SPED 2011 Technical Briefs

Pipe Stress for Pipers

Presented by David Diehl, P.E. - Intergraph

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Project Work Flow

- The Piping Designer handles most of the piping work
  - Positioning equipment
  - Sizing pipe
  - Routing pipe
  - Supporting weight
- The Piping Engineer steps in when required
  - Assuring safe design
  - Calculating equipment and component loads
  - Sizing supports

What the Designer Does/Can Do

- Size pipe (OD)
  - Based on process – flow rate, fluid, & pressure (drop)
- Select material
  - Based on fluid, service & temperature
- Specify insulation - temperature (drop)
- Set thickness/class
  - Based on material, temperature, pressure
  - Refer to ASME B31.3-2010 – Process Piping
    - Design pressure & temperature
      - 301.2 Design Pressure
      - 301.2.1 General
      - *(a)* The design pressure of each component in a piping
      - system shall be not less than the pressure at the most
      - severe condition of coincident internal or external pressure
      - and temperature (minimum or maximum) expected
      - during service, except as provided in para. 302.2.4.
      - *(b)* The most severe condition is that which results
      - in the greatest required component thickness and the
      - highest component rating.

What the Designer Does/Can Do

- Size pipe (OD)
  - Based on process – flow rate, fluid, & pressure (drop)
- Select material
  - Based on fluid, service & temperature
- Specify insulation - temperature (drop)
- Set thickness/class
  - Based on material, temperature, pressure
  - Refer to ASME B31.3-2010 – Process Piping
    - Design pressure & temperature
      - 301.3 Design Temperature
      - The design temperature of each component in a piping
      - system is the temperature at which, under the coincident
      - pressure, the greatest thickness or highest component
      - rating is required in accordance with para. 301.2. (To
      - satisfy the requirements of para. 301.2, different components
      - in the same piping system may have different
      - design temperatures.)

What the Designer Does/Can Do

- Size pipe (OD)
  - Based on proc
- Select material
  - Based on fluid
- Specify insulation
- Set thickness/cla
  - Based on mate
  - Refer to ASME
    - Design pr
    - Listed Co
      - PA
      - PR
      - 303
      - Co
      - liste
      - for
      - with

(10)

Table 326.1 Component Standards

Standard or Specification	Designation
Bolting	
Square and Hex Bolts and Screws (Inch Series)	ASME B18.2.1
Square and Hex Nuts (Inch Series)	ASME B18.2.2
Metallic Fittings, Valves, and Flanges	
Cast Iron Pipe Flanges and Flanged Fittings	ASME B16.1
Malleable Iron Threaded Fittings	ASME B16.3
Gray Iron Threaded Fittings	ASME B16.4
Pipe Flanges and Flanged Fittings	ASME B16.5
Factory-Made Wrought Steel Buttwelding Fittings	ASME B16.9
Face-to-Face and End-To-End Dimensions of Valves	ASME B16.10
Forged Fittings, Socket-Welding and Threaded	ASME B16.11
Ferrous Pipe Plugs, Bushings, and Locknuts With Pipe Threads	ASME B16.14
Cast Bronze Threaded Fittings, Class 125 and 250 [Notes (1), (2)]	ASME B16.15
Cast Copper Alloy Solder Joint Pressure Fittings	ASME B16.18
Wrought Copper and Copper Alloy Solder Joint Pressure Fittings	ASME B16.22
Cast Copper Alloy Pipe Flanges and Flanged Fittings: Classes 150, 300, 600, 900, 1500, and 2500	ASME B16.24
Cast Copper Alloy Fittings for Flared Copper Tubes	ASME B16.26
Valves-Flanged, Threaded, and Welding End	ASME B16.34
Orifice Flanges, Class 300, 600, 900, 1500, and 2500	ASME B16.36
Malleable Iron Threaded Pipe Unions, Class 150, 250, and 300	ASME B16.39
Ductile Iron Pipe Flanges and Flanged Fittings, Class 150 and 300	ASME B16.42
Large Diameter Steel Flanges, NPS 26 Through NPS 60	ASME B16.47
Steel Line Blanks	ASME B16.48
Bioprocessing Equipment [Note (3)]	ASME BPE
Flanged Steel Pressure-Relief Valves	API 526
Check Valves: Flanged, Lug, Wafer and Butt-welding	API 594
Metal Plug Valves—Flanged, Threaded, and Welding Ends	API 599
Bolted Bonnet Steel Gate Valves for Petroleum and Natural Gas Industries	API 600
Steel Gate, Globe, and Check Valves for Sizes DN 100 and Smaller for the Petroleum and Natural Gas Industries	API 602
Corrosion-Resistant, Bolted Bonnet Gate Valves — Flanged and Butt-Welding Ends	API 603

What the Designer Does/Can Do

- Size pipe (OD)
  - Based on process – flow rate, fluid, & pressure (drop)
- Select material
  - Based on fluid, service & temperature
- Specify insulation - temperature (drop)
- Set thickness/class
  - Based on material, temperature, pressure
  - Refer to ASME B31.3-2010 – Process Piping
    - Design pressure & temperature
    - Listed Components
    - Straight pipe
      - 304 PRESSURE DESIGN OF COMPONENTS
      - 304.1 Straight Pipe
      - 304.1.1 General
      - *(a)* The required thickness of straight sections of pipe
      - shall be determined in accordance with eq. (2):
        - $t_m = t + c$ (2)
      - The minimum thickness, *T*, for the pipe selected, considering
      - manufacturer's minus tolerance, shall be not
      - less than t_m.

$$t = \frac{PD}{2(SEW + PY)}$$

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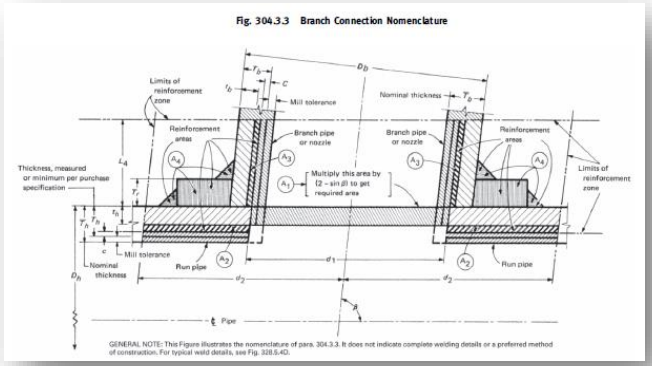
- Set thickness/class
  - Based on material, temperature, pressure
  - Refer to ASME B31.3-2010 – Process Piping
    - Design pressure & temperature
    - Listed Components
    - Straight pipe
    - Fabricated branch connections
      - 304.3.3 Reinforcement of Welded Branch Connections.
      - Added reinforcement is required to meet the
      - criteria in paras. 304.3.3(b) and (c) when it is not inherent
      - in the components of the branch connection.



What the Designer Does/Can Do

- Route pipe
  - Pressure drop / general hydraulics
  - Serviceability
  - Vents & drains or slope

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  - Refer to ASME B31.1-2010 – Power Piping

Table 121.5 Suggested Pipe Support Spacing

Nominal Pipe Size, NPS	Suggested Maximum Span			
	Water Service		Steam, Gas, or Air Service	
	ft	m	ft	m
1	7	2.1	9	2.7
2	10	3.0	13	4.0
3	12	3.7	15	4.6
4	14	4.3	17	5.2
6	17	5.2	21	6.4
8	19	5.8	24	7.3
12	23	7.0	30	9.1
16	27	8.2	35	10.7
20	30	9.1	39	11.9
24	32	9.8	42	12.8

GENERAL NOTES:

(a) Suggested maximum spacing between pipe supports for horizontal straight runs of standard and heavier pipe at maximum operating temperature of 750°F (400°C).

(b) Does not apply where span calculations are made or where there are concentrated loads between supports, such as flanges, valves, specialties, etc.

(c) The spacing is based on a fixed beam support with a bending stress not exceeding 2,300 psi (15.86 MPa) and insulated pipe filled with water or the equivalent weight of steel pipe for steam, gas, or air service, and the pitch of the line is such that a sag of 0.1 in. (2.5 mm) between supports is permissible.

What the Designer Does/Can Do

- Route pipe
  - Pressure drop /
  - Serviceability
  - Vents & drains
- Support pipe dead
  - Rules based
  - Refer to ASME
  - or MSS SP-69

TABLE 3. MAXIMUM HORIZONTAL PIPE HANGER AND SUPPORT SPACING

NOMINAL PIPE OR TUBE SIZE	1 STD WT STEEL PIPE WATER SERVICE ft	1 STD WT STEEL PIPE WATER SERVICE m	2 STD WT STEEL PIPE VAPOR SERVICE ft	2 STD WT STEEL PIPE VAPOR SERVICE m	3 COPPER TUBE WATER SERVICE ft	3 COPPER TUBE WATER SERVICE m	4 COPPER TUBE VAPOR SERVICE ft	4 COPPER TUBE VAPOR SERVICE m	5 FIRE PROTECTION	6 CAST IRON & DUCTILE IRON PRESSURE	7 CAST IRON SOIL	8 ASBESTOS CEMENT	9 GLASS	10 PLASTIC	11 FIBERGLASS REINFORCED
1/4	7	2.1	8	2.4	5	1.5	5	1.5	FOLLOW REQUIREMENTS OF THE NATIONAL FIRE PROTECTION ASSOCIATION. SEE SECTION 14.	12 ft (3.7m) MAX SPACING MIN OF ONE (1) HANGER PER PIPE SECTION CLOSE TO JOINT ON THE BARREL. ALSO AT CHANGE OF DIRECTION AND BRANCH CONNECTIONS.	10 ft (3.0m) MAX SPACING MIN OF ONE (1) HANGER PER PIPE SECTION CLOSE TO JOINT ON THE BARREL. ALSO AT CHANGE OF DIRECTION AND BRANCH CONNECTIONS.	FOLLOW PIPE MANUFACTURER'S RECOMMENDATIONS.	8 ft (2.4m) MAX SPACING. FOLLOW MANUFACTURER'S RECOMMENDATIONS. SEE SECTION 18.	FOLLOW PIPE MANUFACTURER'S RECOMMENDATIONS FOR MATERIAL AND SERVICE TEMPERATURE.	FOLLOW PIPE MANUFACTURER'S RECOMMENDATIONS FOR SPACING AND SERVICE CONDITION.
3/8	7	2.1	8	2.4	5	1.5	6	1.8							
1/2	7	2.1	8	2.4	5	1.5	6	1.8							
3/4	7	2.1	9	2.7	5	1.5	7	2.1							
1	7	2.1	9	2.7	6	1.8	8	2.4							
1 1/4	7	2.1	9	2.7	7	2.1	9	2.7							
1 1/2	9	2.7	12	3.7	8	2.4	10	3.0							
2	10	3.0	13	4.0	8	2.4	11	3.4							
2 1/2	11	3.4	14	4.3	9	2.7	13	4.0							
3	12	3.7	15	4.6	10	3.0	14	4.3							
3 1/2	13	4.0	16	4.9	11	3.4	15	4.6							
4	14	4.3	17	5.2	12	3.7	16	4.9							
5	16	4.9	19	5.8	13	4.0	18	5.5							
6	17	5.2	21	6.4	14	4.3	20	6.1							
8	19	5.8	24	7.3	16	4.9	23	7.0							
10	22	6.1	26	7.9	18	5.5	25	7.6							
12	23	7.0	30	9.1	19	5.8	28	8.5							
14	25	7.6	32	9.8											
16	27	8.2	35	10.7											
18	28	8.5	37	11.3											
20	30	9.1	39	11.9											
24	32	9.8	42	12.8											
30	33	10.1	44	13.4											

NOTE: (1) FOR SPACING SUPPORTS INCORPORATING TYPE 40 SHIELDS, SEE TABLE 5.
(2) DOES NOT APPLY WHERE SPAN CALCULATIONS ARE MADE OR WHERE THERE ARE CONCENTRATED LOADS BETWEEN SUPPORTS SUCH AS FLANGES, VALVES, SPECIALTIES, ETC., OR CHANGES IN DIRECTION REQUIRING ADDITIONAL SUPPORTS.

MSS STANDARD PRACTICE SP-69

What the Designer Does/Can Do

- Route pipe
  - Pressure drop / general hydraulics
  - Serviceability
  - Vents & drains or slope
- Support pipe deadweight
  - Rules based
  - Refer to ASME B31.1-2010 – Power Piping
  - or MSS SP-69
  - Our suggested 4 steps:
    - Support concentrated loads (valves, etc.)
    - Use maximum span spacing (L) on horizontal straight runs; use ¾ L on horizontal runs with bends
    - Support risers at one or more locations, preferring locations above center of gravity
    - Utilize available steel

But what about hot pipe?

- Effects of thermal strain can be significant
  - Equipment load / alignment
  - Piping fatigue failure over time
- Example
  - Steel pipe grows about 1 inch per every 100 F temperature increase
    - 12 inch pipe at 350F, locked between two anchors, will exert a load of 800,000 lbf on those two anchors, or buckle

$F = kx$ $\quad A = {}^{\pi}/_{4}\,(OD^2 - (OD - 2t)^2);\ OD = 12.75, t = .375$

$k = AE/L$ $\quad A = 14.579 \quad E = 29.5 * 10^6 \quad \alpha = 1.879 * 10^{-3}$

$x = \alpha L$ $\quad F = 14.579 * 29.5 * 10^6 * 1.879 * 10^{-3}$

$F = AE\alpha$ $\quad F = 808000\ lbf$

But what about hot pipe?

- Effects of thermal strain can be significant
  - Equipment load / alignment
  - Piping fatigue failure over time
- Example
  - Steel pipe grows about 1 inch per e
    - 12 inch pipe at 650F, locked betv
    - those two anchors or buckle
- Some lines can be checked by rule
  - Reference the B31.3 Rule

319.4 Flexibility Analysis

319.4.1 Formal Analysis Not Required. No formal analysis of adequate flexibility is required for a piping system which

(a) duplicates, or replaces without significant change, a system operating with a successful service record

(b) can readily be judged adequate by comparison with previously analyzed systems

(c) is of uniform size, has no more than two points of fixation, no intermediate restraints, and falls within the limitations of empirical eq. (16)[9]

$$\frac{Dy}{(L-U)^2} \le K_1 \qquad (16)$$

where

D = outside diameter of pipe, mm (in.)
E_a = reference modulus of elasticity at 21°C (70°F), MPa (ksi)
K_1 = 208 000 S_A/E_a, $(mm/m)^2$
= 30 S_A/E_a, $(in./ft)^2$
L = developed length of piping between anchors, m (ft)
S_A = allowable displacement stress range in accordance with eq. (1a), MPa (ksi)
U = anchor distance, straight line between anchors, m (ft)
y = resultant of total displacement strains, mm (in.), to be absorbed by the piping system

But what about hot pipe?

- Effects of thermal strain can be significant
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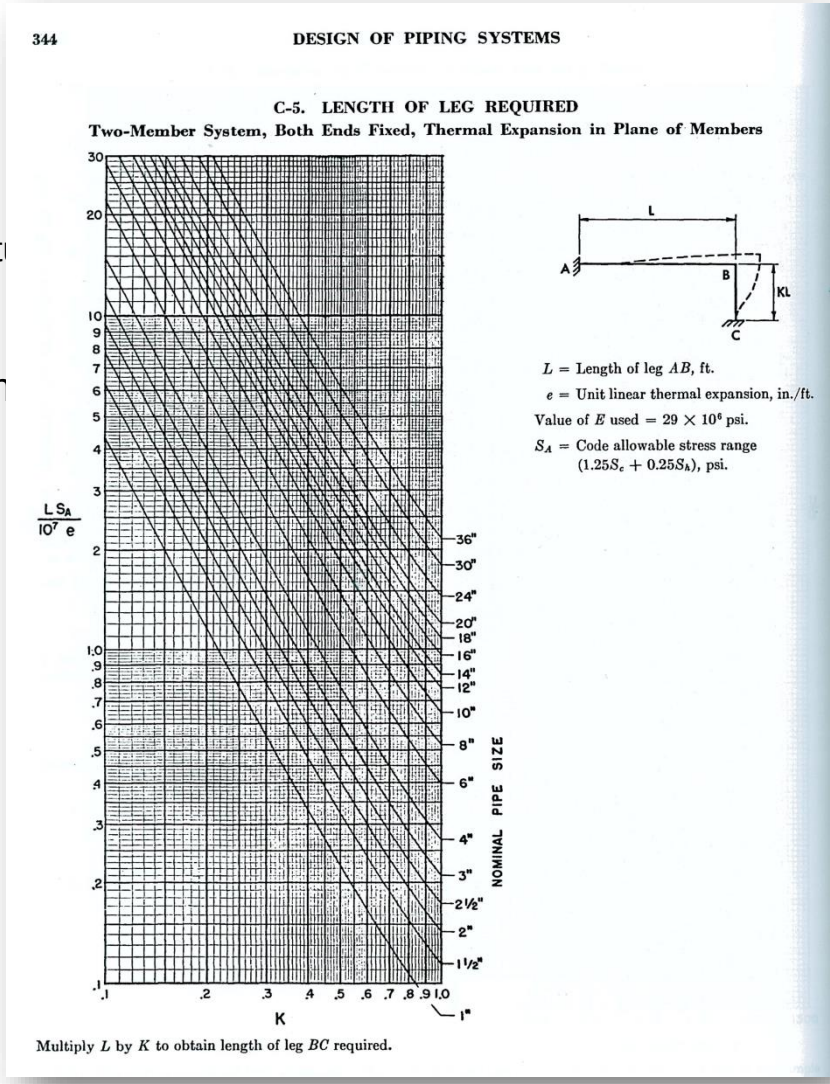
  - Piping fatigue failure over time
- Example
  - Steel pipe grows about 1 inch per every 100 F temperat
    - 12 inch pipe at 650F, locked between two anchors, will buckle
- Some lines can be checked by rule or simplified meth
  - Reference the B31.3 Rule
  - Reference the Kellogg Chart Methods
    Design of Piping Systems, M. W. Kellogg Company

Stress:



344 DESIGN OF PIPING SYSTEMS

C-5. LENGTH OF LEG REQUIRED

Two-Member System, Both Ends Fixed, Thermal Expansion in Plane of Members

Multiply L by K to obtain length of leg BC required.

But what about hot pipe?

- Effects of thermal strain can be significant
  - Equipment load / alignment
  - Piping fatigue failure over time
- Example
  - Steel pipe grows about 1 inch per every 100 F temperature
    - 12 inch pipe at 650F, locked between two anchors, will ex
      buckle
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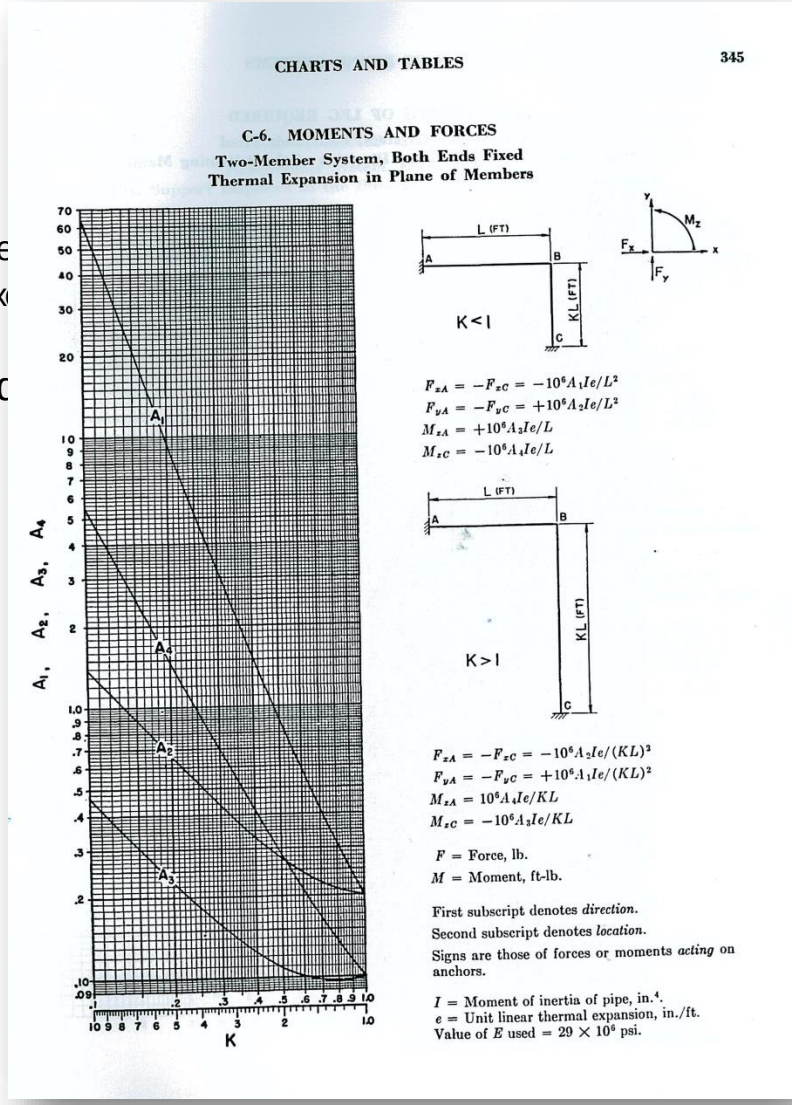
Load:



CHARTS AND TABLES 345

C-6. MOMENTS AND FORCES

Two-Member System, Both Ends Fixed
Thermal Expansion in Plane of Members

A_1, A_2, A_3, A_4

K

L (FT)

KL (FT)

K < 1

$F_{xA} = -F_{xC} = -10^6 A_1 Ie/L^2$
$F_{yA} = -F_{yC} = +10^6 A_2 Ie/L^2$
$M_{xA} = +10^6 A_3 Ie/L$
$M_{xC} = -10^6 A_4 Ie/L$

K > 1

$F_{xA} = -F_{xC} = -10^6 A_2 Ie/(KL)^2$
$F_{yA} = -F_{yC} = +10^6 A_1 Ie/(KL)^2$
$M_{xA} = 10^6 A_4 Ie/KL$
$M_{xC} = -10^6 A_3 Ie/KL$

F = Force, lb.
M = Moment, ft-lb.

First subscript denotes *direction*.
Second subscript denotes *location*.
Signs are those of forces or moments *acting* on anchors.

I = Moment of inertia of pipe, in.^{4}.
e = Unit linear thermal expansion, in./ft.
Value of E used = 29×10^6 psi.

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  - Equipment load / alignment
  - Piping fatigue failure over time
- Example
  - Steel pipe grows about 1 inch per every 100 F temperature increase
    - 12 inch pipe at 650F, locked between two anchors, will exert a load of 800,000 lbf on those two anchors or buckle
- Some lines can be checked by rule or simplified methods
  - Reference the B31.3 Rule
  - Reference the Kellogg Chart Methods
- Because of the interaction of thermal growth and piping layout, most humans cannot predict the effects of thermal strain in piping systems

Critical Line List – the handoff for ensuring safe design

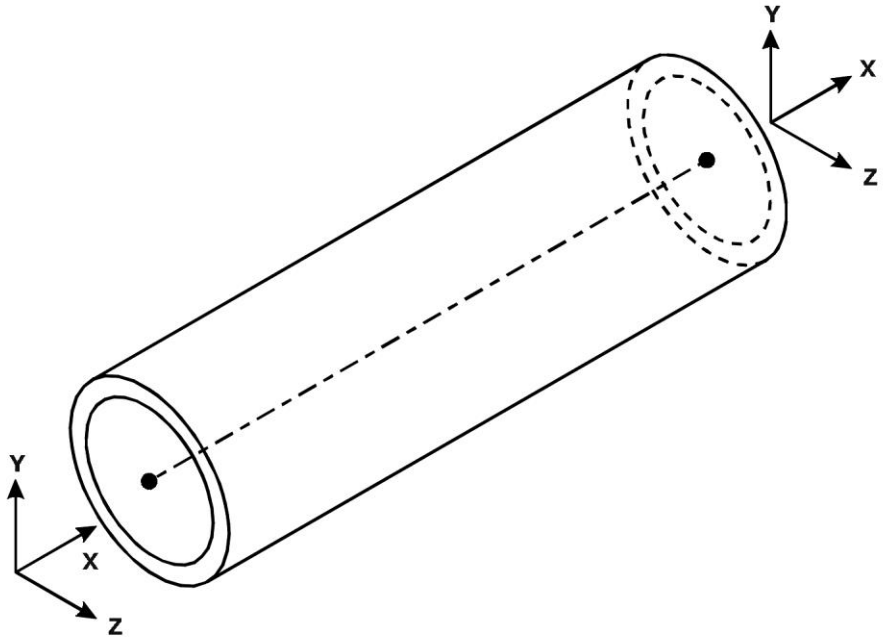
- Piping designers are usually equipped with a Critical Line List to determine which lines need checking
- A simple check: OD*Delta T>1450

Critical Line List – the handoff for ensuring safe design

- A sample Critical Line List - (Introduction to Pipe Stress Analysis by Sam Kannappan, P.E., ABI Enterprises, Inc, 2008)
  - Lines 3 inch and larger that are:
    - connected to rotating equipment
    - subject to differential settlement of connected equipment and/or supports, or
    - with temperatures less than 20F
  - Lines connected to reciprocating equipment such as suction and discharge lines to and from reciprocating compressors
  - Lines 4 inch and larger connected to air coolers, steam generators, or fired heater tube sections
  - Lines 6 in. and larger with temperatures of 250 F and higher
  - All lines with temperatures of 600 F and higher
  - Lines 16 in. and larger
  - All alloy lines
  - High pressure lines (over 2000 psi). Although systems over 1500 psi are sometimes a problem, particularly with restraint arrangements
  - Lines subject to external pressure
  - Thin-walled pipe or duct of 18 in. diameter and over, having an outside diameter over wall thickness ratio (d/t) of more than 90
  - Lines requiring proprietary expansion devices, such as expansion joints and Victaulic couplings
  - Underground process lines. Pressures >1000 psi in underground piping inevitably generates high thrust forces, even at very low expansion temperature differentials. Attention is required on burial techniques, changes in direction, ground entry/exit, or connection to equipment or tanks. Other examples include pump/booster stations, terminals, meter stations and scraper traps
  - Internally lined process piping & jacketed piping
  - Lines in critical service
  - Pressure relief systems. Also relief valve stacks with an inlet pressure greater than 150 psig
  - Branch line tie-ins of matched size, particularly relief systems tied together or large, branch piping of similar size as piping being connected

Engineers will use a piping program to evaluate pipe stress and collect other important data

- Piping program represents pipe as a simple beam element that can bend (rather than do other things)



- This beam shows the interaction of forces and moments that load the system and the displacements and rotations of the beam ends

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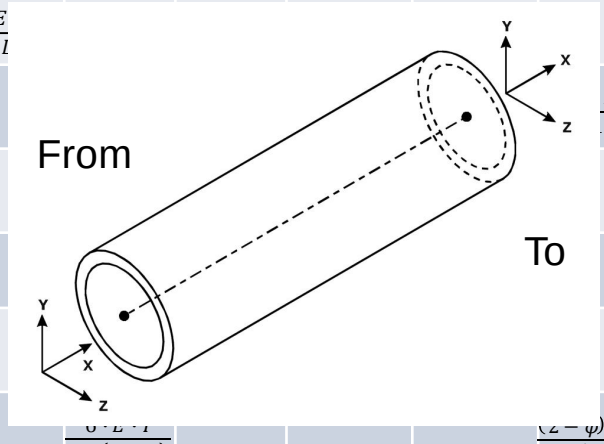
- This interaction is represented by the beam (pipe) stiffness (the k in F=kx)

The stiffness matrix for a pipe element

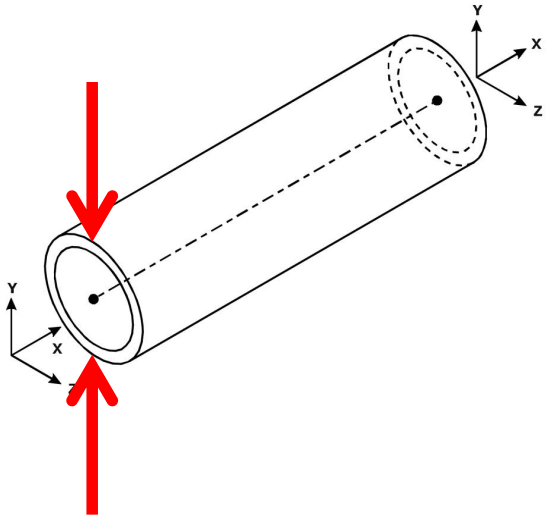
		"From"						"To"					
		X	Y	Z	RX	RY	RZ	X	Y	Z	RX	RY	RZ
"From"	X	$\frac{E \cdot A}{L}$						$\frac{-E}{L}$					
	Y		$\frac{12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$				$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$						$\frac{E \cdot I}{+\varphi)}$
	Z			$\frac{12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$		$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$							
	RX				$\frac{2 \cdot G \cdot I}{L}$								
	RY			$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$		$\frac{(4+\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$							
	RZ		$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$				$\frac{(4+\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$		$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$				$\frac{(2-\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$
"To"	X	$\frac{-E \cdot A}{L}$						$\frac{E \cdot A}{L}$					
	Y		$\frac{-12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$				$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$		$\frac{12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$				$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$
	Z			$\frac{-12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$		$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$				$\frac{12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$		$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$	
	RX				$\frac{-2 \cdot G \cdot I}{L}$						$\frac{2 \cdot G \cdot I}{L}$		
	RY			$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$		$\frac{(2-\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$				$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$		$\frac{(4+\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$	
	RZ		$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$				$\frac{(2-\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$		$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$				$\frac{(4+\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$



Engineers will use a piping program to evaluate pipe stress and collect other important data

- Piping program represents pipe as a simple beam element that can bend (rather than do other things)
- This beam shows the interaction of forces and moments that load the system and the displacements and rotations of the beam ends
- This interaction is represented by the beam (pipe) stiffness (the k in F=kx)
- The user includes the piping supports and restraints in this stiffness model

		"From"					
		X	Y	Z	RX	RY	RZ
"From"	X	$\frac{E \cdot A}{L}$					
	Y		$\frac{12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)} + 10^{12}$				$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$
	Z			$\frac{12 \cdot E \cdot I}{L^3 \cdot (1+\varphi)}$		$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$	
	RX				$\frac{2 \cdot G \cdot I}{L}$		
	RY			$\frac{-6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$		$\frac{(4+\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$	
	RZ		$\frac{6 \cdot E \cdot I}{L^2 \cdot (1+\varphi)}$				$\frac{(4+\varphi) \cdot E \cdot I}{L \cdot (1+\varphi)}$



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- This beam shows the interaction of forces and moments that load the system and the displacements and rotations of the beam ends
- This interaction is represented by the beam (pipe) stiffness (the k in F=kx)
- The user includes the piping supports and restraints in this stiffness model
- Piping loads (such as pipe weight, thermal strain, wind load, etc.) populate the load vector (the F in F=kx)

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- This beam shows the interaction of forces and moments that load the system and the displacements and rotations of the beam ends
- This interaction is represented by the beam (pipe) stiffness (the k in F=kx)
- The user includes the piping supports and restraints in this stiffness model
- Piping loads (such as pipe weight, thermal strain, wind load, etc.) populate the load vector (the F in F=kx)
- With the system k and the several F's, the program solves for the system position under load (the x in F=kx)

While commonly called a pipe stress program, stress is only one part of the value in these packages

- Those displacements are important
  - In checking for clash
  - In checking pipe position (sag, support liftoff)
- As are system forces and moments
  - In sizing supports and restraints
  - In checking flange loads
  - In evaluating equipment loads

The engineer's task

- Convert the system "analog" into a digital model used by the program
  - Analog can be a sketch, a stress isometric, a concept
  - There can be several competing interpretations of this analog-to-digital conversion – this is where the subtleties of F=kx come in play
- Set the loads to be evaluated
  - The F in F=kx
  - System in operation, system at startup, anticipated upsets
- Establish the evaluation criteria for the analysis
  - Equipment loads from industry standards
    - Pumps, compressors, turbine, heaters
  - System deflections limits by company standards or industry guidelines
    - Max sag, slide limits
  - Pipe stress from the Piping Code
- Review the results and resolve any design deficiencies
  - First, verify the model and applied loads
  - Compare displacements, loads, and stresses to their allowable limits.
  - Test proposed "fixes" to resolve problems
  - Here, too, an understanding of the model operation (F=kx) is quite helpful in diagnosing and fixing problems
  - Send proposed changes back to the designer for approval

So what are these stresses?

- What is stress?
  - Used here, stress is a measure of the pipe's ability to carry the required load
  - But there are different criteria for stress limits
- Stress can be used to predict system collapse
  - Caused by piping loads that can cause system failure by material yield
  - Gravity loads, pressure, wind loads are typical (force-based) loads evaluated in this manner
- Stress can also be used to predict the formation of a through-the-wall crack over time
  - These are fatigue failures are caused by repeated load cycling
  - This stress is measured by the changing stress from installation to operating position
  - Thermal strain of the piping and the (hot-to-cold) motion of piping connections (e.g. vessel nozzle connections) are typical (strain-based) loads evaluated in this manner

But these predicted stresses cannot be measured in the "real world"

- These are (Piping) Code-defined stress calculations
- Stress equations have evolved over the years to allow a standard, simplified evaluation of the piping system safety
- Many piping components have a load multiplier (the Stress Intensification Factor or SIF) to increase the calculated stress
  - To incorporate weakness of the component (e.g. an elbow or tee) under load
  - Without changing the material-based, allowable stress limit
- Many piping codes do not evaluate the state of stress in the operating condition

Here are the B31.3 stress equations

- Let $S_b = \sqrt{(i_i M_i)^2 + (i_o M_o)^2}/Z$ and $S_t = T/2Z$
- Collapse
  - Longitudinal stress due to sustained loads:

$$S_L = S_{lp} + F_{ax}/A + S_b \leq S_h$$

  - Longitudinal stress due to sustained loads and occasional loads:

$$S_{lp} + (F_{ax}/A + S_b)_{sus} + (F_{ax}/A + S_b)_{occ} \leq 1.33 S_h$$

- Fatigue
  - Expansion stress range:

$$\sqrt{\left({S_b}^2 + {S_t}^2\right)} \leq f[1.25(S_c + S_h) - S_L]$$

-or-

$$\sqrt{\left({S_b}^2 + {S_t}^2\right)} \leq f(1.25 S_c + 0.25 S_h)$$

B31.3 also mentions structural response

- Stress is not the only concern here:
- Loads:

319.5 Reactions

Reaction forces and moments to be used in design of restraints and supports for a piping system, and in evaluating the effects of piping displacements on connected equipment, shall be based on the reaction range R for the extreme displacement conditions, considering the temperature range defined in para. 319.3.1(b), and using E_a. The designer shall consider instantaneous maximum values of forces and moments in the original and extreme displacement conditions (see para. 319.2.3), as well as the reaction range, in making these evaluations.

319.5.2 Maximum Reactions for Complex Systems.

For multianchor piping systems and for two-anchor systems with intermediate restraints, eqs. (21) and (22) are not applicable. Each case must be studied to estimate location, nature, and extent of local overstrain, and its effect on stress distribution and reactions.

B31.3 also mentions structural response

- Stress is not the only concern here:
- Displacements:

319.6 Calculation of Movements

Calculations of displacements and rotations at specific locations may be required where clearance problems are involved. In cases where small-size branch pipes attached to stiffer run pipes are to be calculated separately, the linear and angular movements of the junction point must be calculated or estimated for proper analysis of the branch.

Let’s take a look at a Pipe Flexibility and Stress Analysis Program

CAESAR II

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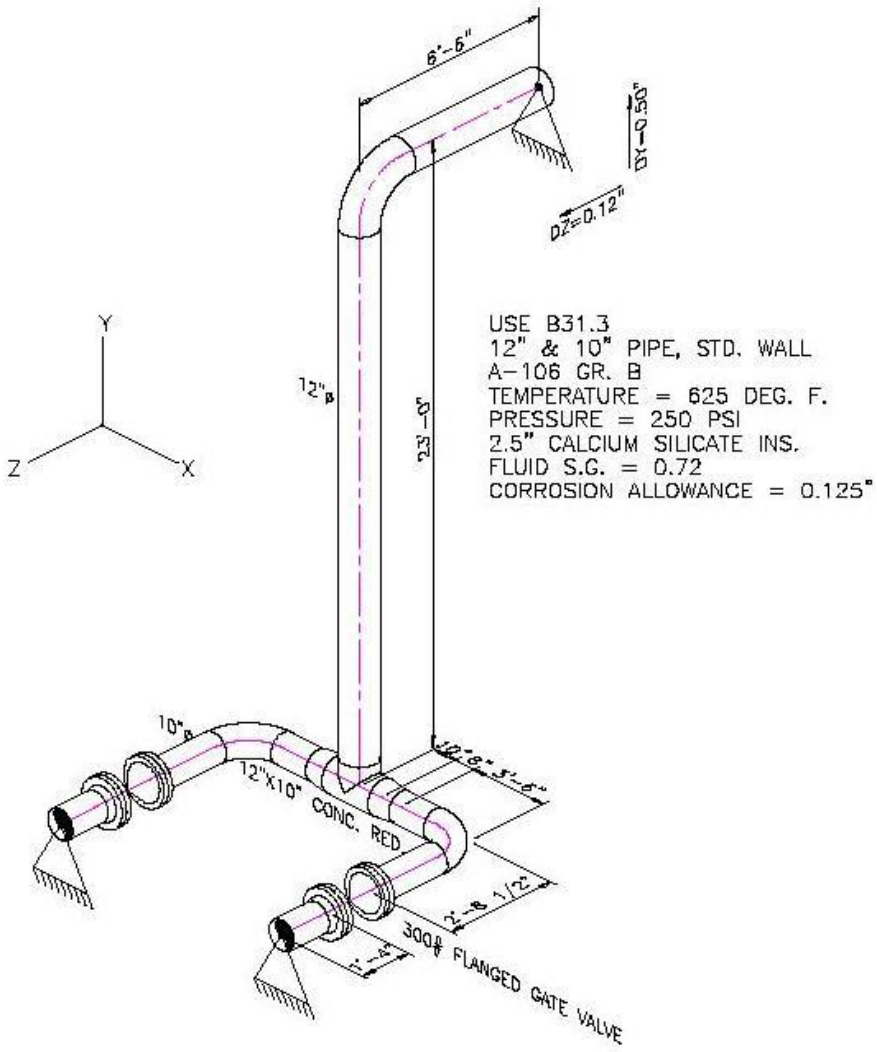
CAESAR II input session

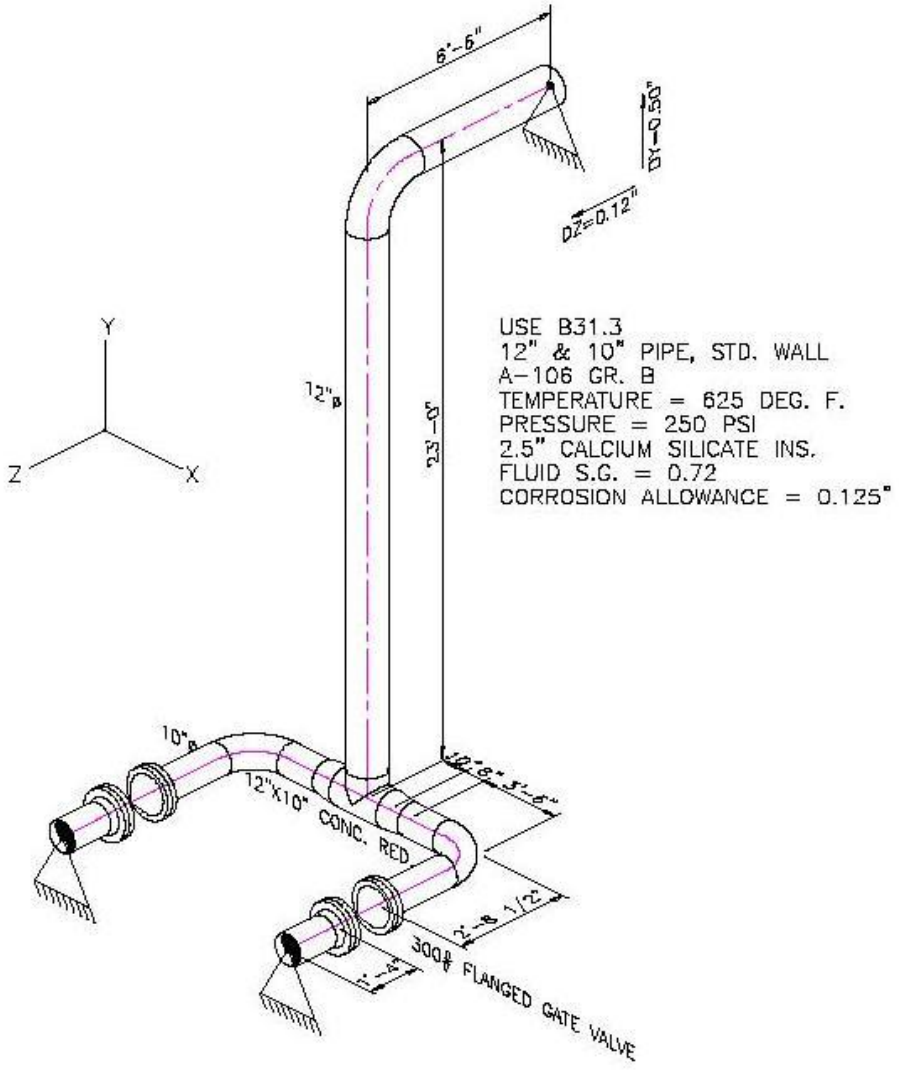
- Preparing the drawing
- Building the model
- Setting the loads

Example

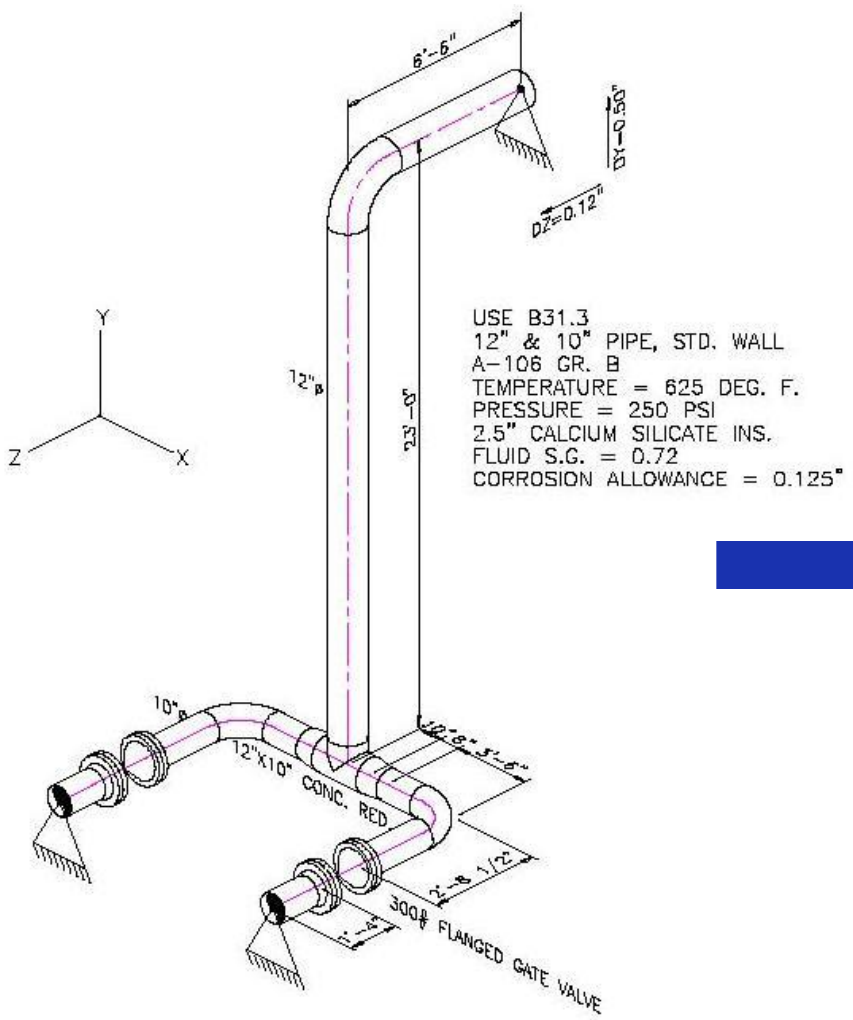

6'-6"
12"ø
Y
Z
X
USE B31.3
12" & 10" PIPE, STD. WALL
A-106 GR. B
TEMPERATURE = 625 DEG. F.
PRESSURE = 250 PSI
2.5" CALCIUM SILICATE INS.
FLUID S.G. = 0.72
CORROSION ALLOWANCE = 0.125"
10"ø
12"X10" CONC. RED
300# FLANGED GATE VALVE

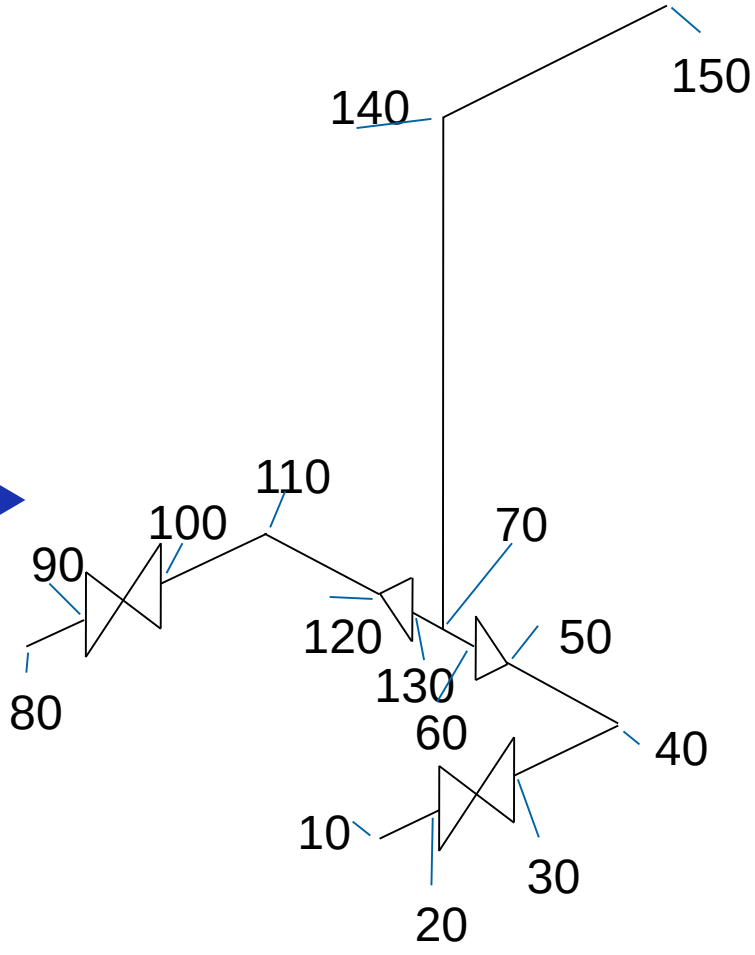
Collect & Digitize Data

- Pipe layout
- Boundary conditions
- Loads
- Stress criteria
- Node numbers

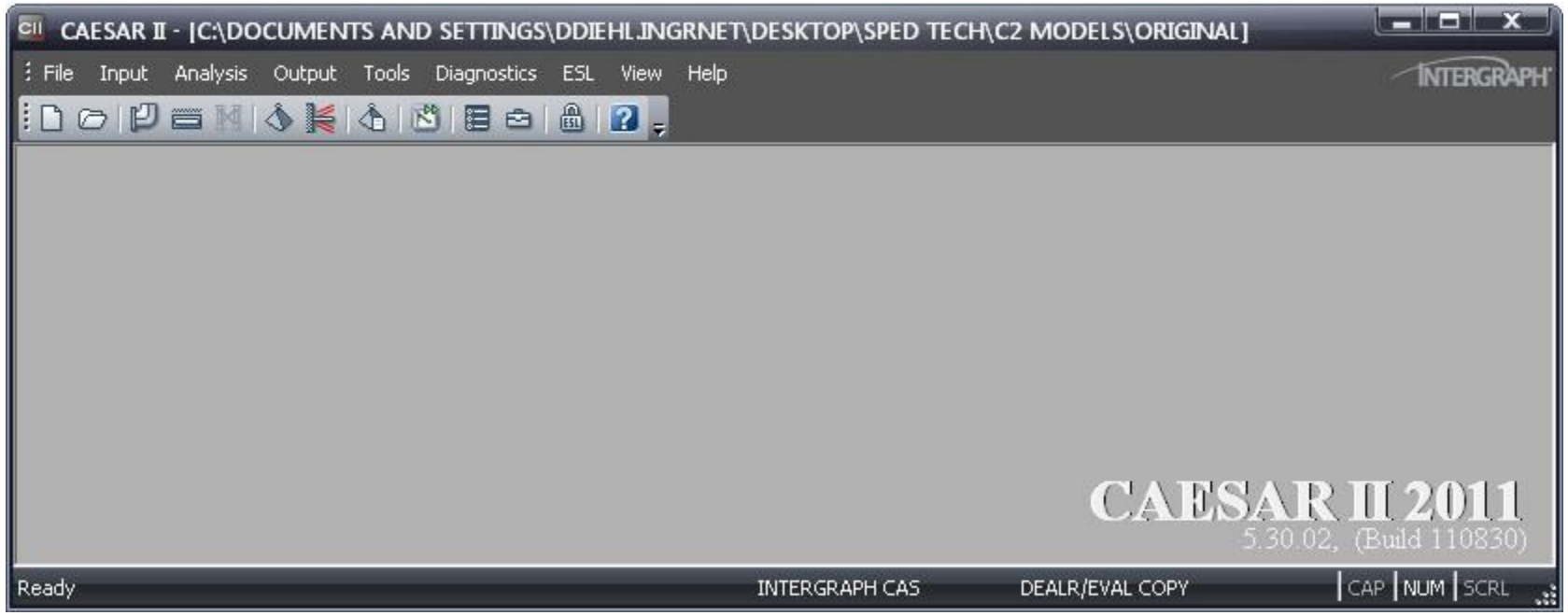


Assign Nodes



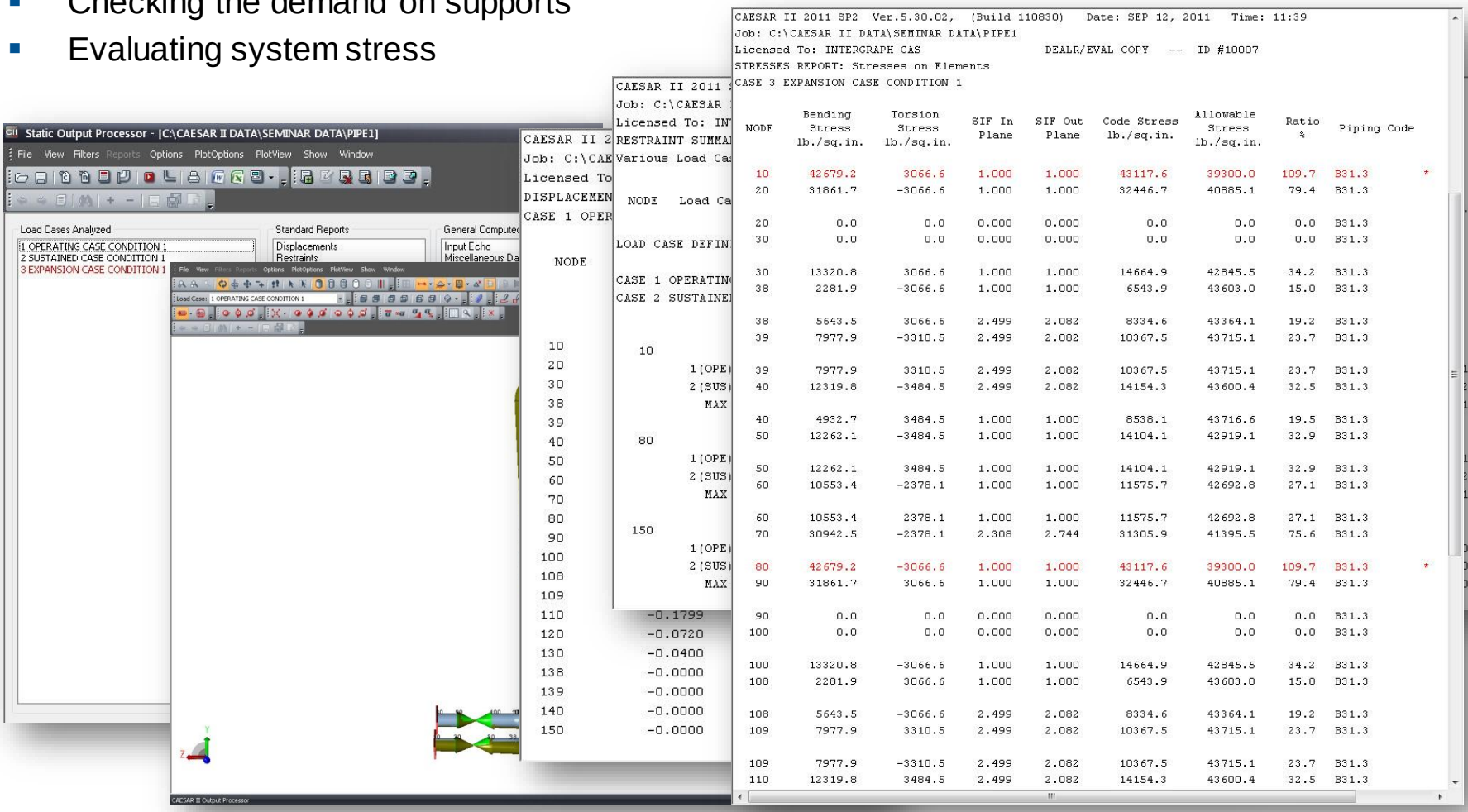


Start CAESAR II



CAESAR II results review

- Checking the model
- Reviewing the system deflections in the operating position
- Checking the demand on supports
- Evaluating system stress



Additional system checks that may control design

- Flange screening

Maximum allowable non-shock pressure *(psig)* and temperature ratings for steel pipe flanges and flanged fittings according the American National Standard ANSI B16.5 - 1988.

From: http://www.engineeringtoolbox.com/ansi-flanges-pressure-temperature-d_342.html

Maximum Allowable non-shock Pressure *(psig)*

Temp *(°F)*	Pressure Class *(lb)*						
	150	300	400	600	900	1500	2500
	Hydrostatic Test Pressure *(psig)*						
	450	1125	1500	2225	3350	5575	9275
-20 to 100	285	740	990	1480	2220	3705	6170
200	260	675	900	1350	2025	3375	5625
300	230	655	875	1315	1970	3280	5470
400	200	635	845	1270	1900	3170	5280
500	170	600	800	1200	1795	2995	4990
600	140	550	730	1095	1640	2735	4560
650	125	535	715	1075	1610	2685	4475
700	110	535	710	1065	1600	2665	4440
750	95	505	670	1010	1510	2520	4200
800	80	410	550	825	1235	2060	3430
850	65	270	355	535	805	1340	2230
900	50	170	230	345	515	860	1430
950	35	105	140	205	310	515	860
1000	20	50	70	105	155	260	430

Additional system checks that may control design

- Nozzle load checks

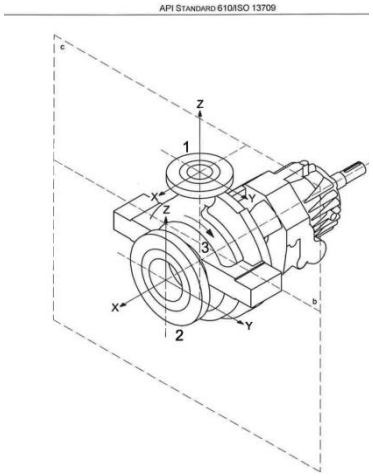


Table 5 (*continued*)

Location/orientation	Nozzle-loading force as a function of flange size — USC units lbf								
	Nominal size of flange (NPS)								
	≤ 2	3	4	6	8	10	12	14	16
Each top nozzle									
F_X	160	240	320	560	850	1 200	1 500	1 600	1 900
F_Y	130	200	260	460	700	1 000	1 200	1 300	1 500
F_Z	200	300	400	700	1 100	1 500	1 800	2 000	2 300
F_R	290	430	570	1 010	1 560	2 200	2 600	2 900	3 300
Moment ft·lbf									
Each nozzle									
M_X	340	700	980	1 700	2 600	3 700	4 500	4 700	5 400
M_Y	170	350	500	870	1 300	1 800	2 200	2 300	2 700
M_Z	260	530	740	1 300	1 900	2 800	3 400	3 500	4 000
M_R	460	950	1 330	2 310	3 500	5 000	6 100	6 300	7 200

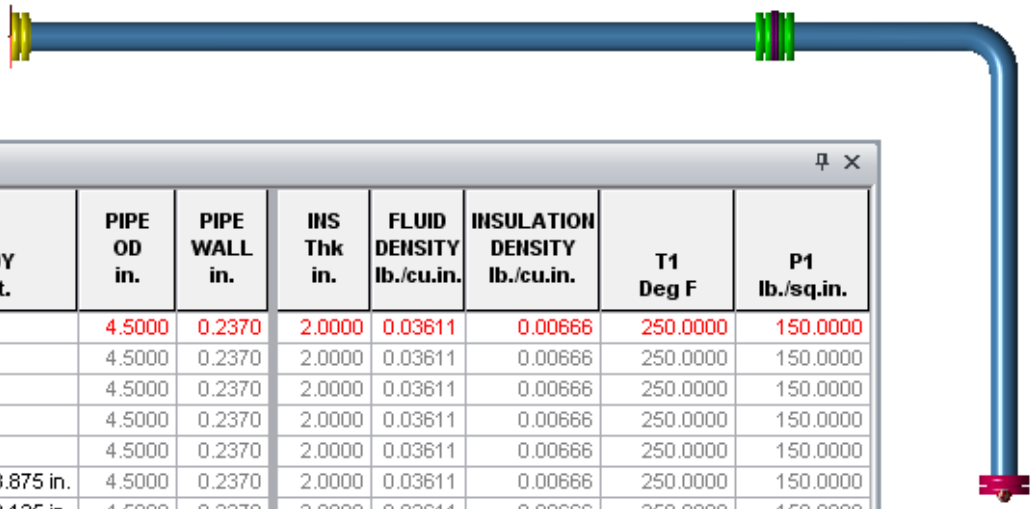
NOTE 1 See Figures 21 through 25 for orientation of nozzle loads (X, Y and Z).

NOTE 2 Each value shown above indicates range from minus that value to plus that value; for example 160 indicates a range from −160 to +160.

Check flange loads and (top discharge) nozzle loads

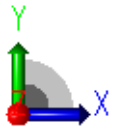
Legend

	Key	Node	DX1	DY1	DZ1	RX1	RY1	RZ1
1		40	0.0500	0.1000	0.0000	0.0000	0.0000	0.0000

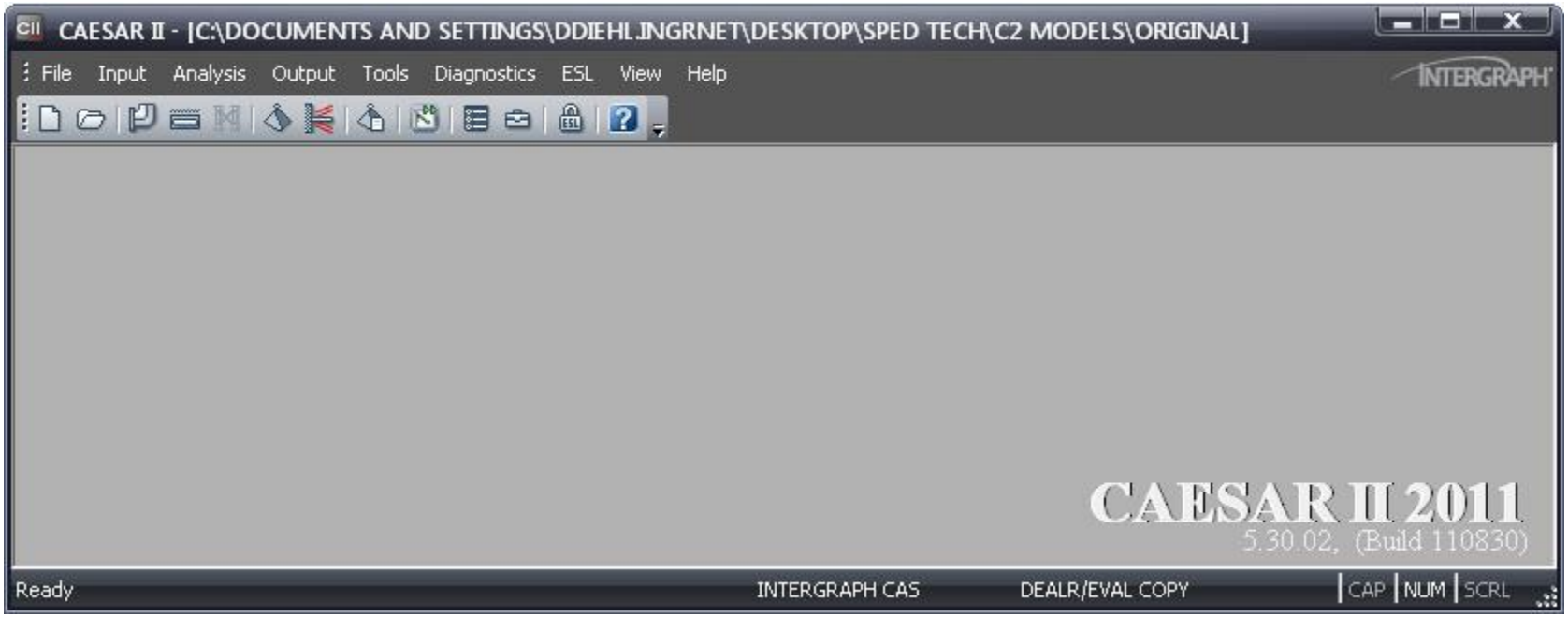


Elements

	FROM NODE	TO NODE	DX ft.	DY ft.	PIPE OD in.	PIPE WALL in.	INS Thk in.	FLUID DENSITY lb./cu.in.	INSULATION DENSITY lb./cu.in.	T1 Deg F	P1 lb./sq.in.
1	10	13	3.125 in.		4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000
2	13	16	9 ft. 5.750 in		4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000
3	16	20	3.125 in.		4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000
4	20	25	3.125 in.		4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000
5	25	30	2 ft. 8.875 in		4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000
6	30	35		-5 ft. 8.875 in.	4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000
7	35	40		-3.125 in.	4.5000	0.2370	2.0000	0.03611	0.00666	250.0000	150.0000

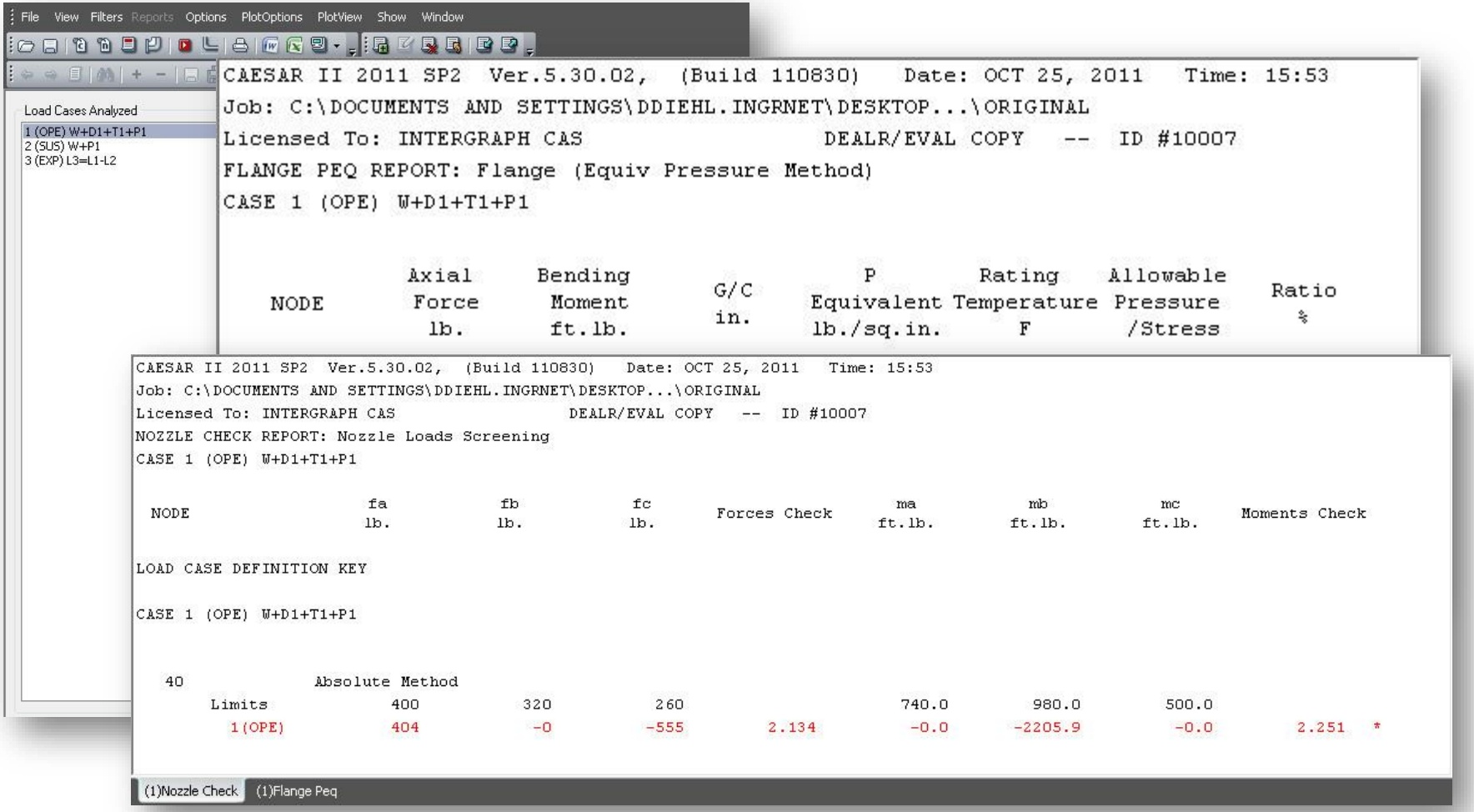


Return to CAESAR II

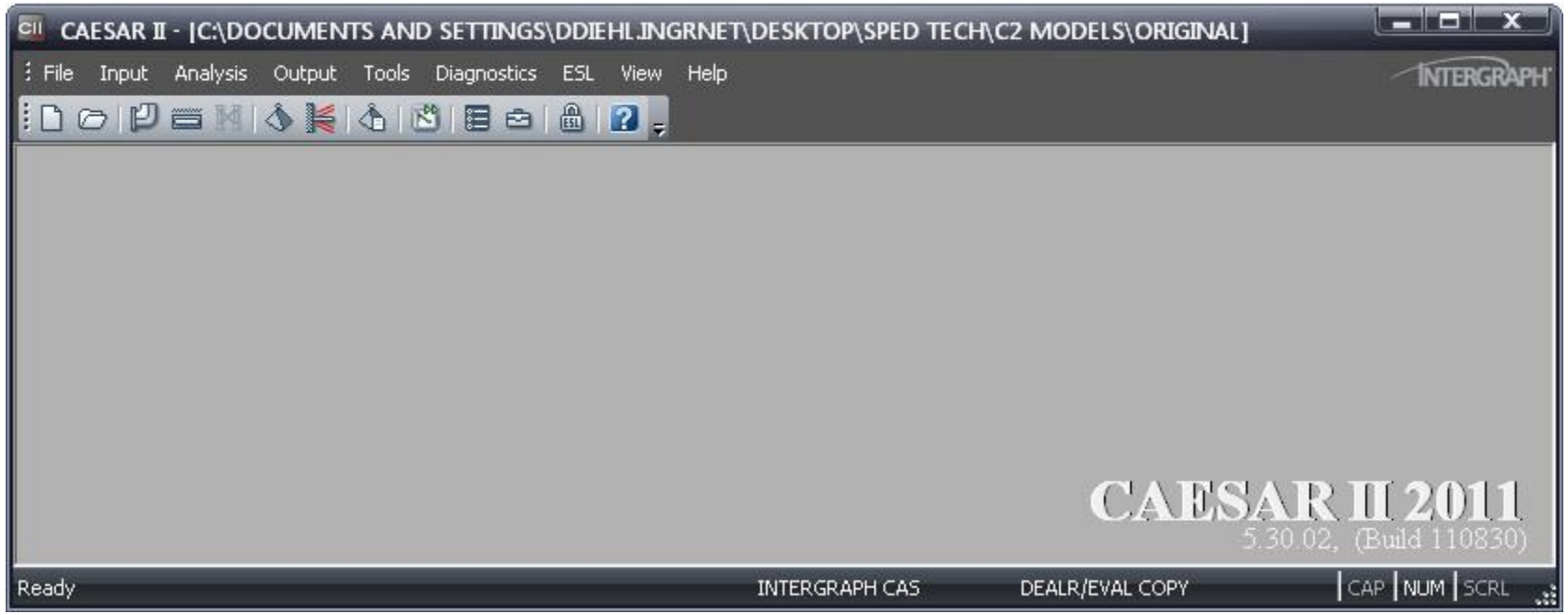


CAESAR II results review

- Flange equivalent pressure check
- API 610 nozzle check

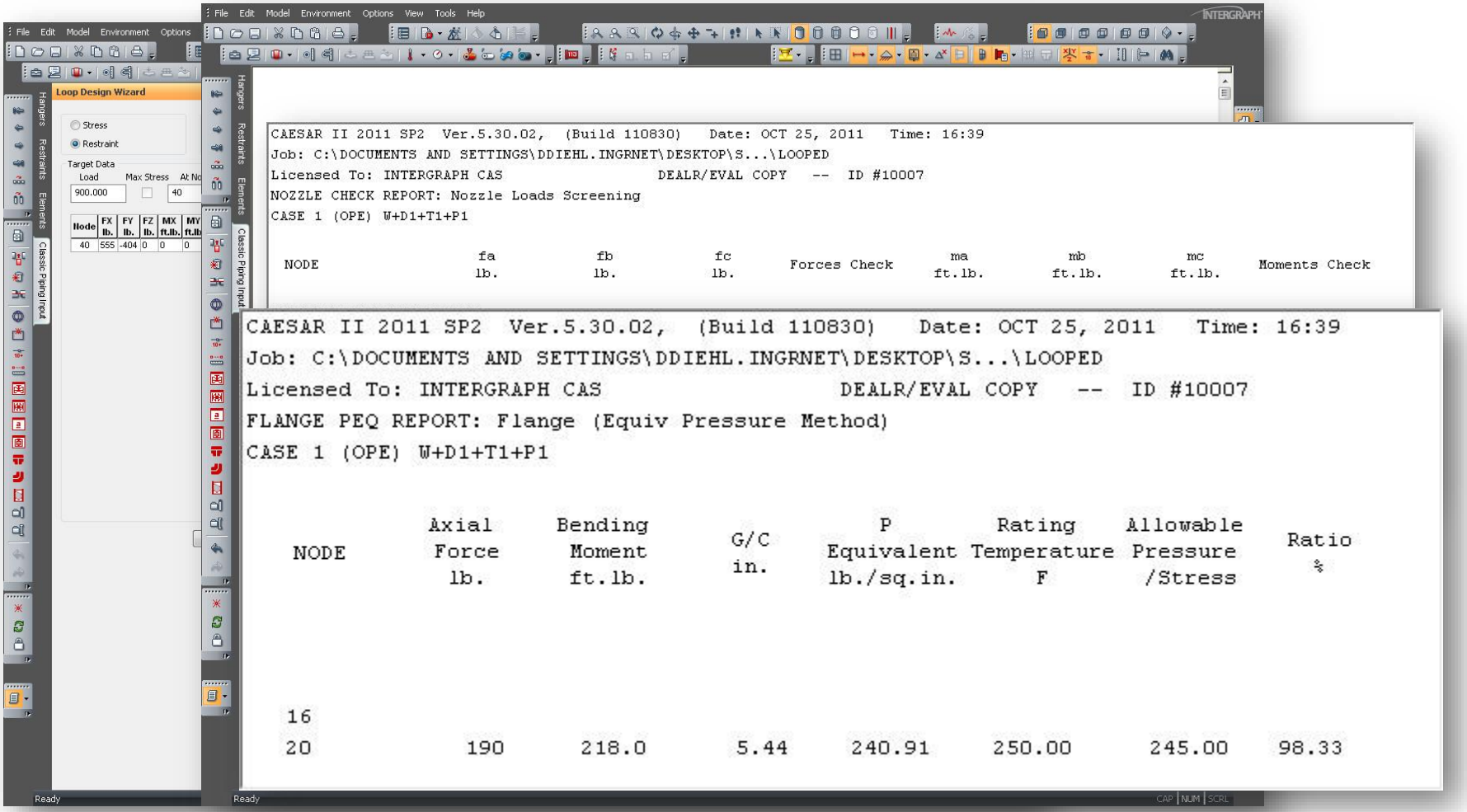


Return to CAESAR II – size the loop & select a hanger



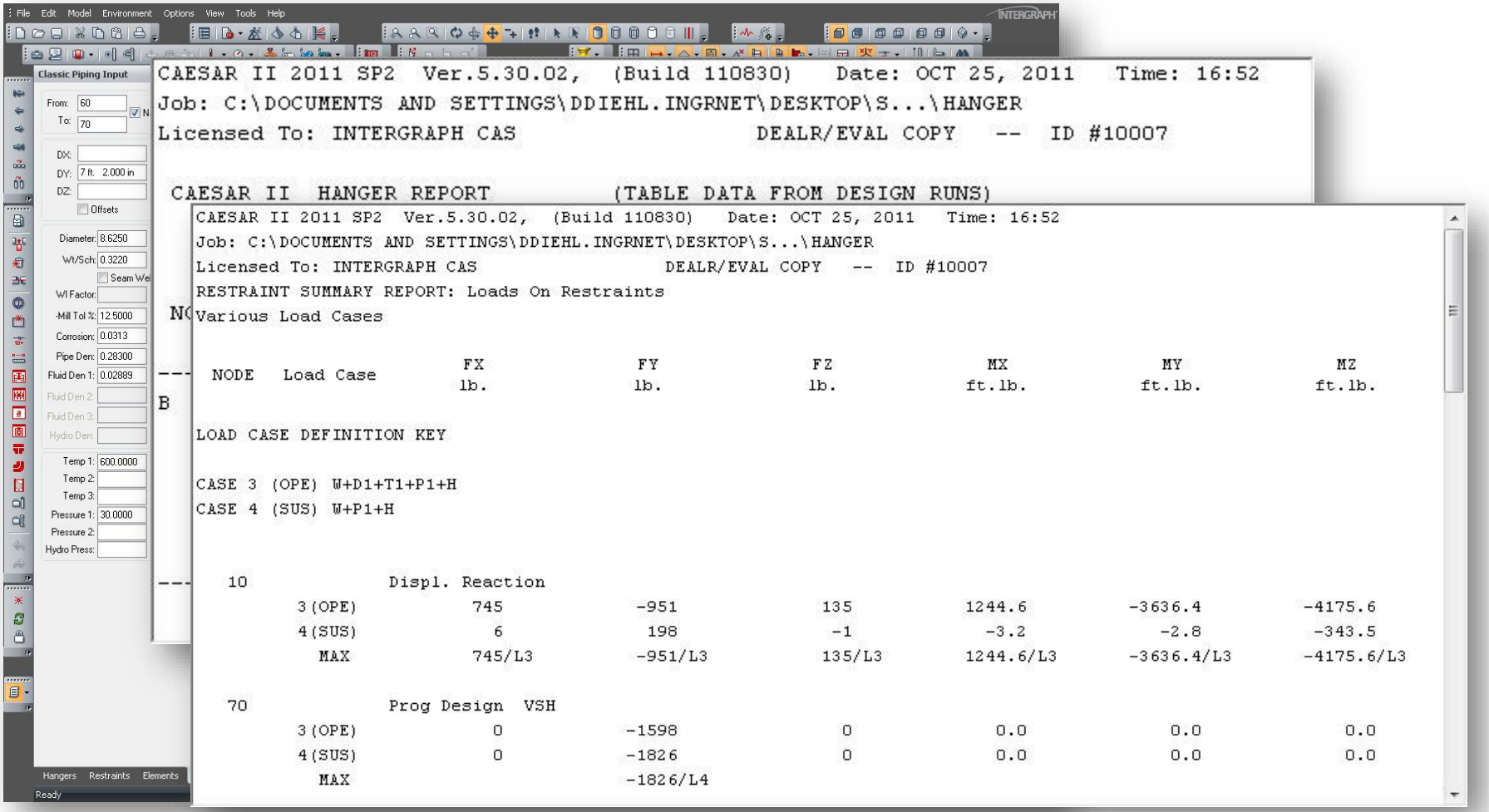
Design capabilities now found in pipe stress programs

- Loop optimizer



Design capabilities now found in pipe stress programs

- Hanger sizing



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Job: C:\DOCUMENTS AND SETTINGS\DDIEHL.INGRNET\DESKTOP\S...\HANGER
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CAESAR II  HANGER REPORT          (TABLE DATA FROM DESIGN RUNS)
```

```
CAESAR II 2011 SP2  Ver.5.30.02,  (Build 110830)   Date: OCT 25, 2011   Time: 16:52
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RESTRAINT SUMMARY REPORT: Loads On Restraints
Various Load Cases

  NODE   Load Case          FX          FY          FZ          MX          MY          MZ
                           lb.         lb.         lb.       ft.lb.      ft.lb.      ft.lb.

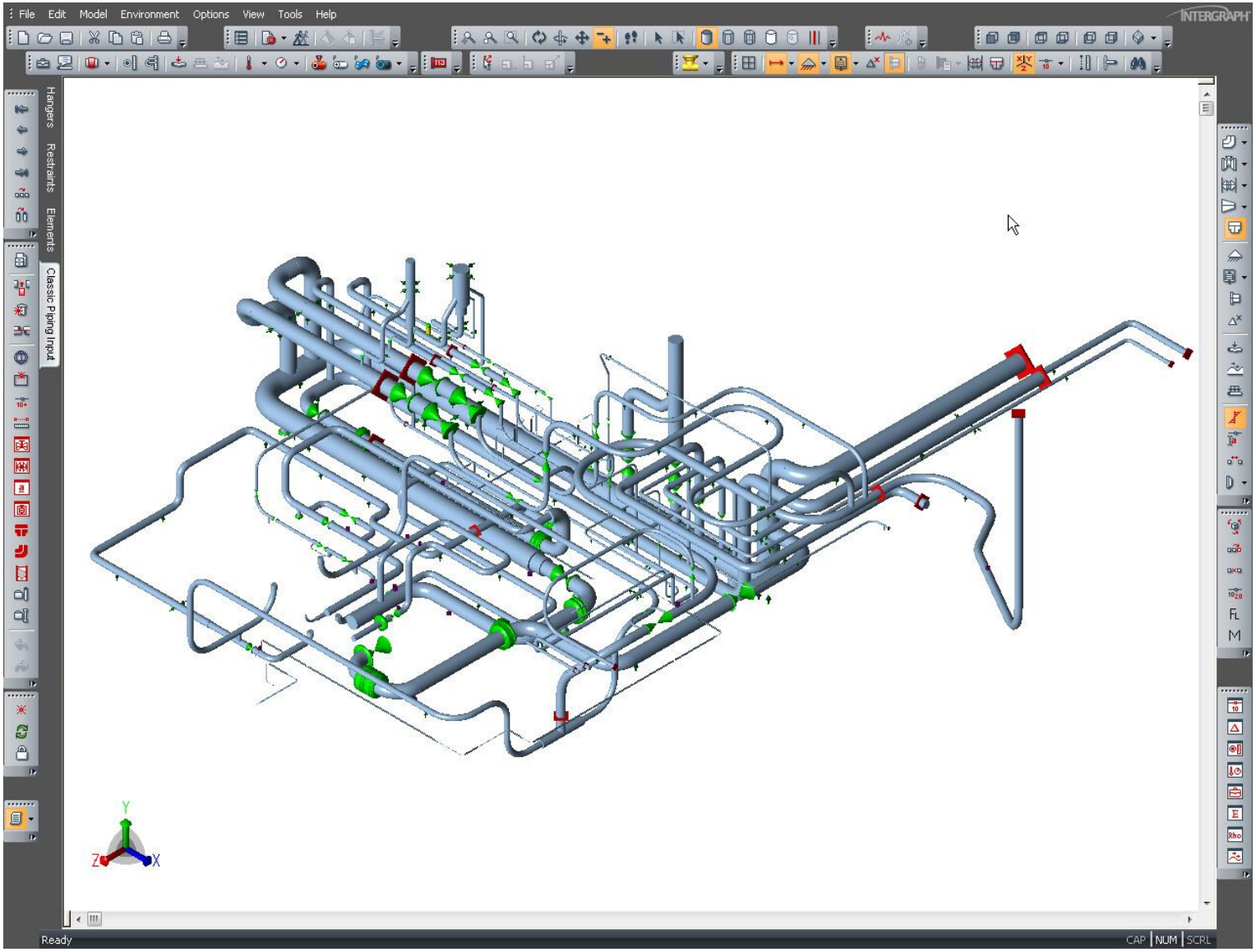
LOAD CASE DEFINITION KEY

CASE 3  (OPE)  W+D1+T1+P1+H
CASE 4  (SUS)  W+P1+H


   10              Displ. Reaction
         3(OPE)          745          -951          135        1244.6      -3636.4     -4175.6
         4(SUS)            6           198           -1          -3.2        -2.8        -343.5
           MAX          745/L3       -951/L3       135/L3     1244.6/L3   -3636.4/L3  -4175.6/L3

   70              Prog Design  VSH
         3(OPE)            0         -1598            0         0.0         0.0         0.0
         4(SUS)            0         -1826            0         0.0         0.0         0.0
           MAX                       -1826/L4
```

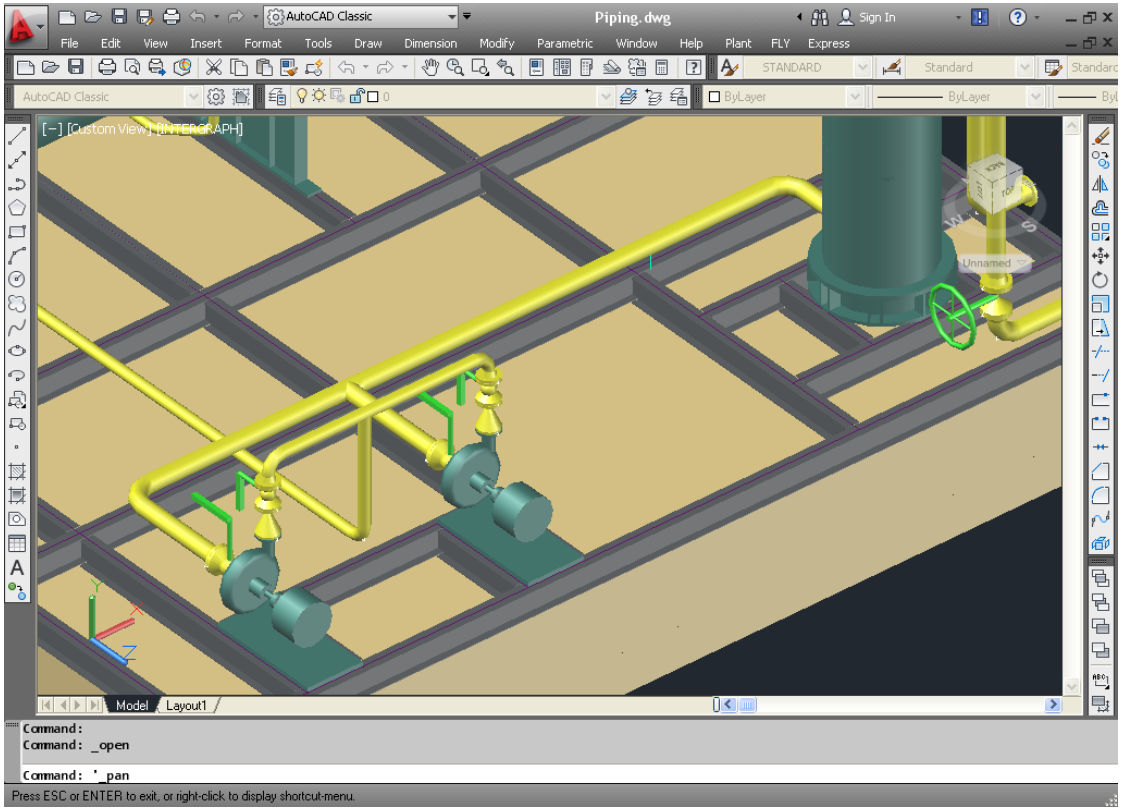
Here's a big job



... and some serious load cases

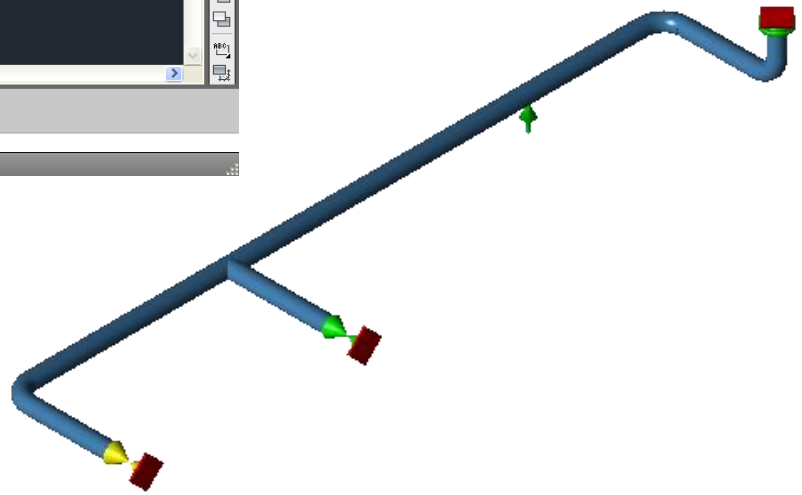
S/N	TYPE	COMBINATION	METHOD	DESCRIPTION
L1	HGR	W		Spring hanger selection - Dead weight calculation
L2	HGR	W + T1 + P1 + D7		Spring hanger selection - Operating displacement calculation
L3	HYD	WW + HP		Hydrotest Case
L4	SUS	W + P1 + H		Sustained Loading Case
L5	OPE	W + T1 + P1 + H + D7		Operating Case with Operating Temperature
L6	OPE	W + T2 + P1 + H + D8		Operating Case with Maximum Design Temperature
L7	OPE	W + T3 + P1 + H + D9		Operating Case with Minimum Design Temperature
L8	OPE	W + T1 + P1 + H + D7 + U1		Acceleration in actual boundary condition in N-S direction
L9	OPE	W + T1 + P1 + H + D7 + U2		Acceleration in actual boundary condition in E-W direction
L10	OPE	W + T1 + P1 + H + D7 + U3		Acceleration in actual boundary condition in Vertical direction
L11	OPE	W + T1 + P1 + H + D7 + WIN1		Extreme Wind in actual boundary condition in the most severe direction
L12	EXP	D1		Loading & Unloading - Hull Sagging Displacement
L13	EXP	D2		Loading & Unloading - Hull hogging Displacement
L14	EXP	D3		Wave - Hull Sagging Displacement
L15	EXP	D4		Wave - Hull Hogging Displacement
L16	EXP	D5		Wave - Module acceleration displacement added to hull sagging
L17	EXP	D6		Wave - Module acceleration displacement added to hull hogging
L18	SUS	L10 - L7	**Algebraic**	Acceleration only in N-S direction
L19	SUS	L11 - L7	**Algebraic**	Acceleration only in E-W direction
L20	SUS	L12 - L7	**Algebraic**	Acceleration only in Vertical direction
L21	SUS	L19 + L20 + L21	**Algebraic**	Combined Acceleration only #1
L22	SUS	L19 + L20 - L21	**Algebraic**	Combined Acceleration only #2
L23	SUS	L19 - L20 + L21	**Algebraic**	Combined Acceleration only #3
L24	SUS	L19 - L20 - L21	**Algebraic**	Combined Acceleration only #4
L25	SUS	-L19 + L20 + L21	**Algebraic**	Combined Acceleration only #5
L26	SUS	-L19 + L20 - L21	**Algebraic**	Combined Acceleration only #6
L27	SUS	-L19 - L20 + L21	**Algebraic**	Combined Acceleration only #7
L28	SUS	-L19 - L20 - L21	**Algebraic**	Combined Acceleration only #8
L29	SUS	L19 + L20 + L21	**SRSS**	Combined Acceleration for Occasional Stress Compliance
L30	EXP	2L30	**Algebraic**	Combined Acceleration Range for Extreme Fatigue Case
L31	OCC	L12 - L6	**Algebraic**	Extreme Wind only in the most severe direction
L32	EXP	L14-L13	**Algebraic**	Loading & Unloading - Displacement range
L33	EXP	L16 - L15 + L18 - L17	**Algebraic**	Wave - Displacement range
L34	OPE	L7 + L13 + L15 + L17 + L22	**Algebraic**	Design Case Equipment Load for Sagging #1
L35	OPE	L7 + L13 + L15 + L17 + L23	**Algebraic**	Design Case Equipment Load for Sagging #2
L36	OPE	L7 + L13 + L15 + L17 + L24	**Algebraic**	Design Case Equipment Load for Sagging #3
L37	OPE	L7 + L13 + L15 + L17 + L25	**Algebraic**	Design Case Equipment Load for Sagging #4
L38	OPE	L7 + L13 + L15 + L17 + L26	**Algebraic**	Design Case Equipment Load for Sagging #5
L39	OPE	L7 + L13 + L15 + L17 + L27	**Algebraic**	Design Case Equipment Load for Sagging #6
L40	OPE	L7 + L13 + L15 + L17 + L28	**Algebraic**	Design Case Equipment Load for Sagging #7
L41	OPE	L7 + L13 + L15 + L17 + L29	**Algebraic**	Design Case Equipment Load for Sagging #8
L42	OPE	L7 + L13 + L15 + L17 + L22	**Algebraic**	Design Case Equipment Load for Hogging #1
L43	OPE	L7 + L13 + L15 + L17 + L23	**Algebraic**	Design Case Equipment Load for Hogging #2
L44	OPE	L7 + L13 + L15 + L17 + L24	**Algebraic**	Design Case Equipment Load for Hogging #3
L45	OPE	L7 + L13 + L15 + L17 + L25	**Algebraic**	Design Case Equipment Load for Hogging #4
L46	OPE	L7 + L13 + L15 + L17 + L26	**Algebraic**	Design Case Equipment Load for Hogging #5
L47	OPE	L7 + L13 + L15 + L17 + L27	**Algebraic**	Design Case Equipment Load for Hogging #6
L48	OPE	L7 + L13 + L15 + L17 + L28	**Algebraic**	Design Case Equipment Load for Hogging #7
L49	OPE	L7 + L13 + L15 + L17 + L29	**Algebraic**	Design Case Equipment Load for Hogging #8
L50	OPE	L35 THROUGH L50	**Maximum**	Intermediate Load Case for L51
L51	OPE	L51 + L32	**Absolute**	Maximum Restraint Load
L52	SUS	L6	**Algebraic**	ASME B31.3 Code Compliance - Sustained Stress
L53	SUS	L6 + L30	**Absolute**	ASME B31.3 Code Compliance - Sustained Stress
L54	OCC	L6 + L30 + L32	**Absolute**	ASME B31.3 Code Compliance - Occasional Stress
L55	EXP	L9 - L6	**Algebraic**	ASME B31.3 Code Compliance - Expansion Stress Range
L56	EXP	L10 - L6	**Algebraic**	ASME B31.3 Code Compliance - Expansion Stress Range
L57	EXP	L7 - L8	**Algebraic**	ASME B31.3 Code Compliance - Expansion Stress Range
L58	FAT	L57+ L31 + L33 + L34	**Absolute**	Fatigue Stress Range from the Extreme condition
L59	FAT	L57 + kL31 + L33 + kL34	**Absolute**	Fatigue Stress Range from Loading / Unloading
L60	FAT	L31 + L34	**Absolute**	Fatigue Stress Range from On-site Wave

Working with the designer – bringing CADWorx layout to CAESAR II



CADWorx Model

Exported CAESAR II Model



Working with the designer – using the designer's data in S3D

- Creating PCFs for CAESAR II use
- Importing the PCF
- Importing S3D graphics into the CAESAR II environment

Next step?

- The designer initiates the analysis

Final Questions / General Discussion

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Thank you

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