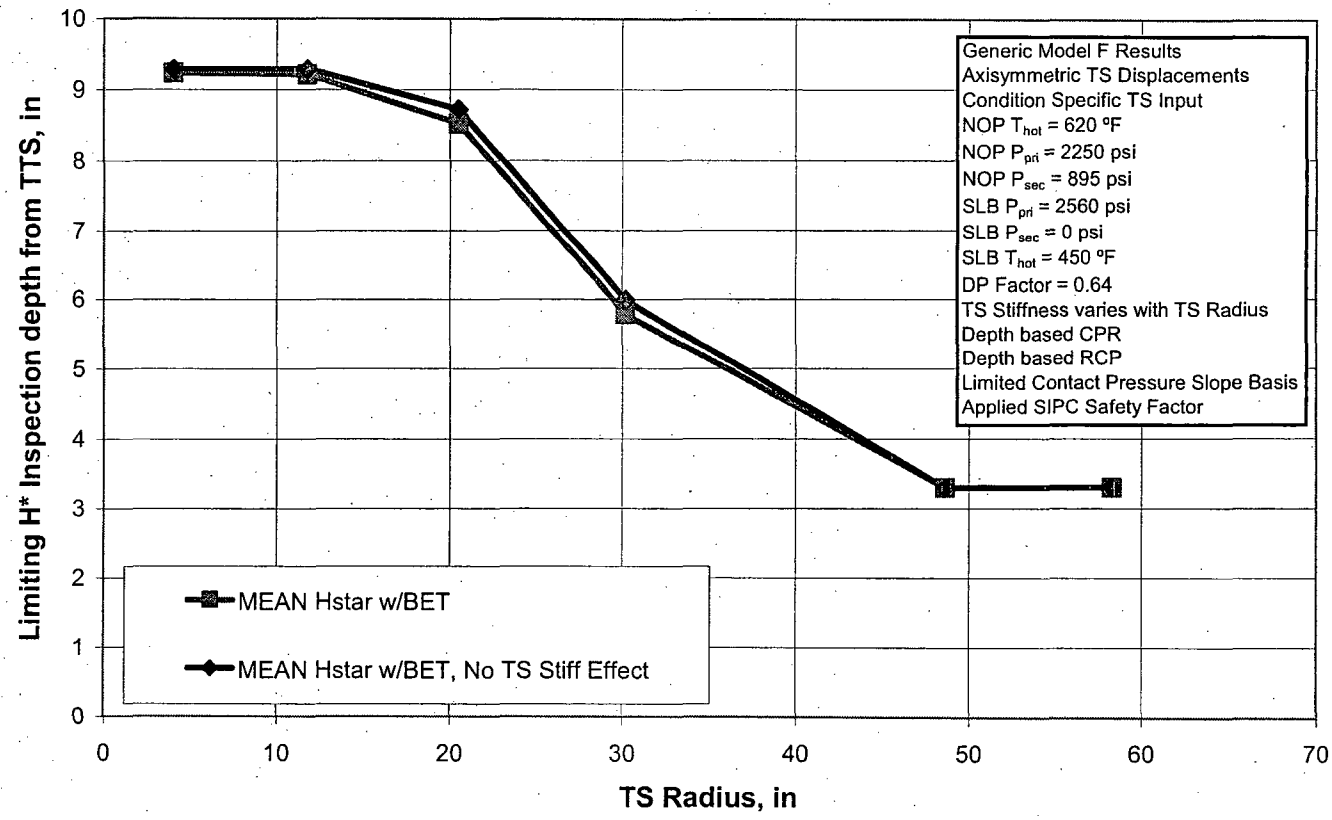
- For all tube bore locations, Net FEA results are less than Thick Shell results.
  - Actual tubesheet bore dilation is less than uniform Thick Shell estimate.
- Thick Shell results adjusted by Net FEA difference by scaling *d*.
- Final *d* Scaling based **trends** from FEA Results.

Tubesheet Stiffness Study



- Minimum *d* value at inside tube bore (Maximum Deformation)
- Maximum *d* value at outside tube bore (Minimum Net Deformation)

Tubesheet Stiffness Study



Radius specific tubesheet stiffness modification has small effect on mean H* (ALSP).

Summary of Tubesheet Stiffness Study

- No changes to existing H* analysis approach.
- Same contact pressure equations used with adjusted TS Stiffness.
- TS Interaction coefficients and TS Collar thickness vary with TS Radius.
  - Tends to increase H* at TS Radius > 30”.

Residual Contact Pressure

- Residual contact pressure (RCP) is defined as that contact pressure due to hydraulic expansion only
- Mean value of RCP is defined based on pullout tests in progress
- Variability of RCP is defined analytically using a 2D FEA model

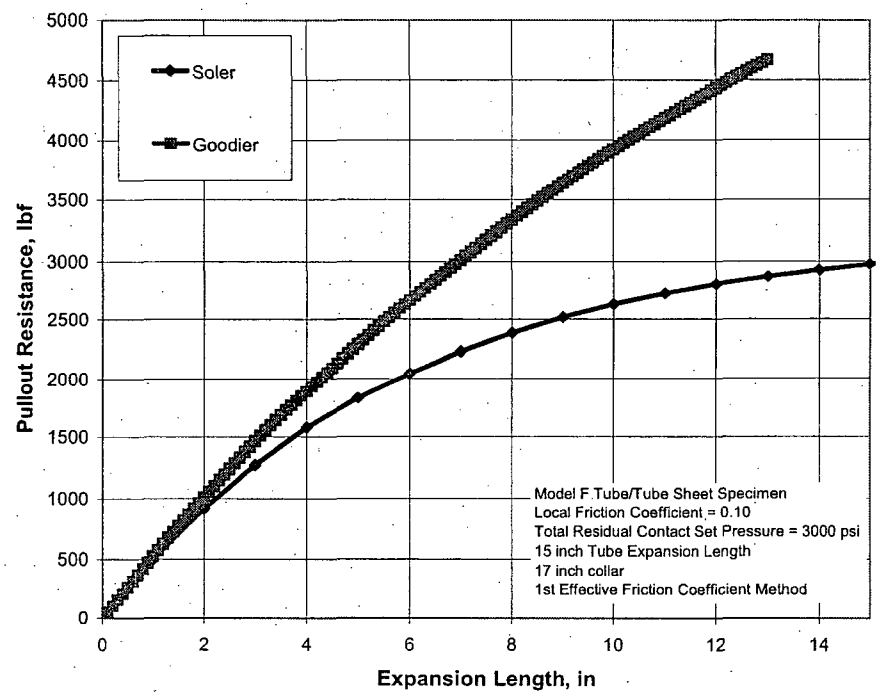
Residual Contact Pressure Study

- H* analysis tool can accommodate any RCP distribution.
- Test results will be used to benchmark modified Goodier pullout model.
- Friction model can be applied as constant value or non-linearly varying model.
- Recent literature results are considered.
  - Soler, Bazergui et al., Allam et al., etc.

References

- Goodier, J., Schoessow, G., Transactions of the ASME, Vol. 65, July 1943, pp. 489-496.
- Soler, A., et al., 1984, *Mechanical Design of Heat Exchanger and PressureVessel Components*, Arcturus Publishers, Cherry Hill.
- Allam, M., Bazergui, A. , ASME JPVT, Vol. 124, February 2002, pp. 22-31
- Merah, N., Al-Zayer, A., Shuaib, A., Arif, A., Int. J. of Pressure Vessels and Piping, Vol. 80, 2003, pp. 879–885

Residual Contact Pressure Study



- Goodier: Assumes constant coefficient of friction.
- Soler: Effective coefficient of friction that varies with expansion length.
- Both models require test data to benchmark values.

H* FEA Residual Contact Pressure Sensitivity Preliminary Results

- Sensitivity study considers the following set of parameters:
  - Unexpanded Tube OD
  - Tubesheet Hole ID
  - Tube wall thickness
  - Tube Yield Stress
  - Tube Elastic Modulus
  - Tubesheet Elastic Modulus
  - Expansion Pressure
  - Level of Strain Hardening
- End result of sensitivity study will be a distribution of residual contact pressure due to $\pm 2\sigma$ variation of each parameter

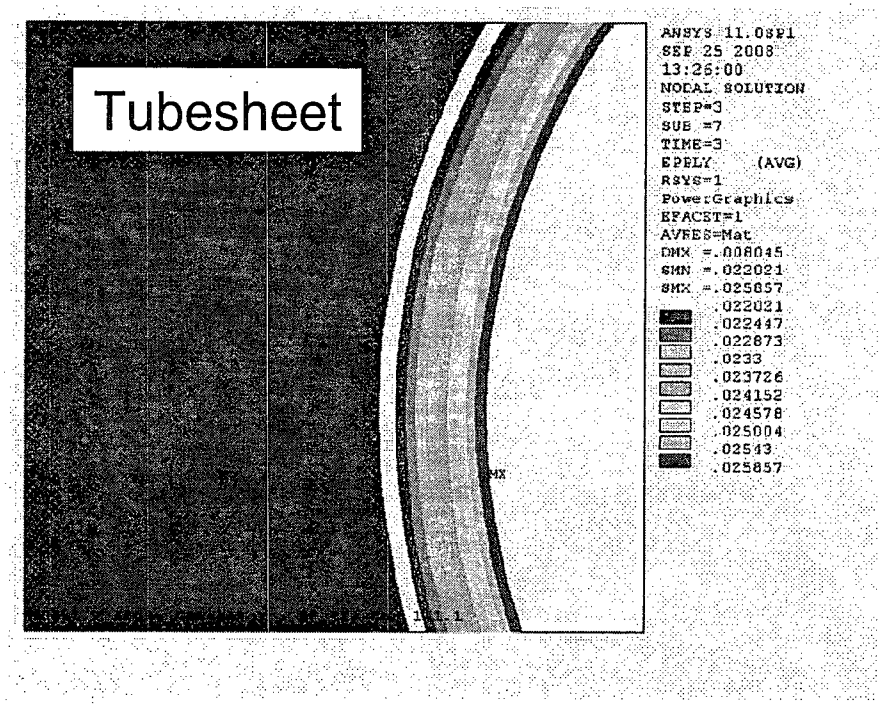
H* FEA Residual Contact Pressure Sensitivity Analysis

- Full 360°, concentric cylinder, 2-D plane stress representation of tube and tubesheet
- Surface-to-surface contact between tube OD and tubesheet hole ID
- Pressure load applied to tube ID
- Multi-linear isotropic strain hardening

H* FEA Residual Contact Pressure Sensitivity Preliminary Results

	Mean Value (psi)	-2σ Value (psi)	+2σ Value (psi)
Tube OD	371	108	544
TS ID	371	530	134
Tube Wall Thickness	371	391	348
Tube Y_S	371	1364	30*
Tube E	371	304	438
TS E	371	438	307
Expansion Pressure	371	200	542
Strain Hardening Curve	371	711	36

H* FEA Residual Contact Pressure Sensitivity Preliminary Results



Typical Plastic Strain contours for mean case plotted on undeformed configuration.

- Finite element model can accommodate radial springback.
- Finite element model conservatively applies expansion pressure to minimize resulting contact pressure distribution.
- Values for plastic strain and deformation in the expanded tube are similar to original study results.

H* FEA Residual Contact Pressure Sensitivity Preliminary Results

- Average residual contact pressure approximately 371 psi
  - H* analysis currently assumes either 300 psi or zero
- Residual contact pressure most sensitive to variations in the tube yield stress
- Residual contact pressure is sensitive to variations in post-yield strain hardening behavior

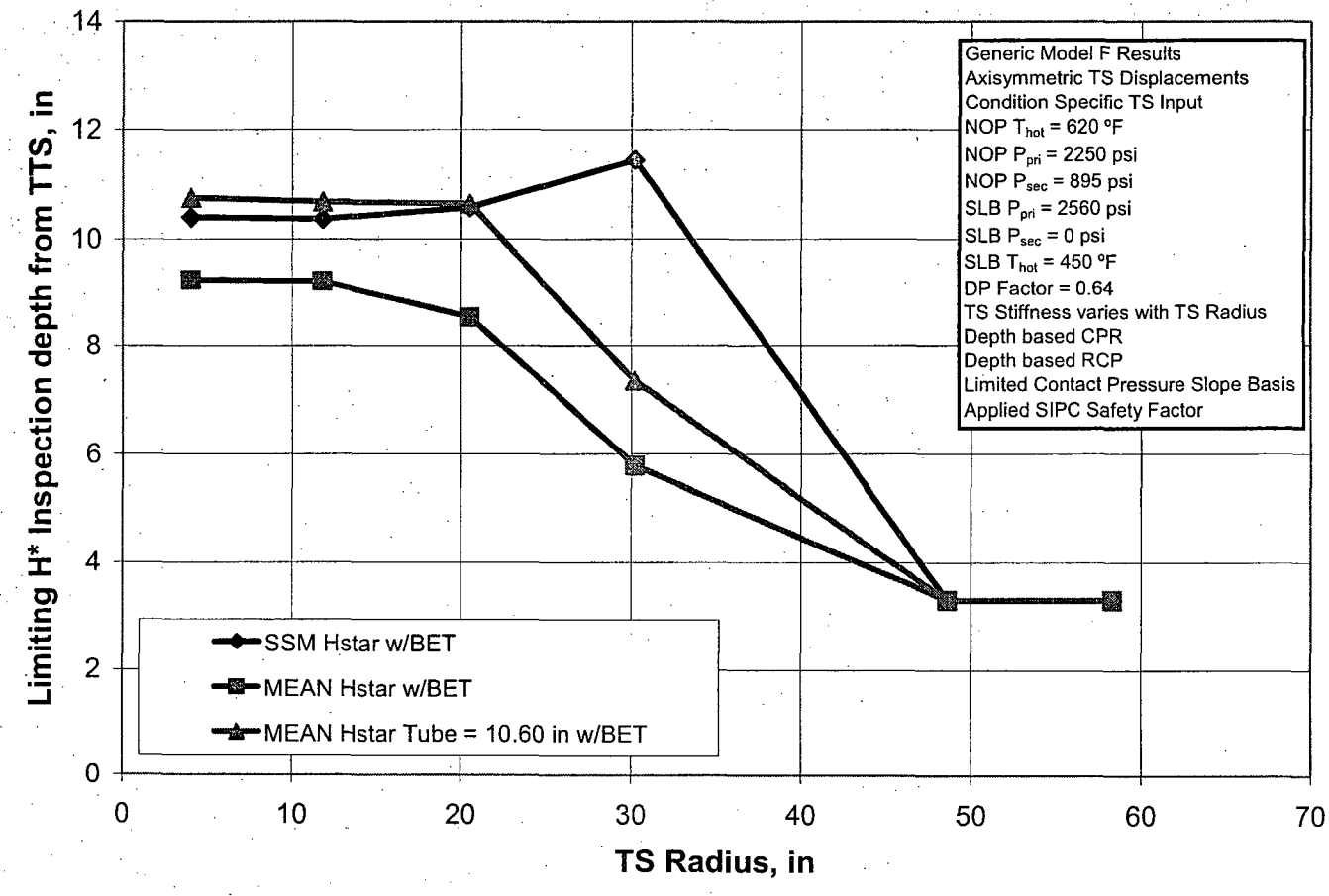
H* Results

- ALSP and 3D TS Displacement Compared
- Simplified Statistical Method (SSM) from Guidelines used to combine variability at 95/50 confidence level as a shortcut
- Minimum H* value of 3 in. is established to meet leakage requirements
- BET bounding value of 0.3 inch is conservatively added to minimum H* value

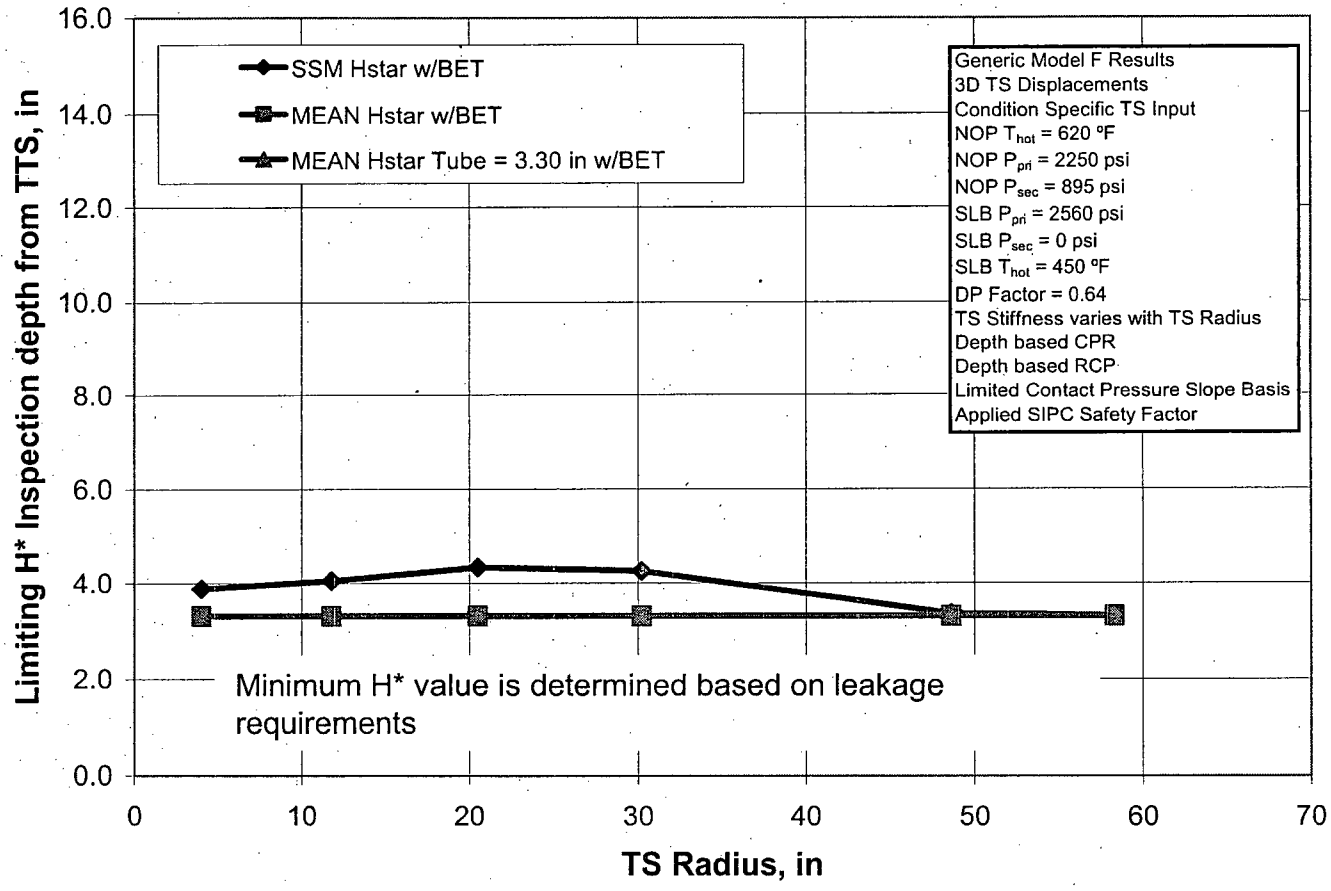
H* Results

- Bounding Model F Operating Conditions
- 6 Parameters in H* study:
  - Coefficient of Friction (μ = 0.3, 1σ = 0.1)
  - Residual Contact Pressure (μ = 200 lbf/in, 1σ = 50 lbf/in)
  - Tubesheet Young's Modulus (1σ = 0.85%)
  - Tube Young's Modulus (1σ = 0.85%)
  - Tubesheet CTE (1σ = 1.62%)
  - Tube CTE (1σ = 2.33%)
  - Linear RCP model assumed.
- Each parameter varied independently

ALSP Model H* Results

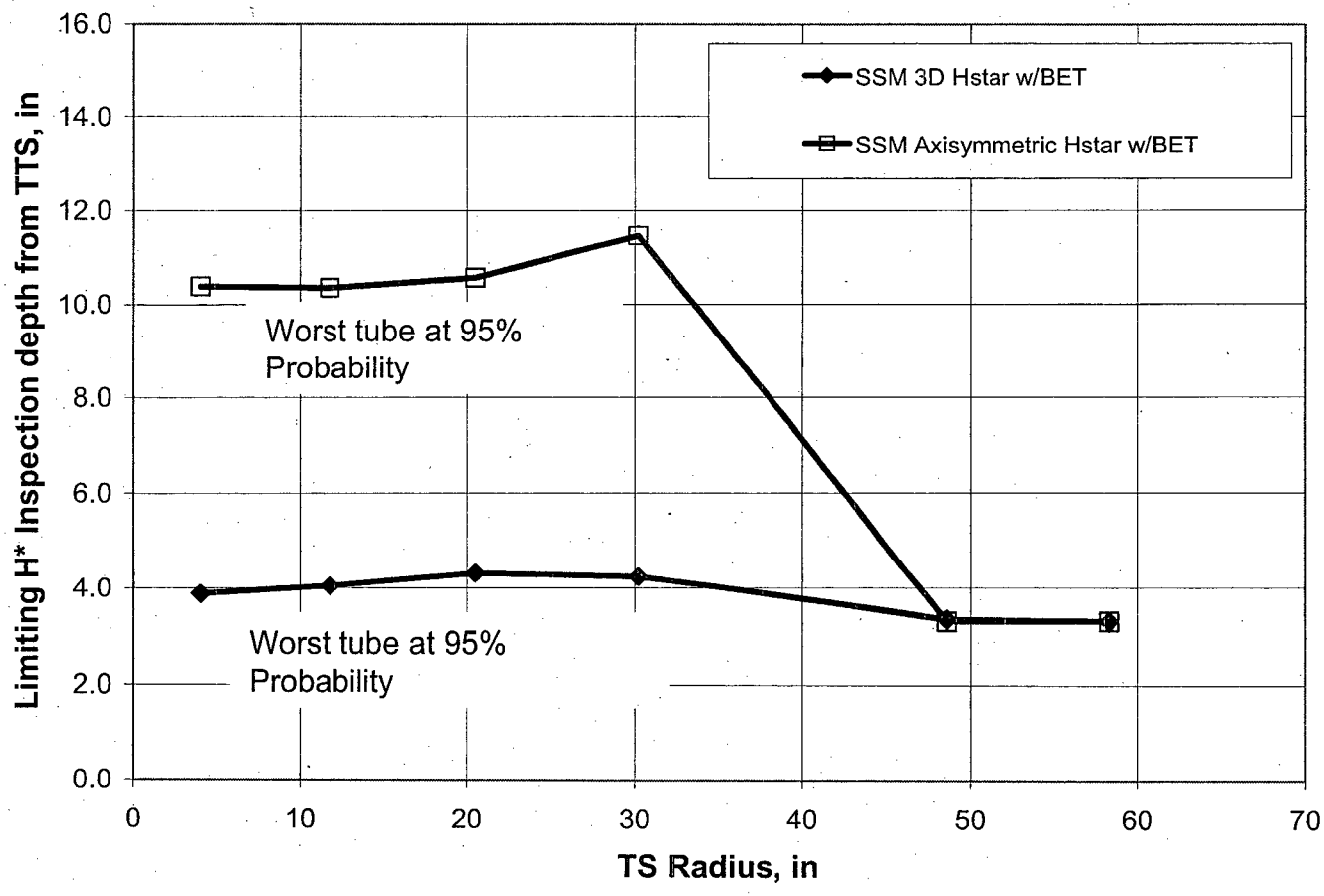


3D FEA H* Results

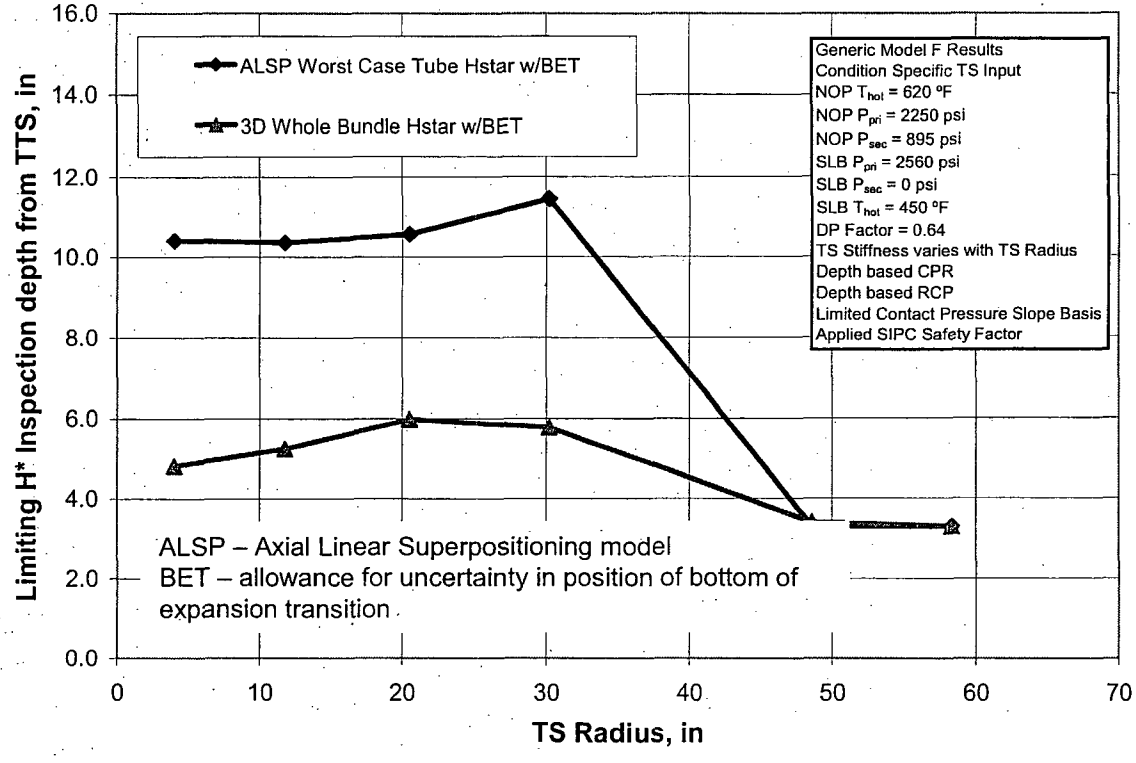


3D TS Displacements much less sensitive to variability

Comparison of 95% H* Results

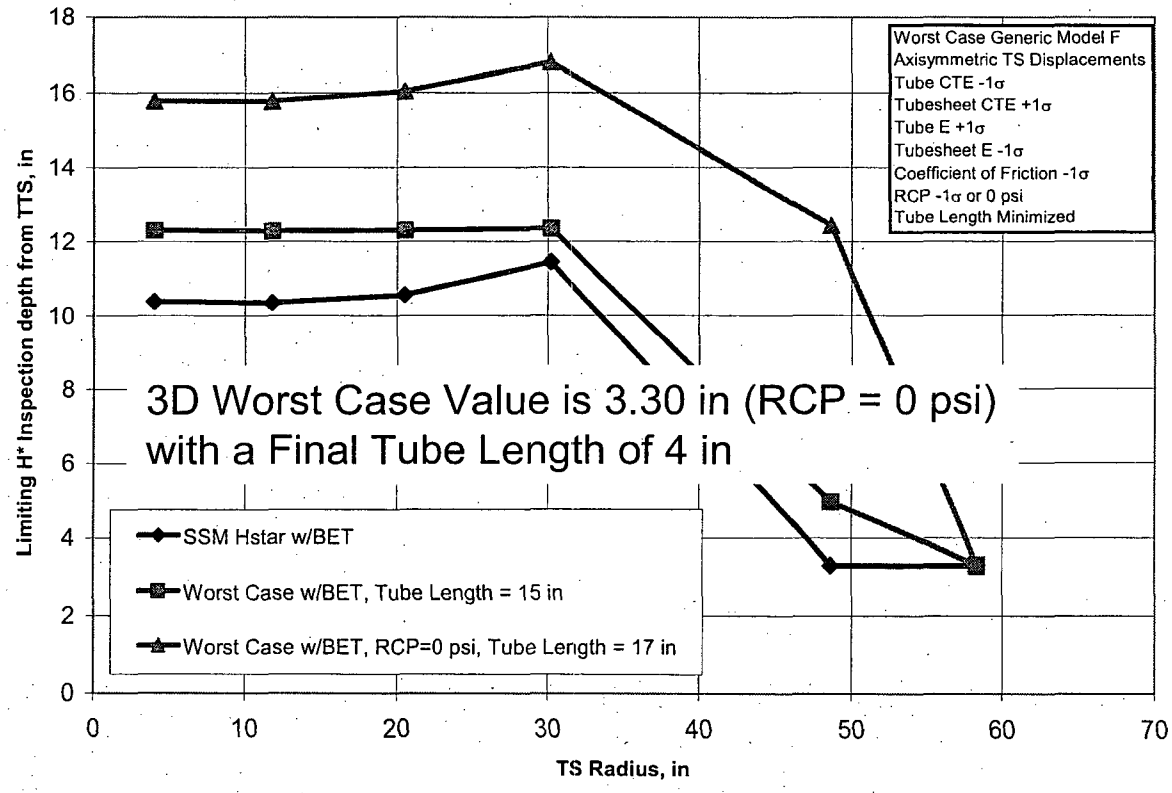


Current H* Results



- Contact pressure distributions for all operating cases compared and worst case results at each radius used for H*.
- Worst case tube H* defined using SIPC, limiting TS sector, worst condition result (ALSP).
- Whole bundle (95/50) analysis based on 5500 tubes in Model F bundle.

Comparison of Variability



Sensitivity of ALSP Results to Variation in Parameters Changes with respect to Tube Length and Value of RCP

Conclusions

- 3D FEA model is reference basis for H*
- Realistic boundary conditions applied
- Model of record, ALSP, is conservative with respect to 3D FEA
- Both H* analysis models are consistent with established theory and current literature and are supported by test data
- Crevice pressure model utilizes depth based approach
- Tubesheet radius dependent stiffness has little effect on H* but significant effect on uncertainties
- Residual contact pressure variability determined analytically; mean RCP tests in progress