Introductory Examples Manual for LS-DYNA[®] Users

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Introduction

This document presents some LS-DYNA examples providing a basic guide in different disciplines like:

- Structural static (stress analysis, buckling analysis and modal analysis)
- Structural dynamic (vibrations and impact)
- Thermal analysis (heat transfer via conduction, convection and radiation)

This guide is mainly addressed to first-time users. The input files are always present for each problem, using the KEYWORD input format. For sake of briefness, in most problems the full node and element definitions (also, some load segments) are omitted.

Several of the problems present a closed-form solution, while others (the majority) a reference solution obtained by using an arbitrary refined mesh (NAFEMS Benchmarks). In these cases, the obtained value vs. the reference solution value is reported. Most of the problems are implicit ones. Problem-specific keywords are listed under the title of each problem.

This guide refers to LS-DYNA v.971, but most of the problems also run on the 970, 960 and 950 versions. All problems have been tested using a double precision executable.

To report inaccuracies and/or comments, please contact support at the following email address: support@lstc.com

Benchmark References

Example 1. Skew Plate with Normal Pressure (thin shell mesh) *The Standard NAFEMS Benchmarks*, NAFEMS Report TNSB, Rev. 3, October, 1990, Test LE6.

Example 2. Skew Plate with Normal Pressure (thick shell mesh) *The Standard NAFEMS Benchmarks*, NAFEMS Report TNSB, Rev. 3, October, 1990, Test LE6.

Example 3. Elliptical Thick Plate under Normal Pressure (coarse mesh) Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test LE10.

Example 4. Elliptical Thick Plate under Normal Pressure (fine mesh) Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test LE10.

Example 5. Snap-Back under Displacement Control *NAFEMS Non-Linear Benchmarks*, NAFEMS Report NNB, Rev. 1, October, 1989, Test NL4.

Example 6. Straight Cantilever Beam with Axial End Point Load *NAFEMS Non-Linear Benchmarks*, NAFEMS Report NNB, Rev. 1, October, 1989, Test NL6.

Example 7. Lee's Frame Buckling Problem *NAFEMS Non-Linear Benchmarks*, NAFEMS Report NNB, Rev. 1, October, 1989, Test NL7.

Example 8. Pin-Ended Double Cross: In-Plane Vibration *The Standard NAFEMS Benchmarks*, NAFEMS Report TNSB, Rev. 3, October, 1990, Test FV2.

Example 9. Simply Supported Thin Annular Plate (coarse mesh) Abbassian, F., Dawswell, D.J., and Knowles, N.C., *NAFEMS Selected Benchmarks for Natural Frequency Analysis*, November, 1987, Test 14.

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Example 11. Transient Response to a Constant Force Biggs, J.M., *Introduction to Structural Dynamics*, McGraw-Hill Book Co., Inc., New York, New York, 1964, pg. 50, ex. E. Example 12. Simply Supported Square Plate: Out-of Plane Vibration (solid mesh) Abbassian, F., Dawswell, D.J., and Knowles, N.C., *NAFEMS Free Vibration Benchmarks*, October, 2001, Test FV52.

Example 13. Simply Supported Square Plate: Out-of-Plane Vibration (thick shell mesh) Abbassian, F., Dawswell, D.J., and Knowles, N.C., *NAFEMS Free Vibration Benchmarks*, October, 2001, Test FV52.

Example 14. Simply Supported Square Plate: Transient Forced Vibration (solid mesh) Maguire, J., Dawswell, D.J., and Gould, L.,*NAFEMS Selected Benchmarks for Forced Vibration*, February, 1989, Test 21T.

Example 15. Simply Supported Square Plate: Transient Forced Vibration (thick shell mesh)

Maguire, J., Dawswell, D.J., and Gould, L., *NAFEMS Selected Benchmarks for Forced Vibration*, February, 1989, Test 21T.

Example 16. Transient Response of a Cylindrical Disk Impacting a Deformable Surface Thomson, W.T., *Vibration Theory and Applications*, 2nd Printing, Prentice-Hall, Inc., Englewood Cliffs, New Jersey, 1965, pg. 110, ex. 4.6-1.

Example 17. Natural Frequency of a Linear Spring-Mass System Timoshenko, S.P., and Young, D.H., *Vibration Problems in Engineering*, 3rd Edition, D. Van Nostrand Co., Inc., New York, New York, 1955, pg.1.

Example 18. Natural Frequency of a Nonlinear Spring-Mass System Timoshenko, S.P., and Young, D.H., *Vibration Problems in Engineering*, 3rd Edition, D. Van Nostrand Co., Inc., New York, New York, 1955, pg. 141.

Example 19. Buckling of a Thin Walled Cylinder Under Compression Timoshenko, S.P., and Gere, J.M., *Theory of Elastic Stability*, McGraw-Hill Book Co., Inc., New York, New York, 1961, pg. 457.

Example 20. Membrane with a Hot Spot Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test T1.

Example 21. 1D Transient Heat Transfer with Radiation Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test T2.

Example 22. 1D Transient Heat Transfer in a Bar Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test T3. Example 23. 2D Heat Transfer with Convection Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test T4.

Example 24. 3D Thermal Load Davies, G.A.O., Fenner, R.T., and Lewis, R.W., *NAFEMS Background to Benchmarks*, June, 1992, Test LE11.

Example 25. Cooling of a Billet via Radiation Siegal, R., and Howell, J.R., *Thermal Radiation Heat Transfer*, 3rd Edition, Hemisphere Publishing Corporation, 1981, pg. 229, problem 21.

Example 26. Pipe Whip

Lerencz, R.M., *Element-by-Element Preconditioning Techniques for Large-Scale, Vectorized Finite Element Analysis in Nonlinear Solid and Structural Mechanics*, Ph.D. Thesis, Department of Mechanical Engineering, Stanford University, Palo Alto, California, March, 1989, pg. 142, pipe whip.

Example 27. Aluminum Bar Impacting a Rigid Wall

Lerencz, R.M., *Element-by-Element Preconditioning Techniques for Large-Scale, Vectorized Finite Element Analysis in Nonlinear Solid and Structural Mechanics*, Ph.D. Thesis, Department of Mechanical Engineering, Stanford University, Palo Alto, California, March, 1989, pg. 86, rod impact.

1. Skew Plate with Normal Pressure (thin shell mesh)

Keywords:

*CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLUTION

Description:

A skew plate of equal side lengths L and thickness t is subjected to a normal pressure P on the top face (Figure 1.1). The plate is meshed with thin shell elements with a 4 x 4 density. The plate is simply supported on four side faces, $U_z = 0$. Determine the maximum principal stress at plate center point E on the bottom surface.



Figure 1.1 – Sketch representing the structure.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Pressure	Linear	Linear	-	Implicit	1-Linear

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

 $L = 1.0 \ m, \ t = 0.01 \ m$

Material Data:

Mass Density	$\rho = 7.80 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.07 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Load:

Pressure $P = 7.0 \times 10^2 Pa$

Element Types:

Belytschko-Tsay shell (elform=2) S/R Hughes-Liu shell (elform=6) Fully integrated shell (elform=16)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA maximum principal stress at plate center Point E (Node 13) on bottom surface plus its Z-displacement, U_Z , are compared with *Standard NAFEMS Benchmarks*, Test LE6.

Reference Condition - Point E (Node 13)	Max Principal Stress (Pa)	$U_{z}(m)$
NAFEMS Benchmark Test LE6	0.802×10^{6}	-
Belytschko-Tsay shell (elform=2)	0.781×10^{6}	-1.616×10^{-5}
S/R Hughes-Liu shell (elform=6)	0.715×10^{6}	-1.507×10^{-5}
Fully integrated shell (elform=16)	0.696×10^{6}	-1.404×10^{-5}

These nodal displacement results were generated by *DATABASE_NODOUT keyword while the maximum principal stress results were generated by *DATABASE_ ELOUT.

LS-DYNA stress and strain output corresponds to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different post-processors).

Lobatto integration (intgrd=1 - *CONTROL_SHELL) was employed since it has an advantage in that the inner and outer integration points are on the shell surfaces. Gauss integration is the default through thickness integration rule (the default number of through thickness integration points is nip=2 - *SECTION_SHELL) in LS-DYNA, where 1-10 integration points may be specified, whereas, with Lobatto integration, 3-10 integration points may be specified (for 2 point integration, the Lobatto rule is very inaccurate).

For this coarse meshing, the one-point quadrature (low order) Belytschko-Tsay shell (elform=2) provides a good stress comparison (Figure 1.2).

The higher order, selectively reduced integration Hughes-Liu shell (elform=6) and the fully integrated Belytschko-Tsay shell (elform=16), which uses a 2x2 in-plane quadrature, provide comparatively stiffer results (Figure 1.3 and 1.4), probably due to the coarse meshing.



Figure 1.2 – Element formulation 2 (Belytschko-Tsay).



Figure 1.3 – Element formulation 6 (S/R Hughes-Liu).



Figure 1.4 – Element formulation 16 (fully integrated).

Input Deck:

*KE3 *TTT	WORD							
Skev	v Plate v	with Norma	l Pressure	(thin she	ll mesh)			
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\$#	dnorm 2	diverg 1	istif 1	nlprint 2				
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*001	ידים. כח. ער גרו	т.т. Т.т.	0.0	Ŧ	2			
\$#	wrpang	esort	irnxx	istupd	theory	bwc	miter	proj
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*DEE	INE CUR	VE						
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp	
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\$#		al		01				
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\$#	eid	pid	nl n	2 n3	n4	n5	n6 n	7 n8
	T	T	1 I	o 7	2			
	16	1	19 24	4 25	20			
*NOI	DE	-	17 2		20			
\$#	nid		x	v		Z	tc r	С
	1		0.0	0.0		0.0	3	
	25	1 86602	540 0	50000000		0 0	3	
*PAF	25 27	1.00002	510 0			0.0	5	
\$# t	itle							
mate	erial ty	pe # 1 (E	lastic)					
\$#	pid	secid	mid	eosid	hqid	qrav	adpopt	tmid
	- 1	1	1		-	-		
*SEC	CTION_SH	ELL						
\$#	secid	elform	shrf	nip	propt	qr/irid	icomp	setyp
	1	2	0.0	5	0	0.0		
\$	1	б	0.0	5	0	0.0		
\$	1	16	0.0	5	0	0.0		
44	t1	t2	t3	t4	nloc	marea		
γĦ		0 010000	0.010000	0.010000	0	0.0		
ې# 0.	010000	0.010000						
≎# 0. *MA1	.010000 [_ELASTI	C.010000						
₽# 0. *MA] \$#	010000 C_ELASTI mid	c ro	е	pr	da	db	not used	
># 0. *MAT \$#	010000 [_ELASTI mid 1	c ro 7800.0002	e .1000e+11	pr 0.300000	da 0.0	db 0.0	not used 0.0	
># 0. *MA7 \$# *LO2	010000 F_ELASTI mid 1 AD_SEGME	ro 7800.0002	e .1000e+11	pr 0.300000	da 0.0	db 0.0	not used 0.0	
° [#] 0. *MAJ \$# *LOZ \$#	O10000 C_ELASTI mid 1 AD_SEGME lcid	ro 7800.0002 NT 1 000000	e .1000e+11 at	pr 0.300000 nl	da 0.0 n2	db 0.0 n3 7	not used 0.0 n4	
₽# *MA] \$# *LO# \$#	O10000 C_ELASTI mid 1 AD_SEGME lcid 1	C ro 7800.0002 NT sf 1.000000	e .1000e+11 at 0.0	pr 0.300000 nl 1	da 0.0 n2 6 7	db 0.0 n3 7	not used 0.0 n4 2	
₽# 0. *MAJ \$# *LOZ \$#	O10000 F_ELASTIO mid 1 AD_SEGMEN 1 cid 1 1	C ro 7800.0002 NT sf 1.000000 1.000000	e .1000e+11 at 0.0 0.0	pr 0.300000 nl 1 2 3	da 0.0 n2 6 7 8	db 0.0 n3 7 8 9	not used 0.0 n4 2 3 4	
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1	1.000000	0.0	7	12	13	8
1	1.000000	0.0	8	13	14	9
1	1.000000	0.0	9	14	15	10
1	1.000000	0.0	11	16	17	12
1	1.000000	0.0	12	17	18	13
1	1.000000	0.0	13	18	19	14
1	1.000000	0.0	14	19	20	15
1	1.000000	0.0	16	21	22	17
1	1.000000	0.0	17	22	23	18
1	1.000000	0.0	18	23	24	19
1	1.000000	0.0	19	24	25	20

*END

Notes:

2. Skew Plate with Normal Pressure (thick shell mesh)

Keywords:

*CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLUTION

Description:

A skew plate of equal side lengths L and thickness t is subjected to a normal pressure P on the top face (Figure 2.1). The plate is meshed with thick shell elements with a 4 x 4 density. The plate is simply supported on four side faces of the bottom surface, $U_z = 0$. Determine the maximum principal stress at plate center point E on the bottom surface.



Figure 2.1 – Sketch representing the structure.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Pressure	Linear	Linear	-	Implicit	1-Linear

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

 $L = 1.0 \ m, \ t = 0.01 \ m$

Material Data:

Mass Density	$\rho = 7.80 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.07 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Load:

Pressure $P = 7.0 \times 10^2 Pa$

Element Types:

S/R 2x2 IPI thick shell (elform=2) Assumed strain 2x2 IPI thick shell (elform=3) Assumed strain RI thick shell (elform=5)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA maximum principal stress at plate center Point E (Node 113) on bottom surface plus its Z-displacement, U_z , are compared with *Standard NAFEMS Benchmarks*, Test LE6.

Reference Condition - Point E (Node113)	Max Principal Stress (Pa)	$U_{z}(m)$
NAFEMS Benchmark Test LE6	0.802×10^{6}	-
S/R 2x2 IPI thick shell (elform=2)	0.709×10^{6}	-1.496×10^{-5}
Assumed strain 2x2 IPI thick shell (elform=3)	0.021×10^{6} est.	-0.084×10^{-5}
Assumed strain RI thick shell (elform=5)	0.211×10^{6}	-0.849×10^{-5}

These nodal displacement results were generated by *DATABASE_NODOUT keyword while the maximum principal stress (nodal) results were generated by *DATABASE_ELOUT.

At least two elements through the thickness are usually recommended to capture bending response for assumed strain 2x2 IPI thick shell (elform=3) and assumed strain RI thick shell (elform=5) formulations.

LS-DYNA stress and strain output corresponds to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different post-processors).

Lobatto integration (intgrd=1 - *CONTROL_SHELL) was employed since it has an advantage in that the inner and outer integration points are on the shell surfaces. Gauss integration is the default through thickness integration rule (the default number of through thickness integration points is nip=2 - *SECTION_TSHELL) in LS-DYNA, where 1-10 integration points may be specified, whereas, with Lobatto integration, 3-10 integration points may be specified (for 2 point integration, the Lobatto rule is very inaccurate).

Only the higher order selectively reduced 2x2 IPI thick shell (elform=2) provides a reasonable stress comparison (Figure 2.2). As with other higher order options, this formulation provides a comparatively stiff result, again probably due to the coarse meshing.

The higher order assumed strain 2x2 IPI thick shell (elform=3) and assumed strain RI thick shell (elform=5) formulations do not provide acceptable solutions (Figure 2.3 and 2.4) since at least two elements through the thickness are usually recommended to capture bending response.



Figure 2.2 – Element formulation 2 (S/R 2x2 IPI).



Figure 2.3 – Element formulation 3 (assumed strain 2X2 IPI).



Figure 2.4 - Element formulation 5 (assumed strain RI).

Input Deck:

*KE	IYWORD							
*TI	TLE							
Ske	ew Plate y	with Normal	Pressure	(thick sh	ell)			
*CC	NTROL_IM	PLICIT_GENE	RAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.0	2	1	2			
*CC	NTROL_IM	PLICIT_SOLU	TION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	1	11	15	0.001000	0.010000	0.0	0.900000	1.000000
\$#	dnorm	diverg	istif	nlprint				
	2	1	1	2				
\$#	arcctl	arcdir	arclen	arcmth	arcdmp			
	0	1	0.0	1	2			
*CC	NTROL_SH	ELL						
\$#	wrpang	esort	irnxx	istupd	theory	bwc	miter	proj
2	20.00000	0	0	0	2	2	1	
\$#	rotascl	intgrd	lamsht	cstyp6	tshell	nfail1	nfail4	
	0.0	1						
*CC	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
1	.000000	0	0.0	0.0	0.0			
*DP	TABASE_E	LOUT						
\$#	dt/cycl							
	1.0E-01							
*DA	TABASE_B	INARY_D3PLC	T					
\$#	dt/cycl							
C	0.100000							
*DP	TABASE_H	ISTORY_TSHE	LL					
\$#	eid1	eid2	eid3	eid4	ei5	eid6	eid7	eid8
	б	7	10	11				
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\$#	lcid 1	sdi	r 0	sfa 0.0	sfo 0.0	offa 0.0	offo 0.0	dattyp	
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	1	.0000000	0	700	.0000000				
*EL	EMENT_TS	HELL							
\$#	eid	pid	n1	n2	n3	n4	n5	n6 n7	7 n8
	1	1	1	6	7	2	101	106 107	102
	16	1	19	24	25	20	119	124 125	5 120
*NO	DE								
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	125	1.866	02540	0.	50000000	(0.010		
*PA	RT								
\$#	title								
mat	erial ty	pe # 1	(Elast	ic)					
\$#	pid	seci	d	mid	eosid	hgid	grav	adpopt	tmid
	1		1	1					
*SE	CTION_TS	HELL							
\$#	secid	elfor	n	shrf	nip	propt	qr/irid	icomp	tshear
	1	:	2	0.0	5	0	0.0		
\$	1		3	0.0	5	0	0.0)	
\$	1		5	0.0	5	0	0.0)	
*MA'	T_ELASTI	C							
\$#	mid	r	С	е	pr	da	db	not used	
	1	7800.00	02.100	0e+11	0.300000	0.0	0.0	0.0	
*LO	AD_SEGME	NT							
\$#	lcid	S	£	at	nl	n2	n3	n4	
	1	1.00000	0	0.0	101	106	107	102	
	1	1.00000	0	0.0	102	107	108	103	
	1	1.00000	0	0.0	103	108	109	104	
	1	1.00000	0	0.0	104	109	110	105	
	1	1.00000	0	0.0	106	111	112	107	
	1	1.00000	0	0.0	107	112	113	108	
	1	1.00000	0	0.0	108	113	114	109	
	1	1.00000	0	0.0	109	114	115	110	
	1	1.00000	0	0.0	111	116	117	112	
	1	1.00000	0	0.0	112	117	118	113	
	1	1.00000	0	0.0	113	118	119	114	
	1	1.00000	0	0.0	114	119	120	115	
	1	1.00000	0	0.0	116	121	122	117	
	1	1.00000	0	0.0	117	122	123	118	
	1	1.00000	0	0.0	118	123	124	119	
	1	1.00000	0	0.0	119	124	125	120	

*END

Notes:

3. Elliptical Thick Plate under Normal Pressure (coarse mesh)

Keywords:

*CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLUTION *CONTROL_IMPLICIT_SOLVER

Description:

An elliptical thick plate with thickness t is subjected to a normal pressure P on its top surface (Figure 3.1). The plate is meshed with solid hexahedra element with a 4 x 6 x 4 density. Face CC'D'D has no Y-direction displacement, $U_y = 0$; face ABB'A' has no Xdirection displacement, $U_x = 0$; the X and Y displacements of face BCC'B' are fixed, $U_x = U_y = 0$; and the mid-plane (face BCC'B') has no X-, Y-, and Z-direction displacement, $U_x = U_y = U_z = 0$. Determine the direct stress along Y-direction at point D.



Figure 3.1 – Sketch representing the structure.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Pressure	Linear	Linear	-	Implicit	1-Linear

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

a = 1.0 m, b = 2.0 m, c = 1.75 m, d = 1.25 m, t = 0.60 m

Material Data:

Mass Density	$\rho = 7.80 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.07 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Load:

Pressure	$P = 1.0 \times 10^{6}$	Pa

Element Types:

Constant stress solid (elform=1) Fully integrated S/R solid (elform=2) Fully integrated S/R solid - for poor aspect ratio (eff) - (elform=-1) 8 point enhanced strain solid (elform=18)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA Y-direction stress at plate edge Point D (Node 29) on top surface plus its Z-displacement, U_z , are compared with *NAFEMS Background to Benchmarks*, Test LE10.

Reference Condition - Point D (Node 29)	Axial Stress $\sigma_{_{yy}}$ (Pa)	$U_{z}(m)$
NAFEMS Benchmark Test LE10	-5.38×10^{6}	-
Constant stress solid (elform=1)	-4.78×10^{6} est	-1.022×10^{-4}
Fully integrated S/R solid (elform=2)	-4.13×10^{6}	-0.802×10^{-4}
Fully integrated S/R solid (elform=-1)	-5.35×10^{6}	-1.005×10^{-4}
8 point enhanced strain solid (elform=18)	-6.40×10^{6}	-0.973×10^{-4}

Estimated/extrapolated result calculated from -3.67×10^6 Pa centroid value.

These nodal displacement results were generated by *DATABASE_NODOUT keyword while the axial stress (nodal) results were generated by *DATABASE_ELOUT (*elout* file) and *DATABASE_EXTENT_BINARY (*eloutdet* file provides detailed element output at integration points and connectivity nodes) keyword entries.

You can set intout=stress or intout= all (*DATABASE_EXTENT_BINARY) and have stresses output for all the integration points to a file called *eloutdet* (*DATABASE_ELOUT governs the output interval and *DATABASE_HISTORY_ SOLID governs which elements are output). Setting nodout=stress or nodout=all in *DATABASE_EXTENT_BINARY will write the extrapolated nodal stresses to *eloutdet*.

LS-DYNA stress and strain output corresponds to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different post-processors).

For this coarse mesh, the one-point quadrature (low order) constant stress solid (elform=1) element formulation (the LS-DYNA default) provides a fair stress comparison (Figure 3.2). Refinement of the mesh should provide a better comparison.

The higher order, fully integrated selectively reduced solid (elform=2), provides a comparatively stiff result (Figure 3.3), probably due to the coarse meshing.

The aspect ratio of the elements varies throughout the coarse meshing. An available option is the higher order, fully integrated S/R solid (the so-called efficient formulation

choice) intended to address poor aspect ratios (elform=-1). This formulation provides a good comparison for this coarse meshing (Figure 3.4).

The 8 point enhanced strain solid (elform=18), developed for linear statics only, over predicts the stress (Figure 3.5); no explanation is currently available.



Figure 3.2 – Element formulation 1 (constant stress).



Figure 3.3 – Element formulation 2 (fully integrated S/R).



Figure 3.4 – Element formulation -1 (fully integrated S/R).



Figure 3.5 – Element formulation 18 (8 point enhanced strain).

Input Deck:

*KI	EYWORD							
*T]	TLE							
Ell	liptical '	Thick Plate	e under No	rmal Press	ure (coars	e mesh)		
*CC	ONTROL_IM	PLICIT_GENE	RAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.100000	2	1	2			
*CC	ONTROL_IM	PLICIT_SOLU	JTION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	1	11	15	0.001000	0.010000	1.00e+10	0.900000	1.00e-10
*C(ONTROL_IM	PLICIT_SOLV	/ER					
\$#	lsolvr	lprint	negev	order	drcm	drcprm	autospc	autotol
	4	2	2	0	1	0.0	1	0.0
\$#	lcpack							
	2							
*C(ONTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
1	L.000000	0	0.0	0.0	0.0			
*DA	ATABASE_E	LOUT						
\$#	dt	binary	lcur	ioopt				
1.	.0000E-9	0	0					
*DA	ATABASE_B	INARY_D3PLC	T					
\$#	dt/cycl							
1	L.000000							
*DA	ATABASE_E	XTENT_BINAR	RΥ					
\$#	neiph	neips	maxint	strflg	sigflg	epsflg	rtflg	engflg
\$#	cmpflg	ieverp	beamip	dcomp	shge	stssz	n3thdt	ialemat
\$#	nintsld	pkp_sen	sclp	hydro	msscl	therm	intout	nodout
	8		1.0				stress	stress
*DA	ATABASE_H	ISTORY_SOLI	D					
\$#	id1	id2	id3	id4	id5	id6	id7	id8

	1	5	9	1	3				
*DEF	INE_CUR	VE							
\$#	lcid	sdir	sfa	sf	o off	a ofi	Eo	dattyp	
<u>A 11</u>	T	0	0.0	0.	0 0.	0 0	. 0		
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\$#	eid	pid	n1	n2 n	3 n4	n5	n6	n7	n8
4 11	1	1	1	10 1	3 4	2	11	14	5
	_	_	_			_			-
	96	1	168 1	72 17	4 170	169	173	175	171
*NOD	E								
\$#	nid		x		У	Z	tc	rc	
	1	2.00000	0000	0.	0	0.0	2		
	175		0.0	2.7500000	0 0.6	0000002	4		
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tilate ¢#	riai ty	pe # I (I	siastic) mid	eosi	d hai	d are	с <i>т</i>	dnont	tmid
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*SEC	TTON SO	т.тр	1						
\$#	secid	elform	aet						
4.0	1	1	1						
\$	1	2	2	1					
\$	1	-1	1	1					
\$	1	18	3	1					
*MAT	_ELASTI	С							
\$#	mid	ro	e	P	r d	a o	lb not	used	
	1	7800.0002	2.1000e+11	0.30000	0 0.	0 0	.0	0.0	
*LOA	D_SEGME	NT			-	•	2	4	
ŞĦ	ICIA 1	1 000000	at	n	L n		13	n4 21	
	1	1.000000	0.0	2	y 3 1 3	э. 7	20	 22	
	1	1 000000	0.0	3	1 3 5 4	, . 1 ,	12	33	
	1	1.000000	0.0	3	7 4	3 4	15	39	
	1	1.000000	0.0	3	3 3	9 (59	65	
	1	1.000000	0.0	6	5 6	9 '	71	67	
	1	1.000000	0.0	3	9 4	5 '	73	69	
	1	1.000000	0.0	6	9 7	3 '	75	71	
	1	1.000000	0.0	6	7 7	1 9	99	95	
	1	1.000000	0.0	9	5 9	9 10)1	97	
	1	1.000000	0.0	7	1 7	5 10)3	99	
	1	1.000000	0.0	9	9 10	3 10)5	101	
	1	1.000000	0.0	4	1 12	5 12	27	43	
	1	1.000000	0.0	4	3 12 F 12	7 12	29	45	
	1	1.000000	0.0	12	5 13 7 13	1 1. 3 1.	25	120	
	⊥ 1	1.000000	0.0	12	, 13 5 10	9 14	19	±∠∍ 7२	
	1	1.000000	0.0	7	3 14	9 1	51	75	
	1	1.000000	0.0	.12	9 13	5 1	53	149	
	1	1.000000	0.0	14	9 15	3 1	55	151	
	1	1.000000	0.0	7	5 15	1 10	59	103	
	1	1.000000	0.0	10	3 16	9 1'	71	105	
	1	1.000000	0.0	15	1 15	5 1'	73	169	
	1	1.000000	0.0	16	9 17	3 1	75	171	

*END

Notes:

1. One should remember that the constant stress solid (elform=1), the fully integrated S/R solid (elform=2), and the fully integrated S/R solid (the so-called efficient formulation choice) intended to address poor aspect ratios (elform=-1) were originally developed for performing highly nonlinear, dynamic deformation simulations.

4. Elliptic Thick Plate under Normal Pressure (fine mesh)

Keywords:

*CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLUTION *CONTROL_IMPLICIT_SOLVER

Description:

An elliptical thick plate with thickness t is subjected to a normal pressure P on its top surface (Figure 4.1). The plate is meshed with solid hexahedra element with an 8 x 12 x 4 density. Face CC'D'D has no Y-direction displacement, $U_y = 0$; face ABB'A' has no X-direction displacement, $U_x = 0$; the X and Y displacements of face BCC'B' are fixed, $U_x = U_y = 0$; and the mid-plane (face BCC'B') has no X-, Y-, and Z-direction displacement, $U_x = U_z = 0$. Determine the direct stress along Y-direction at point D.



Figure 4.1 – Sketch representing the structure.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Pressure	Linear	Linear	-	Implicit	1-Linear

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

a = 1.0 m, b = 2.0 m, c = 1.75 m, d = 1.25 m, t = 0.60 m

Material Data:

Mass Density	$\rho = 7.80 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.07 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Load:

Pressure	$P = 1.0 \times 10^{6}$	Pa

Element Types:

Constant stress solid (elform=1) Fully integrated S/R solid (elform=2) Fully integrated S/R solid (elform=-1) 8 point enhanced strain solid (elform=18)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA Y-direction stress at plate edge Point D (Node 77) on top surface plus its Z-displacement, U_z , are compared with *NAFEMS Background to Benchmarks*, Test LE10.

Reference Condition - Point D (Node 77)	Axial Stress $\sigma_{_{yy}}$ (Pa)	$U_{z}(m)$
NAFEMS Benchmark Test LE10	-5.38×10^{6}	-
Constant stress solid (elform=1)	-5.30×10 ⁶ est	-1.051×10^{-4}
Fully integrated S/R solid (elform=2)	-4.70×10^{6}	-0.947×10^{-4}
Fully integrated S/R solid (elform=-1)	-4.76×10^{6}	-0.991×10^{-4}
8 point enhanced strain solid (elform=18)	-6.28×10^{6}	-0.982×10^{-4}

Estimated/extrapolated result calculated from -4.07×10^6 Pa centroid value.

These nodal displacement results were generated by *DATABASE_NODOUT keyword while the axial stress (nodal) results were generated by *DATABASE_ELOUT (*elout* file) and *DATABASE_EXTENT_BINARY (*eloutdet* file provides detailed element output at integration points and connectivity nodes) keyword entries.

You can set intout=stress or intout=all (*DATABASE_EXTENT_BINARY) and have stresses output for all the integration points to a file called *eloutdet* (*DATABASE_ELOUT governs the output interval and *DATABASE_HISTORY_ SOLID governs which elements are output). Setting nodout=stress or nodout=all in *DATABASE_EXTENT_BINARY will write the extrapolated nodal stresses to *eloutdet*.

LS-DYNA stress and strain output corresponds to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different post-processors).

For this fine mesh, the one-point quadrature (low order) constant stress solid (elform=1) element formulation (the LS-DYNA default) provides a better stress comparison (Figure 4.2), when compared to the coarse mesh.

The higher order, fully integrated selectively reduced solid (elform=2) still provides a comparatively stiff result (Figure 4.3); however, much improved over the coarse mesh.

Doubling the elements in the x-y plane (mesh refinement) appears to have minimized the aspect ratio issue seen in the coarse mesh. The higher order, fully integrated S/R solid

(the so-called efficient formulation choice) intended to address poor aspect ratios (elform=-1) now provides a very similar result (Figure 4.4) to the fully integrated S/R solid (elform=2).

The 8 point enhanced strain solid (elform=18), developed for linear statics only, over predicts the stress result (Figure 4.5) by a fair amount (even with the change in mesh refinement); no explanation is presently available.



Figure 4.2 – Element formulation 1 (constant stress).



Figure 4.3 – Element formulation 2 (fully integrated S/R).



Figure 4.4 – Element formulation -1 (fully integrated S/R).



Figure 4.5 – Element formulation 18 (8 point enhanced strain).

Input Deck:

*KE	YWORD							
*TI	TLE							
Thi	.ck Ellip	tic Plate u	under Norm	al Pressur	e (fine me	sh)		
*CC	NTROL_IM	PLICIT_GEN	ERAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.100000	2	0	2			
*CC	NTROL_IM	PLICIT_SOL	JTION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	1	11	15	0.001000	0.010000	1.00e+10	0.900000	1.00e-10
*CC	NTROL_IM	PLICIT_SOL	/ER					
\$#	lsolvr	lprint	negev	order	drcm	drcprm	autospc	autotol
	4	2	2	0	0	0.0	0	0.0
\$#	lcpack 2							
*CC	NTROL TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
1	.000000	- 0	0.0	0.0	0.0			
*DA	TABASE_E	LOUT						
\$#	dt	binary	lcur	ioopt				
1.	0000E-9	0	0	_				
*DA	TABASE_B	INARY_D3PL	ЭТ					
\$#	dt/cycl							
1	.000000							
*DA	TABASE_E	XTENT_BINA	RY					
\$#	neiph	neips	maxint	strflg	sigflg	epsflg	rtflg	engflg
\$#	cmpflg	ieverp	beamip	dcomp	shge	stssz	n3thdt	ialemat
\$#	nintsld 8	pkp_sen	sclp 1.0	hydro	msscl	therm	intout stress	nodout stress
*DA	TABASE_H	ISTORY_SOL:	ID					
\$#	id1	id2	id3	id4	id5	id6	id7	id8
	1	17	33	49				

*DEE	FINE_CUR	VE								
\$#	lcid	sdi	r	sfa	sfo	offa	a of	fo	dattyp	
	1		C	0.0	0.0	0.	0 0	0.0		
\$#		a	1		01					
		0.	C		0.0					
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*ELE	EMENT_SO	LID								
\$#	eid	pid	nl	nž	2 n3	n4	n5	nб	n7	n8
	1	1	1	10	5 19	4	2	17	20	5
	384	1	574	582	2 584	576	575	583	585	577
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\$#	nid		х		У		Z	tc	rc	
	1	2.000	00000		0.0		0.0	2		
	585		0.0	2	.75000000	0.6	0000002	4		
*PAF	ΥT									
\$# t	tile									
mate	erial ty	pe # 1	(Elasti	Lc)						
\$#	pid	seci	f	mid	eosid	hgid	d gr	av	adpopt	tmid
	1		1	1						
*SEC	CTION_SO	LID								
\$#	secid	elfor	n	aet						
	1		1	1						
\$	1		2	1						
\$	1		-1	1						
\$	1		18	1						
*MAT	r_elasti	С								
\$#	mid	r	С	е	pr	da	a	db no	t used	
	1	7800.00	0 2.100)e+11	0.300000	0.	0 C	0.0	0.0	
*LOA	AD_SEGME	NT								
\$#	lcid	S	E	at	nl	n	2	n3	n4	
	1	1.00000	C	0.0	77	8	7	89	79	
	1	1.00000	C	0.0	79	8	9	91	81	
	1	1.00000	C	0.0	575	58	3 5	585	577	
	1	1.00000	C	0.0	575	58	3 5	585	577	
*ENI	0									

Notes:

1. One should remember that the constant stress solid (elform=1), the fully integrated S/R solid (elform=2), and the fully integrated S/R solid (the so-called efficient formulation choice) intended to address poor aspect ratios (elform=-1) were originally developed for performing highly nonlinear, dynamic deformation simulations.

5. Snap-Back under Displacement Control

Keywords:

```
*CONTROL_IMPLICIT_AUTO
*CONTROL_IMPLICIT_GENERAL
*CONTROL_IMPLICIT_SOLUTION
```

Description:

In this problem the implicit arc length method is used in order to solve the snap-back of the system. With traditional Newton-based methods it is not possible to fully solve this problem, due to the null tangent stiffness matrix at a certain point of the analysis.

Three DOF are present.

A sketch representing the structure is shown below (Figure 5.1) along with a finite element representation (Figure 5.2).



Figure 5.1 – Sketch representing the structure.



Figure 5.2 – The finite element representation of the problem. Four beams are used; the springs are modeled with discrete formulation, the truss is modeled with truss formulation. To avoid element inversion, the beams the springs are very long.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Force	Linear	Nonlinear	-	Implicit	6–Arc length w/BFGS

Units:

non-dimensional

Dimensional Data:

 $L = 2.50 \times 10^3$, $\alpha = 1.00 \times 10^{-2}$, $\alpha L = 2.50 \times 10^{10}$

Material Data:

$$AE = 5.0 \times 10^7$$
, $K_1 = 1.5$, $K_2 = AE / L\sqrt{(1 + \alpha^2)} = 1.9999 \times 10^3$, $K_3 = 0.25$, $K_4 = 1.0$

Load:

Axial Load P = 0.0 varied linearly to 4.0×10^3 (load values of 0.6499×10^3 , 1.300×10^3 , 1.949×10^3 , 2.599×10^3 , 3.243×10^3 , -1.099×10^3)

Element Types:

Truss (resultant) (elform=3) Discrete beam/cable (elform=6)

Material Models:

*MAT_001 or *MAT_ELASTIC

*MAT_074 or *MAT_ELASTIC_SPRING_DISCRETE_BEAM

Results Comparison:

LS-DYNA displacements U_A , U_B , V_C at locations A (Node 1), B (Node 2), and C (Node 3) are compared with *NAFEMS Non-Linear Benchmarks*, Test NL4 for each load value.

	NAFEMS NL4	LS-DYNA	NAFEMS NL4	LS-DYNA	NAFEMS NL4	LS-DYNA
P (load)	U _A (disp)	U _A (disp)	U _B (disp)	U _B (disp)	V _C (disp)	V _C (disp)
0.6499×10 ³	650.0	650.0	0.0904	0.0903	5.241	5.242
1.300×10 ³	1300.0	1300.0	0.2328	0.2329	13.260	13.266
1.949×10 ³	1950.0	1949.0	0.5149	0.5150	27.080	27.079
2.599×10 ³	2600.0	2600.0	1.3440	1.3338	56.500	56.500
3.243×10 ³	3250.0	3250.0	7.0890	7.1053	162.600	162.850
-1.099×10 ³	3900.0	2800.0	4999.0	3898.500	41.950	2047.200

Figure 5.3 shows the displaced geometry at selected load values.

These nodal displacement results (table above and Figures 5.4 and 5.5 below) were generated by *DATABASE_NODOUT keyword while the element stress results (Figures 5.6 and 5.7 below) were generated by *DATABASE_ELOUT.



Figure 5.3 – Displaced geometry at selected loads.



Figure 5.4 – X-displacement vs. applied load for Nodes 1 and 2.






Figure 5.6 – Axial force resultant vs. applied load for Elements 2 and 3.



Figure 5.7 – Axial force resultant vs. applied load for Elements 1 and 4.

Input deck:

*KE	YWORD							
*TI	TLE							
Sna	p-Back U	Inder Displa	acement Co	ntrol				
*CC	NTROL_IM	IPLICIT_AUT)		_			
\$#	iauto	iteopt	itewin	dtmin	dtmax			
	1	20	5	1.000e-09	0.00100			
*CC	NTROL_IM	IPLICIT_GENE	ERAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.001000	2	1	2	1		
*CC	NTROL_IM	IPLICIT_SOLU	JTION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	б	40	15	0.00100	0.01000	0.01000	0.900000	1.000000
\$#	dnorm	diverg	istif	nlprint				
	2	1	1	2				
\$#	arcctl	arcdir	arclen	arcmth	arcdmp			
	0	1	0.0	1	2			
*CC	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
1	.000000	0	0.0	0.0	0.0			
*DA	TABASE_E	LOUT						
\$#	dt	binary						
1.0	000e-04	1						
*DA	TABASE_G	LSTAT						
\$#	dt	binary						
1.0	000e-04	1						
*DA	TABASE_M	IATSUM						
\$#	dt	binary						
1.0	000e-04	1						
*DA	TABASE_N	IODFOR						
\$#	dt	binary						
1.0	000e-04	1						
*DA	TABASE_N	IODOUT						
\$#	dt	binary						

1.00	00e-04	1						
*DAT	ABASE_SI	PCFORC						
\$#	dt	binary						
1.00	00e-04	1						
*DAT	ABASE_BI	INARY_D3PL	OT	1				
şπ α	L/CYCL	lcat/nr	beam	npitc	psetia			
.0	ULUUUU		CROUP					
*DAT	ABASE_NO	JDAL_FORCE	_GROUP					
Ş#	nsid	Cld						
	1							
*DAT	ABASE_HI	ISTORY_BEA	M					
\$#	eid1	eid2	eid3	eid4	ei5	eid6	eid7	eid8
	1	2	3	4				
*DAT	ABASE_HI	ISTORY_NOD	E					
\$#	nidl	nid2	nid3	nid4	ni5	nid6	nid7	nid8
	1	2	3	4	5			
*DEF	INE_CUR	VЕ						
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp	
	1	0	0.0	0.0	0.0	0.0		
\$#		al		01				
		0.0		0.0				
	1.	.00000000	40	000.000000				
*ELE	MENT_BEA	AM						
\$#	eid	pid	n1 r	n2 n3	n4	n5	n6 n'	7 n8
	1	- 1	5	3 0	0	0	0	0 2
	2	2	3	2 0	0	0	0	0 2
	З	3	2	4 0	0	0	0	0 2
	4	4	2	1 0	0	0	0	0 2
*N∩D	- -	-	2	1 0	0	0	0	0 2
¢#	nid		v	V		7	to r	~
γπ	1 -1	1 0000000	+04			0 0		
	2	1.000000000	0 0	0.0		0.0		
	2	2500 000	0.0 25			0.0		
	3	2300.000	000 22	0.00000000		0.0		
	4	0000.000				0.0		
	5	2500.000	000 30	000.000000		0.0		
*BOU	NDARY_SI	PC_NODE	1 6	1.6	1 6	1 6	1 6	1 6
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		<u>^</u>			-			
	T	0	0	T	1	1	1	1
*BOU	I NDARY_SI	0 PC_NODE	U	1	1	1	1	1
*BOU \$#ni	I NDARY_SI d/nsid	0 PC_NODE cid	0 dofx	l dofy	1 dofz	1 dofrx	l dofry	l dofrz
*BOU \$#ni	I NDARY_SI d/nsid 2	0 PC_NODE cid 0	0 dofx 0	l dofy 1	1 dofz 1	1 dofrx 1	l dofry 1	l dofrz 1
*BOU \$#ni	I NDARY_SI d/nsid 2 3	0 PC_NODE cid 0 0	dofx 0 1	dofy 1 0	1 dofz 1 1	1 dofrx 1 1	l dofry 1 1	l dofrz 1 1
*BOU \$#ni	I NDARY_SI d/nsid 2 3 4	0 PC_NODE cid 0 0 0	0 dofx 0 1 1	dofy 1 0 1	1 dofz 1 1 1	1 dofrx 1 1 1	l dofry 1 1 1	l dofrz 1 1 1
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\$#	mid	ro	k	fO	d	cdf	tdf	
	4	1.000000	1.000000	0.0	0.0	0.0	0.0	
\$#	flcid	hlcid	c1	c2	dle	glcid		
	0	0	0.0	0.0	1.000000			
*LOA	D_NODE_	POINT						
\$#	node	dof	lcid	sf	cid	m1	m2	m3
	1	1	1	1.000000				
*SET	_NODE_L	IST						
\$#	sid	da1	da2	da3	da4	solver		
	1	0.0	0.0	0.0	0.0			
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	1	2		4	5			
*ENF	. –	2	5	-	5			

Notes:

- 1. Using the default values (i.e., BFGS without arc length) and an automatic time stepping control, it was possible to solve the problem only up to a certain load. At this point, (a) the BFGS solution method cannot go any further, due to the tangent stiffness matrix becoming close to null, resulting in a FATAL ERROR nonlinear solver failed to find equilibrium, or (b) the solution proceeded with an incorrect solution (no snap-back).
- 2. When the time step was allowed to increase up to 0.010, either by initial time step or dtmax (automatic time stepping control), a solution could be achieved, relatively quickly, but a somewhat noisy in the response.
- 3. Using the default tolerance (default=inactive) for the residual (force) norm appeared to result in non-convergence or inaccurate convergence (i.e. relative convergence was achieved, but the amount of out-of-balance forces became too large to guarantee the accuracy of the solution). The tolerance on force was therefore activated and set to (0.010). If this value is too large, convergence issues will result.
- 4. In addition to employing the BFGS solver with arc length (nsolvr=6), it was found necessary to employ the default arc length, that is, the generalized arc length method (arcctl=0), where the norm of the global displacement vector controls the solution; this includes all nodes. Attempts at employing the option whereby the arc length method was controlled based on the displacement of a single node were unsuccessful.

6. Straight Cantilever Beam with Axial End Point Load

Keywords:

*CONTROL_IMPLICIT_AUTO *CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLVER *CONTROL_IMPLICIT_SOLUTION

Description:

The analysis involves a cantilever beam loaded at one end with a quasi-axial load (axial component=100 normal component). The material is elastic. The X-displacement, the Y-displacement and the Z-rotation of the end point of the beam are determined.

A sketch representing the structure is shown below (Figure 6.1) along with the finite element model (Figure 6.2).



Figure 6.1 – Sketch representing the structure.

Straight Cantilever with Axial End Point Load



Figure 6.2 – Finite element model with applied loads and boundary conditions.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Force	Linear	Nonlinear	-	Implicit	2-Nonlinear w/BFGS

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

The beam has a constant square section (0.1 m x 0.1 m) and a total length of 3.2 m and is meshed with 32 beams of equal length.

Material Data:

Mass Density	$\rho = 7.85 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.10 \times 10^{11} Pa$
Poisson's Ratio	v = 0.0

Load:

Axial Load	$P = 3.844 \times 10^{6}$	Ν
Pressure	$Q = 3.844 \times 10^4$	N

Element Types:

Hughes-Liu beam with cross section integration (elform=1)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA displacements U_X , U_Y , R_Z at the end of the beam (Node 6) are compared with *NAFEMS Non-Linear Benchmarks*, Test NL6.

	$\boldsymbol{U}_{\boldsymbol{X}}(\boldsymbol{m})$	$U_{Y}(m)$	$R_{z}(rad)$
NAFEMS NL6	-5.0404	-1.3472	-3.0725
Node 6	-5.0629	-1.3607	-3.0646

These nodal displacement results were generated by *DATABASE_NODOUT keyword. The X-displacement (U_x) , Y-displacement (U_y) , and Z-rotation (R_z) histories for Node 6 are given in Figure 6.3. Figures 6.4 and 6.5 provide the contour plot of the bending moment and the axial force, respectively, at the end of the step.



Figure 6.3 – X-displacement, Y-displacement, and Z-rotation for Node 6.



Figure 6.4 – Contour plot of the bending moment at the end of the step.



Figure 6.5 – Contour plot of the axial force at the end of the step.

Input deck:

*KE	YWORD							
*TI	TLE							
Str	aight Ca	ntilever w	ith Axial	End Point	Load			
*CO	NTROL_IM	PLICIT_AUT	0					
\$#	iauto	iteopt	itewin	dtmin	dtmax			
	1	11	5		0.010000			
*C0	NTROL_IM	PLICIT_GEN	ERAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.010000	2	0	2			
*C0	NTROL_IM	PLICIT_SOL	VER					
\$#	lsolvr	lprint	negev	order	drcm	drcprm	autospc	autotol
	5	2	2	0	1	0.0	1	0.0
\$#	lcpack							
	2							
*CO	NTROL_IM	PLICIT_SOL	UTION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	2	11	15	0.0010	0.0100	1.00e+10	0.900000	1.00e-10
\$#	dnorm	diverg	istif	nlprint	nlnorm			
	2	1	1	2	1			
\$#	arcctl	arcdir	arclen	arcmth	arcdmp			
	б	1	0.0	1	2			
*C0	NTROL_OU	TPUT						
\$#	npopt	neecho	nrefup	iaccop	opifs	ipnint	ikedit	iflush
	1	3	1	0	0.0	0	1000	5000
\$#	iprtf							
	3							
*CO	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
1	.000000	0	0.0	0.0	0.0			
*DA	TABASE_N	CFORC						
\$#	dt	binary						
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э н 0.	001000	binary 1						
*DAT	ABASE_NO	DOUT						
\$# 0	001000	binary 1						
*DAT	ABASE_N	DAL_FORCE	_GROUP					
\$#	nsid	cid						
*דעם	2 יאפאפד עי	מטע עפטייסי	r					
\$#	nid1	nid2	nid3	nid4	ni5	nid6	nid7	nid8
	6							
*DAT	ABASE_H	ISTORY_BEA	M	4			- 1 17	10
ŞĦ	3	eidz	eras	eid4	erp	erae	eid/	eids
*DEF	INE_CURV	/E						
\$#	lcid	sdir	sfa	sfo	offa	offo	datty	ō
¢#	1	0	1.000000	1.000000	0.0	0.0		
φ #		0.0		0.0				
	1	.00000000	3.84	40088e+06				
*DEF	INE_CURV	/E	c	c				
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\$#	2	al	1.000000	o1	0.0	0.0		
		0.0		0.0				
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"ЕЦЕ \$#	eid	pid	nl r	n2 n3	n4	n5	n6 i	n7 n8
	1	1	1	2 3	0	0	0	0 2
	22	1	2	07	0	0	0	0 0
*NOD	33 E	T	2 9	92 97	0	0	0	0 2
\$#	nid		x	У		Z	tc :	rc
	1		0.0	0.0		0.0		
	1 97	0.15000	0.0	0.0	0.010	0.0		
*BOU	1 97 NDARY_SI	0.15000 PC_NODE	0.0 001	0.0	0.010	0.0		
*BOU \$#ni	1 97 NDARY_SI d/nsid	0.15000 C_NODE cid	0.0 001 dofx	0.0 0.0 dofy	0.010 dofz	0.0 00000 dofrx	dofry	dofrz
*BOU \$#ni	1 97 NDARY_SI d/nsid 1	0.15000 PC_NODE cid 0	0.0 001 dofx 1	0.0 0.0 dofy 1	0.010 dofz 1	0.0 00000 dofrx 1	dofry 1	dofrz 1
*BOU \$#ni *PAR \$# t	1 97 NDARY_SI d/nsid 1 T itle	0.15000 PC_NODE cid 0	0.0 001 dofx 1	0.0 0.0 dofy 1	0.010 dofz 1	0.0 00000 dofrx 1	dofry 1	dofrz 1
*BOU \$#ni *PAR \$# t	1 97 NDARY_SH d/nsid 1 T itle	0.15000 PC_NODE cid 0	0.0 001 dofx 1	0.0 0.0 dofy 1	0.0100 dofz 1	0.0 00000 dofrx 1	dofry 1	dofrz 1
*BOU \$#ni *PAR \$# t \$#	1 97 NDARY_SH d/nsid 1 T itle pid	0.15000 PC_NODE cid 0 secid	0.0 001 dofx 1 mid	0.0 0.0 dofy 1 eosid	0.0100 dofz 1 hgid	0.0 00000 dofrx 1 grav	dofry 1 adpopt	dofrz 1 tmid
*BOU \$#ni *PAR \$# t \$# *SEC	1 97 NDARY_SI d/nsid 1 T itle pid 1 TTION BE2	0.15000 PC_NODE cid 0 secid 1	0.0 001 dofx 1 mid 1	0.0 0.0 dofy 1 eosid	0.010 dofz 1 hgid	0.0 00000 dofrx 1 grav	dofry 1 adpopt	dofrz 1 tmid
*BOU \$#ni *PAR \$# t \$# *SEC \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid	0.15000 PC_NODE cid 0 secid 1 AM elform	0.0 001 dofx 1 mid 1 shrf	0.0 0.0 dofy 1 eosid qr/irid	0.010 dofz 1 hgid cst	0.0 00000 dofrx 1 grav scoor	dofry 1 adpopt	dofrz 1 tmid
*BOU \$#ni *PAR \$# t \$# \$# \$EC \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1	0.15000 PC_NODE cid 0 secid 1 AM elform	0.0 001 dofx 1 mid 1 shrf 0.830000	0.0 0.0 dofy 1 eosid qr/irid 2	0.010 dofz 1 hgid cst 0	0.0 00000 dofrx 1 grav scoor 0.0	dofry 1 adpopt	dofrz 1 tmid
*BOU \$#ni *PAR \$# t \$# *SEC \$# \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1 tsl	0.15000 PC_NODE cid 0 secid 1 AM elform 1 ts2 0.10000	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000	0.010 dofz 1 hgid cst 0 nsloc	0.0 00000 dofrx 1 grav scoor 0.0 ntloc	dofry 1 adpopt	dofrz 1 tmid
*BOU \$#ni *PAR \$# t \$# *SEC \$# \$# 0. *MAT	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEZ secid 1 ts1 100000 ELASTIC	0.15000 PC_NODE cid 0 secid 1 M elform ts2 0.100000	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000	0.010 dofz 1 hgid cst 0 nsloc	0.0 00000 dofrx 1 grav scoor 0.0 ntloc	dofry 1 adpopt	dofrz 1 tmid
*BOU \$#ni *PAR \$# t \$# *SEC \$# \$# 0. *MAT \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TTION_BE2 secid 1 ts1 100000 C_ELASTIC mid	0.15000 PC_NODE cid 0 secid 1 elform 1 ts2 0.100000 C ro	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr	0.010 dofz 1 hgid cst 0 nsloc da	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db	dofry 1 adpopt not used	dofrz 1 tmid
*BOU \$#ni \$# t \$# \$# \$# 0. \$# \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1 ts1 100000 '_ELASTIC mid 1	0.15000 PC_NODE cid 0 secid 1 AM elform 1 ts2 0.100000 C 7850.0002	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e .1000e+11	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0	0.010 dofz 1 hgid cst 0 nsloc da 0.0	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0	dofry 1 adpopt not used 0.0	dofrz 1 tmid
*BOU \$#ni \$# t \$# *SEC \$# 0. \$# \$# *MAT \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEZ secid 1 ts1 100000 '_ELASTIC mid 1 D_NODE_H node	0.15000 PC_NODE cid 0 secid 1 M elform 1 ts2 0.100000 C ro 7850.0002 POINT dof	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e .1000e+11 lcid	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid	0.0 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml	dofry 1 adpopt not used 0.0 m2	dofrz 1 tmid
*BOU \$#ni \$# t \$# *SEC \$# 0. *MAT \$# *LOA \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1 tsl 100000 C_ELASTIC mid 1 D_NODE_H node 6	0.15000 PC_NODE cid 0 secid 1 AM elform 1 ts2 0.100000 C ro 7850.0002 POINT dof 1	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e .1000e+11 lcid 1	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf -1.000000	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml	dofry 1 adpopt not used 0.0 m2	dofrz 1 tmid
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*BOU \$#ni \$# t \$# *SEC \$# 0. \$# *MAT \$# *LOA \$# *LOA \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEZ secid 1 ts1 100000 C_ELASTIC mid 1 D_NODE_I node 6 D_NODE_I 6	0.15000 PC_NODE cid 0 secid 1 M elform 1 ts2 0.100000 c ro 7850.0002 POINT dof 2 20INT dof 2	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 .1000e+11 lcid 1 lcid 2	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf -1.000000	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid cid	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml ml	dofry 1 adpopt not used 0.0 m2 m2	dofrz 1 tmid m3 m3
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*BOU \$#ni \$# t \$# *SEC \$# \$# 0. \$# \$# \$# \$# \$# \$# \$# \$# \$# \$ \$# \$	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1 tsl 100000 C_ELASTIC mid D_NODE_I node 6 D_NODE_L: sid 1 node 1 n n n n n n n n n n n n n	0.15000 PC_NODE cid 0 secid 1 M elform 1 ts2 0.100000 C ro 7850.0002 POINT dof 2 IST_GENERA dal 0.0 0.0	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e.1000e+11 lcid 1 lcid 2 TE da2 0.0 mid 2	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf -1.000000 sf -1.000000 da3 0.0 prd4	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid cid da4 0.0 pid5	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml ml solver	dofry 1 adpopt not used 0.0 m2 m2	dofrz 1 tmid m3 m3
*BOU \$#ni \$# t \$# *SEC \$# *O. \$# *LOA \$# *SET \$# \$# \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1 ts1 100000 '_ELASTIC mid 1 D_NODE_I node 6 D_NODE_L sid 1 nid1 2	0.15000 PC_NODE cid 0 secid 1 M elform 1 ts2 0.100000 C ro 7850.0002 POINT dof 2 IST_GENERA dal 0.0 nid2 92	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e .1000e+11 lcid 1 lcid 2 TE da2 0.0 nid3	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf -1.000000 -1.000000 da3 0.0 nid4	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid cid da4 0.0 nid5	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml ml solver nid6	dofry 1 adpopt not used 0.0 m2 m2 m2 n1d7	dofrz 1 tmid m3 m3 nid8
*BOU \$#ni \$# t \$# *SEC \$# *MAT \$# *LOA \$# *LOA \$# *SET \$# *SET \$#	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEA secid 1 ts1 100000 C_ELASTIC mid D_NODE_I node 6 D_NODE_L i sid 1 nid1 2 _NODE_L:	0.15000 PC_NODE cid 0 secid 1 M elform 1 ts2 0.100000 C ro 7850.0002 POINT dof 1 POINT dof 2 ST_GENERA dal 0.0 nid2 92 IST	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 e .1000e+11 lcid 1 lcid 2 TE da2 0.0 nid3	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf -1.000000 sf -1.000000 da3 0.0 nid4	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid cid da4 0.0 nid5	0.0 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml ml solver nid6	dofry 1 adpopt not used 0.0 m2 m2 m2 n1d7	dofrz 1 tmid m3 m3 nid8
*BOU \$#ni \$# t \$# \$SEC \$# 0.1 \$# *LOA \$# *LOA \$# *LOA \$# \$ \$# \$\$ \$# \$\$	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEZ secid 1 ts1 100000 '_ELASTIC mid 1 D_NODE_I node 6 NODE_LI sid 2 NODE_LI	0.15000 PC_NODE cid 0 secid 1 M elform 1 ts2 0.100000 ro 7850.0002 POINT dof 2 IST_GENERA dal 0.0 nid2 92 IST dal	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 .1000e+11 lcid 1 lcid 2 TE da2 0.0 nid3	0.0 0.0 dofy 1 eosid qr/irid 2 0.100000 pr 0.0 sf -1.000000 da3 0.0 nid4 da3	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid cid da4 0.0 nid5 da4	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml ml solver nid6 solver	dofry 1 adpopt not used 0.0 m2 m2 m2 n1d7	dofrz 1 tmid m3 m3 nid8
*BOU \$#ni \$# t \$# *SEC \$# 0.1 \$# *LOA \$# *LOA \$# *LOA \$# *SET \$# \$ \$# \$ \$# \$	1 97 NDARY_SI d/nsid 1 T itle pid 1 TION_BEZ secid 1 tsl 100000 '_ELASTIC mid 1 D_NODE_I node 6 NODE_LI: sid 2 nid1	0.15000 PC_NODE cid 0 secid 1 M elform 1 0.100000 ro 7850.0002 POINT dof 2 IST_GENERA dal 0.0 nid2 92 IST dal 0.0 nid2	0.0 001 dofx 1 mid 1 shrf 0.830000 tt1 0.100000 .1000e+11 lcid 1 lcid 2 TE da2 0.0 nid3 da2 0.0 nid3	0.0 0.0 dofy 1 eosid qr/irid 2 tt2 0.100000 pr 0.0 sf -1.000000 da3 0.0 nid4 da3 0.0 nid4	0.010 dofz 1 hgid cst 0 nsloc da 0.0 cid cid da4 0.0 nid5 da4 0.0 nid5	0.0 00000 dofrx 1 grav scoor 0.0 ntloc db 0.0 ml ml ml solver nid6 solver nid6	dofry 1 adpopt not used 0.0 m2 m2 m2 nid7	dofrz 1 tmid m3 m3 nid8

*END

Notes:

- 1. Using the default values, with an initial time step dt0=0.010, the problem stops at the 12th iteration due to an energy increase. The *CONTROL_IMPLICIT_GENERAL, *CONTROL_IMPLICIT_SOLUTION, and *CONTROL_IMPLICIT_SOLVER, with no automatic time stepping (*CONTROL_IMPLICIT_AUTO) are considered to be the default keywords.
- 2. Allowing more iterations (*CONTROL_IMPLICIT_SOLUTION) will not help to solve the problem.
- 3. To resolve the energy increase and termination stated above, include the automatic time stepping (*CONTROL_IMPLICIT_AUTO) entry, in particular the specification of dtmax. The following situations occur when using different values of dtmax:

dtmax =blank (10*dt0) or 0.100 (these are actually the same); the current step size will increase right off, eventually two energy increases will occur, where time steps are then decreased, with the simulation then continuing until termination is reached. This takes the least iterations with ASCII result plots somewhat noisy.

dtmax =0.010 (the initial time step); solves very nicely with no energy increases, takes about 50 percent more iterations than dtmax=0.100, with smoother ASCII result plots.

dtmax =0.001 yielded the same results as dtmax =0.010. dt0 appeared to still be considered in the time step options.

4. It is also possible to achieve a successful solution specifying an initial time step of dt0=0.001 and a similar value for the maximum allowable time step (dtmax=0.001 in the *CONTROL_IMPLICIT_AUTO keyword). Using these parameters will increase the number of iterations significantly.

7. Lee's Frame Buckling Problem

Keywords:

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*CONTROL_IMPLICIT_AUTO
*CONTROL_IMPLICIT_GENERAL
*CONTROL_IMPLICIT_SOLUTION
```

Description:

The problem involves a framed structure deforming under the action of a load applied on one node. The frame is pinned to the ground at two nodes and the load is applied on Node 56, as shown in Figure 7-1. The finite element model is shown in Figure 7.2.

The length of the two beams is 1.2 m. The height of the square cross section is 0.02 m and the thickness is 0.03 m.

When a certain load is reached, the structure undergoes buckling and the load-deflection curve shows a typical snap-back behavior, shown in Figure 7-3.

Arc-length method is required in order to capture the post-buckling behavior of the structure.



Figure 7.1 – Sketch representing the structure.

Lee's Frame Buckling Problem



Figure 7.2 – Finite element model with applied loads and boundary conditions.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Force	Linear	Nonlinear	-	Implicit	6-Arc length w/BFGS

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

The beam has a constant square section (0.1 m x 0.1 m) and a total length of 3.2 m and is meshed with 20 beams of equal length.

Material Data:

Mass Density	$\rho = 7.85 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 7.174 \times 10^{10} Pa$
Poisson's Ratio	v = 0.0

Load:

Load is applied incrementally to the structure until buckling occurs.

Element Types:

Hughes-Liu beam with cross section integration (elform=1)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA displacements U_x and U_y , at the location of the applied load (Node 56), plus the critical (buckling) load P_{crit} , are compared with *NAFEMS Non-Linear Benchmarks*, Test NL7.

	$\boldsymbol{U}_{\boldsymbol{X}}(\boldsymbol{m})$	$U_{Y}(m)$	$P_{crit}(N)$
NAFEMS NL7	-	0.4884	1.8485×10^{4}
Node 56	0.2620	0.4826	1.8228×10^{4}

These nodal displacement results were generated by *DATABASE_NODOUT keyword.

From *DATABASE_NODOUT results, it was also seen that the critical load increment is at 0.364568 of the total load, which would therefore correspond to a load of $P_{crit} = 0.364568 \times 5.0 \times 10^4 = 1.8228 \times 10^4 N$.

Figure 7.4 gives the X-displacement, Y-displacement, and resultant displacement versus load increment for Node 56.



Figure 7.3 – Displaced configuration at the buckling load.



Figure 7.4 – X-displacement, Y-displacement, and resultant displacement versus load increment for Node 56.

Input deck:

*KEYWORD *TITLE Lee's Frame Buckling Problem *CONTROL_IMPLICIT_AUTO \$# iauto iteopt itewin dtmin dtmax 1 20 5 1.000e-09 *CONTROL_IMPLICIT_GENERAL \$# imflag dt0 imform 1 0.003000 2 igs nsbs cnstn form 2 1 2 100 *CONTROL IMPLICIT SOLUTION maxref dctol ectol rctol lstol abstol 15 1.000e-06 1.000e-05 1.000e-05 0.990000 1.000000 istif nlprint \$# nsolvr ilimit 6 30 \$# dnorm diverg 2 1 \$# arcctl arcdir 0 1 1 2 arclen arcdmp arcmth 0.0 1 2 *CONTROL_TERMINATION \$# endtim endcyc 1.000000 0 *DATABASE_GLSTAT endeng 0.0 dtmin endmas 0.0 0.0 \$# dt binary 0.001000 1 *DATABASE_MALOS \$# dt binary 001000 1 *DATABASE_MATSUM *DATABASE_NODFOR \$# dt binary 1.0000e-04 1 *DATABASE_NODOUT \$# dt binary 1.0000e-04 1 *DATABASE_BINARY_D3PLOT \$# dt/cycl lcdt/nr 0.010000 0 beam npltc psetid 2 *DATABASE_NODAL_FORCE_GROUP \$# nsid cid 1 *DATABASE_HISTORY_NODE *DATABASE_11010101____ \$# nid1 nid2 nid4 ni5 nid6 nid7 nid8 nid3 56 36 1 *DEFINE_CURVE offo \$# lcid sdir sfa sfo offa dattyp 0 1.000000 1.000000 0.0 1 0.0 \$# al 01 0.0 0.0 1.0000000 5.0000000e+04 *ELEMENT BEAM n4 \$# eid pid n1 n2 n3 n5 nб n7 n8 1 0 1 1 2 3 0 0 0 2 31 1 32 56 61 0 0 0 0 2 *NODE \$# nid x У z tc rc 0.0 0.0 1 0.0 0.17999999 1.2000005 0.10000000 61 *BOUNDARY_SPC_NODE \$#nid/nsid cid dofx dofy dofz dofrx dofrz dofry 1 1 1 1 1 0 1 *BOUNDARY_SPC_NODE \$#nid/nsid cid dofx dofy dofz dofrx dofry dofrz 36 0 1 1 1 1 1 *PART \$# title \$# pid hgid grav secid mid eosid adpopt tmid 1 1 1 *SECTION_BEAM

\$#	secid	elform	shrf	qr/irid	cst	scoor		
	1	1	0.830000	5	0	0.0		
\$#	ts1	ts2	tt1	tt2	nsloc	ntloc		
0.	030000	0.030000	0.02	0.02				
*MAT	_ELASTI	C						
\$#	mid	ro	e	pr	da	db	not used	
	1	7850.0007	.1740e+10	0.0	0.0	0.0	0.0	
*LOA	D_NODE_	POINT						
\$#	node	dof	lcid	sf	cid	ml	m2	m3
	56	2	1	-1.000000				
*SET	_NODE_L	IST						
\$#	sid	dal	da2	da3	da4	solver		
	1	0.0	0.0	0.0	0.0			
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	1	б	36	56				
*END								

Notes:

8. Pin-Ended Double Cross: In-Plane Vibration

Keywords:

```
*CONTROL_IMPLICIT_AUTO
*CONTROL_IMPLICIT_EIGENVALUE
*CONTROL_IMPLICIT_GENERAL
*CONTROL_IMPLICIT_SOLVER
```

Description:

This example shows the behavior of beam elements in a modal analysis. The structure is a double cross pinned to the ground as show in Figure 8.1. All inner nodes have $U_z = R_x = R_y = 0$. On the outer nodes $U_x = U_y = U_z = R_x = R_y = 0$.

The finite element model is shown in Figure 8.2.

The problem requires the extraction of numerically close eigenvalues, making it an ideal benchmark to check the element formulation accuracy.

Each arm of the cross is modeled with 4 beams, for a total of 32 beams. The length of each arm is 5 m.



Figure 8.1 – Sketch representing the structure.

Pin-Ended Double Cross: In-Plane Vibration



Figure 8.2 – Finite element model with end point boundary conditions.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Modal	-	Linear	Linear	-	Implicit	Block Shift and Inverted Lanczos

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

Square section of the beams: 0.125 m x 0.125 m.

Material Data:

Mass Density	$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.00 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Element Types:

Hughes-Liu beam with cross section integration (elform=1) Belytschko-Schwer resultant beam (elform=2) Small displacement, linear Timoshenko beam with exact stiffness (elform=13)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA natural frequencies, first 16 (frequency in *Hertz*), and mode shapes (first 6) are compared with *Standard NAFEMS Benchmarks*, Test FV2.

Mode(s)	NAFEMS FV2 (Hz)	Hughes-Liu Beam (<i>Hz</i>)	Belytschko- Schwer Beam	Timoshenko Beam (<i>Hz</i>)
1	11.336	11.641	11.323	11.365
2, 3	17.709	19.080	17.621	17.803
4, 5, 6, 7, 8	17.709	19.115	17.649	17.832
9	45.345	51.691	44.833	45.620
10, 11	57.390	73.717	55.673	57.399
12, 13, 14, 15, 16	57.390	74.381	55.952	57.706

As seen in the results, using the LS-DYNA default beam (Hughes-Liu - elform=1) results in poor accuracy in the frequency calculation due to its omission of the first (no bending) and second (no rotary inertia) order terms (more elements are often needed in an attempt to overcome this limitation). The Hughes-Liu beam effectively generates a constant moment along its length, so, as with brick and shell elements, meshes need to be reasonably fine to achieve adequate accuracy.

The Belytschko-Schwer beam (elform=2) provides good frequency results throughout most of the range covered, with some minor differences at the higher frequency range. This element is often acceptable.

The Timoshenko beam (elform=13), with its inclusion of second order (rotary inertia and shear distortion) terms, provides very good results throughout the reported frequency range. This element formulation is generally recommended for this type of frequency analysis.

Eigenvalue Results:

From the *eigout* file, generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword:

Hughes-Liu beam (elform=1):

Pin-Ended Double Cross: In-Plane Vibration							
results	of eig	envalue	analysi	s:			
		freque	ncy				
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD			
1	5.349990E+03	7.314363E+01	1.164117E+01	8.590202E-02			
2	1.437182E+04	1.198825E+02	1.907989E+01	5.241119E-02			
3	1.437182E+04	1.198825E+02	1.907989E+01	5.241119E-02			
4	1.442423E+04	1.201009E+02	1.911465E+01	5.231589E-02			
5	1.442423E+04	1.201009E+02	1.911465E+01	5.231589E-02			
б	1.442423E+04	1.201009E+02	1.911465E+01	5.231589E-02			
7	1.442423E+04	1.201009E+02	1.911465E+01	5.231589E-02			
8	1.442423E+04	1.201009E+02	1.911465E+01	5.231589E-02			
9	1.054860E+05	3.247862E+02	5.169132E+01	1.934561E-02			
10	2.145341E+05	4.631783E+02	7.371711E+01	1.356537E-02			
11	2.145341E+05	4.631783E+02	7.371711E+01	1.356537E-02			
12	2.184158E+05	4.673498E+02	7.438102E+01	1.344429E-02			
13	2.184158E+05	4.673498E+02	7.438102E+01	1.344429E-02			
14	2.184158E+05	4.673498E+02	7.438102E+01	1.344429E-02			
15	2.184158E+05	4.673498E+02	7.438102E+01	1.344429E-02			
16	2.184158E+05	4.673498E+02	7.438102E+01	1.344429E-02			

Belytschko-Schwer beam (elform=2):

Pin-Ended Double Cross: In-Plane Vibration									
results	of eig	envalue	analysi	s:					
frequency									
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD					
1	5.061497E+03	7.114420E+01	1.132295E+01	8.831620E-02					
2	1.225757E+04	1.107139E+02	1.762066E+01	5.675155E-02					
3	1.225757E+04	1.107139E+02	1.762066E+01	5.675155E-02					
4	1.229691E+04	1.108915E+02	1.764892E+01	5.666068E-02					
5	1.229691E+04	1.108915E+02	1.764892E+01	5.666068E-02					
б	1.229691E+04	1.108915E+02	1.764892E+01	5.666068E-02					
7	1.229691E+04	1.108915E+02	1.764892E+01	5.666068E-02					
8	1.229691E+04	1.108915E+02	1.764892E+01	5.666068E-02					
9	7.935148E+04	2.816939E+02	4.483298E+01	2.230501E-02					
10	1.223614E+05	3.498019E+02	5.567270E+01	1.796212E-02					
11	1.223614E+05	3.498019E+02	5.567270E+01	1.796212E-02					
12	1.235904E+05	3.515543E+02	5.595161E+01	1.787259E-02					
13	1.235904E+05	3.515543E+02	5.595161E+01	1.787259E-02					
14	1.235904E+05	3.515543E+02	5.595161E+01	1.787259E-02					
15	1.235904E+05	3.515543E+02	5.595161E+01	1.787259E-02					
16	1.235904E+05	3.515543E+02	5.595161E+01	1.787259E-02					

Timoshenko beam (elform=13):

Pin-Ended Double Cross: In-Plane Vibration results of eigenvalue analysis:

		freque	ncy	
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD
1	5.099519E+03	7.141092E+01	1.136540E+01	8.798634E-02
2	1.251262E+04	1.118598E+02	1.780305E+01	5.617016E-02
3	1.251262E+04	1.118598E+02	1.780305E+01	5.617016E-02
4	1.255348E+04	1.120423E+02	1.783209E+01	5.607869E-02
5	1.255348E+04	1.120423E+02	1.783209E+01	5.607869E-02
б	1.255348E+04	1.120423E+02	1.783209E+01	5.607869E-02
7	1.255348E+04	1.120423E+02	1.783209E+01	5.607869E-02
8	1.255348E+04	1.120423E+02	1.783209E+01	5.607869E-02
9	8.216358E+04	2.866419E+02	4.562048E+01	2.191998E-02
10	1.300676E+05	3.606489E+02	5.739905E+01	1.742189E-02
11	1.300676E+05	3.606489E+02	5.739905E+01	1.742189E-02
12	1.314649E+05	3.625808E+02	5.770653E+01	1.732906E-02
13	1.314649E+05	3.625808E+02	5.770653E+01	1.732906E-02
14	1.314649E+05	3.625808E+02	5.770653E+01	1.732906E-02
15	1.314649E+05	3.625808E+02	5.770653E+01	1.732906E-02
16	1.314649E+05	3.625808E+02	5.770653E+01	1.732906E-02

Mode Shapes (first six):

From the *d3plot* file, generated by the *DATABASE_BINARY_D3PLOT keyword, the user can obtain the first six mode shapes (stick view) for the Hughes-Li beam (Figure 8.3), the Belytschko-Schwer Beam (Figure 8.4), and the Timoshenko beam (Figure 8.5). Displacement contouring of the first six mode shapes are given in Figures 8.6, 8.7, and 8.8 for these three element formulations.



Figure 8.3 - Mode shapes for Hughes-Liu beam (stick view).



Figure 8.4 - Mode shapes for Belytschko beam (stick view).



Figure 8.5 - Mode shapes for Timoshenko beam (stick view).



Figure 8.6 - Mode shapes for Hughes-Liu beam (displacement contouring).



Figure 8.7 - Mode shapes for Belytschko beam (displacement contouring).



Figure 8.8 - Mode shapes for Timoshenko beam (displacement contouring).

Input deck:

*KEY	WORD							
*TIT	LE Ended Dec	hla Guar	- Tro Dian	. Tribuctio	-			
*CON	Ended Dou TROL_IMPI	LICIT_AUT	s: In-Plan C	e Vibratio	n			
\$#	iauto 1	iteopt 11	itewin 15	dtmin 0.0	dtmax 0.0			
*CON	TROL_IMPI	LICIT_EIG	ENVALUE					
\$#	neig	center	lflag	lftend	rflag	rhtend	eigmth	shfscl
	16	11.000	0	-1.00e+29	0	1.00e+29	2	0.0
*CON	TROL_IMPI	LICIT_GEN	ERAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1 1	L.00e-04	2	1	2			
*CON	TROL_IMPI	LICIT_SOL	VER					
\$#	lsolvr	lprint	negev	order	drcm	drcprm	autospc	autotol
	16	1	1	0	1	0.0	1	0.0
*DAT	ABASE_BIN	JARY_D3PL	TC ,	7.				
\$# d	t/cycl	lcdt/nr	beam	npltc	psetid			
.0	010000	0	2					
^ 또L또I 승규	MENT_BEAN	1 		0 2		ъГ		7 0
Ş₩	1	1	1 1	2 113 0 0	114	115	110 11	/ 118 2 2
	T	T	1	۷ ۲	0	0	0	J 2
	30	1	61 7	4 76	0	0	0	n 2
*N∩D	52 F	T	01 /	4 70	0	0	0	5 2
\$#	nid		x	V		7	to r	-
ΥII	1		0.0	0.0		0.0		
	-		0.0	0.0		0.0		
	76	2.79029	131 -2	.20970869	1.000	00000		
*BOUI	NDARY_SPO	C_SET						
\$#nio	d/nsid	_ cid	dofx	dofy	dofz	dofrx	dofry	dofrz
	2	0	1	1	1	1	1	0
*BOUI	NDARY_SPO	C_SET						
\$#nio	d/nsid	cid	dofx	dofy	dofz	dofrx	dofry	dofrz
	1	0	0	0	1	1	1	0
*PAR'	Г							
\$# t	itle							
							_	
\$#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
+	1	. 1	1					
* SEC	TION_BEAN	4	-16					
Ş#	secia	eliorm	snri	qr/irid	CSt	scoor		
ė u	1	1	0.833333	2.0	0.0	0.0		
\$# 0	LSI 195000 (LSZ	LLL 0 125000	LLZ	nstoc	nuloc		
U د د ب	125000 (olform	0.125000	0.125000	0.0	0.0		
やり# ぐ	1	21101	0 833333	QI/1110	CSL 0	50001		
दद#	- -	iee	0.0555555 i++	ے ن	0 53	iet		
\$ 0.	01562502	0345e-05	2.0345e-05	4.0690e-05	0.01302083	150		
\$\$#	secid	elform	shrf	ar/irid	cst	scoor		
\$	1	13	0.833333	2	0	0.0		
\$\$#	a	iss	itt	- i	sa	ist		
\$ 0.	01562502.	0345e-05	2.0345e-05	4.0690e-05	0.01302083			
*MAT	_ELASTIC							
\$#	mid	ro	e	pr	da	db	not used	
	1 8	3000.0002	.0000e+11	0.300000	0.0	0.0	0.0	
*SET	_NODE_LIS	ST_GENERA	ΓE					
\$#	sid	dal	da2	da3	da4	solver		
	1	0.0	0.0	0.0	0.0			
\$#	blbeg	blend	b2beg	b2end	b3beg	b3end	b4beg	b4end
	2	4	6	24	27	29	31	42
	44	46	48	59	61	63	65	76
*SET	_NODE_LIS	ST .			_ .	-		
\$#	sid	da1	da2	da3	da4	solver		
ė II	2	0.0	0.0	0.0	0.0			
\$#	nıdl	nıd2	nid3	nid4	nid5	nidó	nid7	nid8
* דיאיד	Ţ	5	30	47	64	60	43	26
- БИД								

Notes:

1. The main difference among these element formulations is the inclusion of different second order terms for rotary inertia and shear distortion. The Euler (Belytschko-Schwer) beam model includes only the first order terms, lateral displacement and bending moment. The simple shear (Hughes-Liu) beam model includes only the translation first order term (no bending) plus shear distortion, while the Timoshenko beam model includes both rotary inertia and shear distortion in addition to the first order effects.

9. Simply Supported Thin Annular Plate (coarse mesh)

Keyword:

*CONTROL_IMPLICIT_EIGENVALUE *CONTROL_IMPLICIT_GENERAL

Description:

A simply-supported annular plate of thickness t=0.06 m is to be analyzed to determine the first nine natural frequencies. The inner radius is 1.8 m and the outer radius is 6.0 m. This coarse mesh analysis has 26 shell elements (circumferential) by 3 elements (radial). All nodes have $U_x = U_y = R_z = 0$. On the outer nodes $U_z = 0$.

A sketch representing the structure is shown below (Figure 9.1) along with the finite element model (Figure 9.2).



Figure 9.1 – Sketch representing the structure.

Simply Supported Thin Annular Plate (coarse mesh)



Figure 9.2 – Coarse mesh finite element model with simply supported boundary conditions on outer nodes.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Modal	-	Linear	Linear	-	Implicit	Block Shift and Inverted Lanczos

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

 $r_{\scriptscriptstyle o}=6~m\,,~r_{\scriptscriptstyle i}=1.8\,m\,,~t=0.06~m$

Material Data:

Mass Density	$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.00 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Element Types:

Fully integrated shell (elform=16)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA natural frequencies, first 10 (frequency in *Hertz*), and mode shapes (first 5) are compared with *NAFEMS Natural Frequency Benchmark* NF14.

Mode(s)	NAFEMS NF14 (Hz)	Coarse Mesh (Hz)		
1	1.870	1.806		
2, 3	5.137	5.423		
4, 5	9.673	10.179		
6	14.850	13.217		
7, 8	15.570	16.239		
9, 10	18.380	16.691		

It is seen that even with this rather coarse mesh refinement, the LS-DYNA natural frequency results provide a fair comparison with the *NAFEMS Selected Benchmarks for Natural Frequency Analysis*, NF14 test values.

Eigenvalue Results:

From the *eigout* file, generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword:

Simply	Supporte	ed Thin	Annular	Plate	(coarse	mesh)			
resu	ılts	o f	eige	n v a	l u e	a n a	lysi	s:	
					- freque	ncy			
	MODE	EIGEN	/ALUE	RAI	DIANS	C	YCLES	PI	ERIOD
	1	1.28707	78E+02	1.1344	195E+01	1.80	5604E+00	5.53	38311E-01
	2	1.16105	53E+03	3.4074	123E+01	5.42	3083E+00	1.84	13970E-01
	3	1.16105	53E+03	3.4074	123E+01	5.42	3083E+00	1.84	13970E-01
	4	4.09082	26E+03	6.3959	957E+01	1.01	7948E+01	9.82	23683E-02
	5	4.09082	27E+03	6.3959	957E+01	1.01	7948E+01	9.82	23683E-02
	6	6.89606	50E+03	8.3042	252E+01	1.32	1663E+01	7.56	56227E-02
	7	1.04112	22E+04	1.0203	354E+02	1.62	3944E+01	6.15	57849E-02
	8	1.04112	22E+04	1.0203	354E+02	1.62	3944E+01	6.15	57848E-02
	9	1.09978	37E+04	1.0487	707E+02	1.669	9070E+01	5.99	91361E-02
	10	1.09978	37E+04	1.0487	707E+02	1.66	9070E+01	5.99	91361E-02

Mode Shapes (first five):

Figures 9.3, 9.4, and 9.5 show the first 5 mode shapes with no contouring while Figures 9.6. 9.7, and 9.8 show the same 5 mode shapes with displacement contouring.

LS-DYNA eigenvalues at time 1.00000E+0 Freq = 1.8056



Figure 9.3 - Mode 1, 1.806 *Hz* (*NAFEMS* 1.870) - no contouring.

LS-DYNA eigenvalues at time 1.00000E+0 Freq = 5.4231





LS-DYNA eigenvalues at time 1.00000E+0 Freq = 10.179



Figure 9.5 - Modes 4 and 5, 10.179 *Hz* (*NAFEMS* 9.673) - no contouring.







Figure 9.7 - Modes 2 and 3, 5.423 Hz (NAFEMS 5.137) - displacement contouring.



Figure 9.8 - Modes 4 and 5, 10.179 Hz (NAFEMS 9.673) - displacement contouring.

Input deck:

*KEY *TIT Simp	WORD LE ly Supp	orted Thin	Annular E	Plate (coar	rse mesh)				
*CONTROL_IMPLICIT_EIGENVALUE									
\$#	neig 10	center 0.0	lflag 1	lftend 1.000000	rflag 1	rhtend 30.00000	eigm	th 2	shfscl 0.0
*CONTROL IMPLICIT GENERAL									
\$#	imflag 1	dt0 1.000000	imform	nsbs	igs	cnstn	form		
*CONTROL_TERMINATION									
\$# (endtim	endcyc	dtmin	endeng	endmas				
1.	000000	0	0.0	0.0	0.0				
*DATABASE_BINARY_D3PLOT									
\$# dt/cycl lcdt/nr beam npltc psetid									
1.000000									
*ELEMENT_SHELL									
\$#	eid	pid	nl r	12 n3	n4	n5	n6	n7	n8
	1	1	1 2	27 28	2				
	78	1	78 10	94 79	53				
*NOD	E					_			
ŞĦ	n1d 1	1 70000	X	y y			tC	rc	
	T	1.79999	995	0.0		0.0			
	104	5.82565	355 –1	43588233		0.0	3	1	
*BOUNDARY SPC SET					0.0	0	-		
\$#ni	d/nsid	cid	dofx	dofv	dofz	dofrx	dof	rv	dofrz
	1	0	1	1	1	0		0	1
*BOUNDARY_SPC_SET									
\$#nio	d/nsid	cid	dofx	dofy	dofz	dofrx	dof	dofry	
	2	0	1	1	0	0		0	1
*PART									
\$# title									
material type # 1 (Elastic)									
Ş#	pid	secid	mid	eosid	hgid	grav	adpo	pt	tmid
*000			Ţ						
SECTION_SHELL									
γπ	3001U	16	0 0	0	propt 1	Q1/1110	100	0	secyp 1
\$#	+1	±0 ±2	+3	+ 4	nloc	marea		0	-
0.	060000	0.060000	0.060000	0.060000	0	0.0			
*MAT ELASTIC									
\$# -		ro	e	pr	da	db	not us	ed	
	1	8000.0002	.0000e+11	0.300000	0.0	0.0	0	.0	
*SET_NODE_LIST_GENERATE									
\$#	sid	dal	da2	da3	da4	solver			
	1	0.0	0.0	0.0	0.0				
\$#	blbeg	blend	b2beg	b2end	b3beg	b3end	b4b	eg	b4end
79 104									
*SET_NODE_LIST_GENERATE									
\$ #	sid	dal	da2	da3	da4	solver			
¢#	⊿ h1bec	U.U hlerd	0.0 h2hca	0.0 h2end	0.0 h3hog	hland	h1h		h4ond
γĦ	Didey	78	DZDEG	DZEIIQ	naned	Dieid	040	cy.	DIGIO
*END	-	70							

Notes:

10. Simply Supported Thin Annular Plate (fine mesh)

Keyword:

*CONTROL_IMPLICIT_EIGENVALUE *CONTROL_IMPLICIT_GENERAL

Description:

A simply-supported annular plate of thickness t=0.06 m is to be analyzed to determine the first nine natural frequencies. The inner radius is 1.8 m and the outer radius is 6.0 m. This fine mesh analysis has 32 shell elements (circumferential) by 5 elements (radial). All nodes have $U_x = U_y = R_z = 0$. On the outer nodes $U_z = 0$.

A sketch representing the structure is shown below (Figure 10.1) along with the finite element model (Figure 10.2).



Figure 10.1 – Sketch representing the structure.
Simply Supported Thin Annular Plate (fine mesh)



Figure 10.2 – Fine mesh finite element model with simply supported boundary conditions on outer nodes.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Modal	-	Linear	Linear	-	Implicit	Block Shift and Inverted Lanczos

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

 $r_{\scriptscriptstyle o}=6~m\,,~r_{\scriptscriptstyle i}=1.8\,m\,,~t=0.06~m$

Material Data:

Mass Density	$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.00 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Element Types:

Fully integrated shell (elform=16)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA natural frequencies, first 10 (frequency in *Hertz*), and mode shapes (first 5) are compared with *NAFEMS Natural Frequency Benchmark* NF14.

Mode(s)	NAFEMS NF14 (Hz)	Fine Mesh (Hz)
1	1.870	1.867
2, 3	5.137	5.197
4, 5	9.673	9.801
6	14.850	14.471
7, 8	15.570	15.665
9, 10	18.380	17.798

It is seen that with only a slight increase in mesh refinement, the LS-DYNA natural frequency results compare nicely with the *NAFEMS Selected Benchmarks for Natural Frequency Analysis*, NF14 test values.

Eigenvalue Results:

From the *eigout* file, generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword:

Simply	Supporte	ed Thin	Annular	Plate	(fine me	sh)		
resu	lts	o f	e i g e	n v a	lue	analy	s i s:	
					- frequen	су		
	MODE	EIGEN	/ALUE	RAI	DIANS	CYCLES	3	PERIOD
	1	1.37701	9E+02	1.1734	465E+01	1.867627E	2+00	5.354388E-01
	2	1.06628	33E+03	3.2653	399E+01	5.197045E	2+00	1.924171E-01
	3	1.06628	33E+03	3.2653	399E+01	5.197045E	2+00	1.924171E-01
	4	3.79194	46E+03	6.1578	878E+01	9.800567E	2+00	1.020349E-01
	5	3.79194	46E+03	6.1578	878E+01	9.800567E	2+00	1.020349E-01
	6	8.26719	93E+03	9.0924	11E+01	1.447102E	2+01	6.910362E-02
	7	9.68814	13E+03	9.8428	36E+01	1.566536E	2+01	6.383511E-02
	8	9.68814	13E+03	9.8428	36E+01	1.566536E	2+01	6.383511E-02
	9	1.25054	15E+04	1.1182	278E+02	1.779794E	2+01	5.618626E-02
	10	1.25054	15E+04	1.1182	278E+02	1.779794E	2+01	5.618626E-02

Mode Shapes (first three):

Figures 10.3, 10.4, and 10.5 show the first 5 mode shapes with no contouring while Figures 10.6. 10.7, and 10.8 show the same 5 mode shapes with displacement contouring.





Figure 10.3 - Mode 1, 1.867 *Hz* (*NAFEMS* 1.870) - no contouring.

LS-DYNA eigenvalues at time 1.00000E+0 Freq = 5.197





LS-DYNA eigenvalues at time 1.00000E+0 Freq = 9.8006



Figure 10.5 - Modes 4 and 5, 9.801 *Hz* (*NAFEMS* 9.673) - no contouring.







Figure 10.7 - Modes 2 and 3, 5.197 Hz (NAFEMS 5.137) - displacement contouring.



Figure 10.8 - Modes 4 and 5, 9.801 Hz (NAFEMS 9.673) - displacement contouring.

Input deck:

*KEYV *TITI	WORD LE LV Supp	orted Thin	Annular D	late (find	mech)				
*CON	ту Бирр Грој, тм	PLICIT EIG	ENVALUE	Tace (IIII					
\$#	neia	center	lflag	lftend	rflag	rhtend	eid	amth	shfscl
4.0	10	0.0	1	1.000000	1	30,00000		2	0.0
*CON	TROL IM	PLICIT GEN	ERAL						
\$# :	imflag 1	dt0 1.000000	imform	nsbs	igs	cnstn	t	Eorm	
*CONT	TROL_TE	RMINATION							
\$# 6	endtim	endcyc	dtmin	endeng	endmas				
1.0	000000	0	0.0	0.0	0.0				
*DAT	ABASE_B	INARY_D3PL	OT						
\$# dt	t/cycl	lcdt/nr	beam	npltc	psetid				
1.0	000000								
*ELEN	MENT_SH	ELL							
\$#	eid	pid	nl n	.2 n3	n4	n5	nб	n7	n8
	1	1	1 3	3 34	2				
	192	1	192 22	4 193	161				
*NODI	E								
\$#	nid		х	У		Z	tc	rc	
	1	1.79999	995	0.0		0.0			
	224	5.88471	174 -1	.17054188		0.0	3	1	
*BOUI	NDARY_S	PC_SET							
\$#nic	d/nsid	cid	dofx	dofy	dofz	dofrx	do	ofry	dofrz
	1	0	1	1	1	0		0	1
*BOUI	NDARY_S	PC_SET					_		
\$#nic	d/nsid	cid	dofx	dofy	dofz	dofrx	do	ofry	dofrz
	_ 2	0	1	1	0	0		0	1
*PAR'	Г 								
\$# t:	itle								
mate	rial ty	pe # 1 (E	lastic)		1				6 d. d
Ş₩	p10 1	secia	1	eosid	ngra	grav	adj	popu	Unita
* 0			T						
ະອະດ. ເ	secid	elform	chrf	nin	propt	ar/irid	i,	aomo	cetim
γπ	1	61101111	0 0	0	propt 1	91/1110 0 0	ΞV	0	secyp 1
\$#	+1	+2	+3	+ 4	nloc	marea		0	±
0.(060000	0.060000	0.060000	0.060000	0	0.0			
*MAT	ELASTI	C	0.000000	0.000000	0	0.0			
\$#		ro	e	pr	da	db	not i	ised	
4.0	1	8000.0002	.0000e+11	0.300000	0.0	0.0		0.0	
*SET	NODE L	IST GENERA	TE						
\$# -	sid	da1	da2	da3	da4	solver			
	1	0.0	0.0	0.0	0.0				
\$#	blbeg	blend	b2beg	b2end	b3beg	b3end	b	4beg	b4end
	193	224	-		-				
*SET_	_NODE_L	IST_GENERA	TE						
\$#	sid	dal	da2	da3	da4	solver			
	2	0.0	0.0	0.0	0.0				
\$#	blbeg	blend	b2beg	b2end	b3beg	b3end	b	4beg	b4end
	1	192							
*END									

Notes:

11. Transient Response to a Constant Force

Keyword:

```
*CONTROL_IMPLICIT_DYNAMICS
```

*CONTROL_IMPLICIT_GENERAL

*CONTROL_IMPLICIT_SOLVER

*CONTROL_IMPLICIT_SOLUTION

Description:

A mass m=25.9067 *lbf-s²/in* is attached in the middle of a steel beam of length 1=240 inches and geometric properties shown below. The beam is subjected to a dynamic load F(t) with a rise time of 0.075 seconds and a maximum constant value of 2000 pound-force. The weight of the beam is considered negligible. Determine the time of maximum displacement response t_{max} and the maximum displacement response y_{max} . Additionally, determine the maximum bending stress σ_{bend} in the beam. The attached mass is modeled with a lumped mass element at the central node of the beam.

A sketch representing the structure is shown below (Figure 11.1) along with the finite element model (Figure 11.2).



Figure 11.1 – Sketch representing the structure.



, X

Figure 11.2 – Finite element model with applied load (Node 12) and boundary conditions. In-plane boundary conditions and lumped mass (Node 12) are not shown.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Force	Linear	Linear	-	Implicit	2-Nonlinear w/BFGS

Units:

 $lbf-s^2/in$, in, s, lbf, psi, lbf-in (blob, inch, second, pound force, pound force/inch², pound force-inch)

Dimensional Data:

 $L = 240.0 \text{ in}, h = 18.0 \text{ in}, I_z = 800.6 \text{ in}^4$

As a cross-section-integrated beam is used, the cross sectional dimension is calculated. Given $I_z = 800.6 in^4$ and h = 18.0 in, a thickness of t = 1.647 in is obtained.

Material Data:

$\rho = 1.0 \times 10^{-20} \ lbf - s^2 / in^4$
$E = 3.0 \times 10^7 \ psi$
v = 0.3
$m = 25.9067 \ blobs$

Load:

Lateral Load $F(t) = *DEFINE_CURVE$

Element Types:

Hughes-Liu beam with cross section integration (elform=1)

Lumped mass (*ELEMENT_MASS entry)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA results for time of the maximum displacement response t_{max} , the maximum displacement response y_{max} , and the maximum bending stress p_{end} in the beam are compared with J.M. Biggs' studies in *Introduction to Structural Dynamics* (pg. 50).

	Time - t_{max} (s)	Disp. - y_{max} (<i>in</i>)	Stress - \mathcal{B}_{bend} (psi)		
Biggs	0.0920	0.3310	1.8600×10^4		
Node 12/Element 10	ode 12/Element 10 0.0930		1.8151×10 ⁴		

These nodal time/displacement results (Figure 11.3) were generated by *DATABASE_ NODOUT keyword while the element stress results (Figure 11.4) were generated by *DATABASE_ELOUT.

LS-DYNA stress and strain output corresponds to integration point locations.

Lobatto integration (qr=4 - 3×3 quadrature - *SECTION_BEAM) was employed since it has an advantage in that the inner and outer integration points are on the beam surfaces. Gauss integration is the default quadrature rule (qr=2 - 2×2 quadrature - *SECTION_BEAM).

The contour plots of the axial beam stresses for the upper (ip=1) and lower (ip=3) surfaces of the beam (Figure 11.5) were obtained from the *d3plot* file at t=0.093 s which were generated by the *DATABASE_BINARY_D3PLOT keyword.



Figure 11.3 – Y-displacement vs. time for Node 12 at the center of the beam.



Figure 11.4 – Axial beam stress vs. time at the upper (ip=1), the point of maximum bending stress, and the lower (ip=3) beam surfaces for Element 10.



ě.

Figure 11.5 – Contour plots of the axial beam stresses for the upper (ip=1) and lower (ip=3) surfaces of the beam at t=0.093 s.

Input deck:

*KE *TT	YWORD							
Trai	nsient Re NTROL IMP	esponse to	o a Constan NAMICS	nt Force				
\$#	imass	gamma	beta					
¥ 11	1	0.500000	0.250000					
*COI	NTROL_IM	PLICIT_GEN	NERAL					
\$#	imflag	dt0	imform	nsbs	iqs	cnstn	form	
	1	0.001000	2	1	2			
*COI	NTROL IM	PLICIT SO	LVER					
\$#	lsolvr	lprint	negev	order	drcm	drcprm	autospc	autotol
	4	- 2	2	0	1	0.0	1	0.0
\$#	lcpack							
	2							
*COI	NTROL IM	PLICIT SO	LUTION					
\$#	nsolvr	ilimit.	maxref	dctol	ectol	rctol	lstol	abstol
4.0	2	11	15	0.0010	0.0100	1.00e+10	0.900000	1.00e-10
¢#	dnorm	divera	ietif	nlprint	0.0100	1.000.10	0.00000	1.000 10
Y 11	2	1	10011	2				
¢#	arcatl	aradir	aralan	arcmth	arcdmp			
ŶΠ	arccer 0	1	0 0	1	ar camp			
*001	ע אידים∩ו. ידוים		0.0	T	2			
¢#	endtim	endava	dtmin	endena	endmag			
ν π	100000	enacyc		endeng	0 0			
יגרד *	.IUUUUU TADACE EI		0.900000	0.0	0.0			
оDA.	IADASE_EI							
⇒# 1 ∩/		DINARY						
1.U	000e-05							
*DA	IABASE_MA	AISUM						
\$# 1 0/		binary						
1.0	000e-05	1						
*DA:	TABASE_NO	DDOU'I'						
\$# •	at	binary						
1.00	000e-05	1						
*DA	TABASE_SE	PCFORC						
\$#	dt	binary						
1.00	000e-05	Ţ						
*DA	TABASE_BI	INARY_D3PI	LOT .	-				
\$# (dt/cycl	lcdt/nr	beam	npltc	psetid			
1.0	000e-05							
*DA	TABASE_BI	INARY_D3TH	HDT .	-				
\$# (dt/cycl	lcdt/nr	beam	npltc	psetid			
2.0	000e-04							
*DA	TABASE_EX	TENT_BINA	ARY					
\$#	neiph	neips	maxint	strflg	sigflg	epsflg	rltflg	engflg
	0	0	0	0	1	1	1	1
\$#	cmpflg	ieverp	beamip	dcomp	shge	stssz	n3thdt	ialemat
	0	0	20	1	1	1	2	
*DA	TABASE_HI	ISTORY_NOI	DE					
\$#	nidl	nid2	nid3	nid4	ni5	nid6	nid7	nid8
	12	1	2					
*DA	TABASE_HI	ISTORY_BEA	MA					
\$#	eid1	eid2	eid3	eid4	ei5	eid6	eid7	eid8
	10	11						
*DEI	FINE_CURV	/E						
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp	
	1	0	1.000000	1.000000	0.0	0.0		
\$#		al		01				
		0.0		0.0				
	0.	.07500000	2.00	000000e+04				
	1.	.00000000	2.00	000000e+04				
*ELI	EMENT_BE	MA						
\$#	eid	pid	nl i	n2 n3	n4	n5	n6 n'	7 n8
	1	1	1	3 22				
	20	1	21	2 41				
*ELI	EMENT_MAS	SS						
\$#	eid	id	mas	ss pid				
	23	12	25.906700	13				

*NOD	E							
\$#	nid		x	У		Z	tc :	rc
	1	(0.0	0.0		0.0		
	41	234.0000	000	0.0	0.9999	96525		
*BOU	NDARY_SP	C_SET						
\$#ni	d/nsid	cid	dofx	dofy	dofz	dofrx	dofry	dofrz
	2	0	1	1	1	1	1	
*BOU	NDARY_SP	C_SET						
\$#ni	d/nsid	cid	dofx	dofy	dofz	dofrx	dofry	dofrz
	3	0	0	0	1	1	1	
*PAR	Т							
\$# t	itle							
Part		1 for Ma	at	1 and Elem	Туре	1		
\$#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
	1	1	1					
*SEC	TION_BEA	M						
\$#	secid	elform	shrf	qr/irid	cst	scoor		
	1	1	0.830000	4	0	0.0		
\$#	ts1	ts2	tt1	tt2	nsloc	ntloc		
1.	647220	1.647220	18.0	18.0				
*MAT	_ELASTIC							
\$#	mid	ro	e	pr	da	db	not used	
	11.	0000e-203	.0000e+07	0.300000	0.0	0.0	0.0	
*LOA	D_NODE_S	ET						
\$#	nsid	dof	lcid	sf	cid	ml	m2	m3
	1	2	1	1.000000				
*SET	_NODE_LI	ST						
\$#	sid	dal	da2	da3	da4	solver		
	1	0.0	0.0	0.0	0.0			
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	12							
*SET	_NODE_LI	ST						
\$#	sid	dal	da2	da3	da4	solver		
	2	0.0	0.0	0.0	0.0			
\$#	nidl	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	1	2						
*SET	_NODE_LI	ST						
\$#	sid	dal	da2	da3	da4	solver		
	3	0.0	0.0	0.0	0.0			
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	3	4	5	6	7	8	9	10
	11	12	13	14	15	16	17	18
	19	20	21	22	23	24	25	26
	27	28	29	30	31	32	33	34
	35	36	37	38	39	40	41	
*END								

Notes:

12. Simply Supported Square Plate: Out-of-Plane Vibration (solid mesh)

Keywords:

*CONTROL_IMPLICIT_EIGENVALUE *CONTROL_IMPLICIT_GENERAL

Description:

Determine the first 10 natural frequencies of a solid simply-supported plate of thickness t=1.0 m. Each side of the plate measure 10.0 m. The plate is meshed with solid hexahedra element with an 8 x 8 x 3 density. On the lower surface, outer boundary nodes, $U_z = 0$.

The finite element model is shown in Figure 12.1.

Simply Supported Square Plate: Out-of-Plane Vibration (solid mesh)



Figure 12.1 – Finite element model with simply supported boundary conditions on lower surface, outer nodes.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Modal	-	Linear	Linear	-	Implicit	Block Shift and Inverted Lanczos

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

Rectangular dimensions of square plate: 10.0 m x 10.0 m x 1.00 m.

Material Data:

Mass Density	$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.00 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Element Types:

Constant stress solid (elform=1) Fully integrated S/R solid (elform=2) Fully integrated S/R solid - for poor aspect ratio (eff) - (elform=-1) Fully integrated S/R solid - for poor aspect ratio (acc) - (elform=-2) Fully integrated quadratic 8 node element with nodal rotations (elform=3)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA natural frequencies, first 10 (frequency in *Hertz*), and mode shapes (4 through 10) are compared with *NAFEMS Free Vibration Benchmarks*, Test FV52.

Mode(s)	NAFEMS FV52 (Hz)	elform=1 (Hz)	elform=2 (<i>Hz</i>)	elform=-1 (Hz)	elform=-2 (Hz)	elform=3 (Hz)
1, 2, 3	rigid body	rigid body	rigid body	rigid body	rigid body	rigid body
4	45.897	44.040	48.508	45.448	46.2060	43.370
5, 6	109.440	106.468	120.388	107.114	109.265	104.703
7	167.890	155.523	169.601	159.102	163.943	153.862
8	193.590	193.582	193.526	193.518	193.526	193.227
9	206.190	200.135	200.188	198.791	200.176	196.485
10	206.190	200.135	200.188	198.873	200.176	197.280

Hourglass control (*HOURGLASS) is necessary for the constant stress solid (elform=1) element formulation (the LS-DYNA default), especially at higher frequencies. Only this element formulation (elform=1) makes use of this feature

The constant stress solid (elform=1), the fully integrated S/R solid (elform=2), and the fully integrated quadratic 8 node element with nodal rotations (elform=3) all provide similar frequency results for this analysis.

The fully integrated quadratic 8 node element with nodal rotations (elform=3) formulation provides two distinct modes and frequencies for modes 9 and 10, whereas, all the other formulations provide the same results for modes 9 and 10. This is perhaps due to the accountability of the nodal rotations.

The aspect ratio of these elements is 3.75 (ratio of side to depth length). It would, however, for this frequency analysis, appear that the element formulations available to address poor aspect ratios (elform=-1 or -2) do not offer much improvement. The constant stress solid (elform=1), the fully integrated S/R solid (elform=2), and the fully integrated quadratic 8 node element with nodal rotations (elform=3) formulation all appear to provide more than adequate results for this frequency study.

The fully integrated S/R solid (the efficient formulation choice) intended to address poor aspect ratios (elform=-1), provided somewhat different shapes for modes 9 and 10. This formulation involves a slight modification of the Jacobian matrix which can lead to a

stiffness reduction for certain modes (according to Borrvall [2009]). However, modes 9 and 10 are not those modes Borrvall offered concerns for in stiffness reduction.

Eigenvalue Results:

From the *eigout* file, generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword:

Constant stress solid (elform=1):

Simply Supported Square Plate: Out-of-Plane Vibration results of eigenvalue analvsis: |----- frequency -----| MODE EIGENVALUE RADIANS CYCLES PERIOD 1 -3.201421E-10 1.789252E-05 2.847682E-06 3.511628E+05 2 1.062290E-09 3.259279E-05 5.187303E-06 1.927784E+05 2.881279E-09 5.367755E-05 8.543047E-06 1.170543E+05 3 7.657028E+04 2.767133E+02 4.404030E+01 2.270648E-02 4 5 4.475080E+05 6.689604E+02 1.064684E+02 9.392462E-03 4.475080E+05 6.689604E+02 1.064684E+02 6 9.392462E-03 6.429910E-03 7 9.548827E+05 9.771810E+02 1.555232E+02 1.479411E+06 1.216310E+03 1.935818E+02 5.165775E-03 1.581263E+06 1.257483E+03 2.001346E+02 4.996637E-03 8 9 1.581263E+06 1.257483E+03 2.001346E+02 4.996637E-03 10

Fully integrated S/R solid (elform=2)

Simply Supported Square Plate: Out-of-Plane Vibration results of eigenvalue analysis:

		freque	ncy	
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD
1	-6.009941E-09	7.752381E-05	1.233830E-05	8.104846E+04
2	9.895302E-10	3.145680E-05	5.006505E-06	1.997401E+05
3	5.558832E-09	7.455757E-05	1.186621E-05	8.427293E+04
4	9.289190E+04	3.047817E+02	4.850752E+01	2.061536E-02
5	5.721748E+05	7.564224E+02	1.203884E+02	8.306451E-03
б	5.721748E+05	7.564224E+02	1.203884E+02	8.306451E-03
7	1.135577E+06	1.065635E+03	1.696010E+02	5.896191E-03
8	1.478563E+06	1.215962E+03	1.935263E+02	5.167257E-03
9	1.582107E+06	1.257818E+03	2.001880E+02	4.995304E-03
10	1.582107E+06	1.257818E+03	2.001880E+02	4.995304E-03

Fully integrated S/R solid - for poor aspect ratio (eff) - (elform=-1)

Simply Supported Square Plate: Out-of-Plane Vibration results of eigenvalue analysis:

		freque	navl	
		Incident	licy	
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD
1	1.043372E-08	1.021456E-04	1.625698E-05	6.151205E+04
2	1.335866E-08	1.155797E-04	1.839507E-05	5.436238E+04
3	1.573062E-08	1.254218E-04	1.996149E-05	5.009645E+04
4	8.154482E+04	2.855605E+02	4.544837E+01	2.200299E-02
5	4.529502E+05	6.730158E+02	1.071138E+02	9.335866E-03
б	4.529502E+05	6.730158E+02	1.071138E+02	9.335866E-03
7	9.993359E+05	9.996679E+02	1.591021E+02	6.285273E-03
8	1.478432E+06	1.215908E+03	1.935178E+02	5.167484E-03
9	1.560100E+06	1.249040E+03	1.987908E+02	5.030413E-03
10	1.561382E+06	1.249553E+03	1.988725E+02	5.028347E-03

Simply Supported Square Plate: Out-of-Plane Vibration							
resul	t s	of eige	nvalue	analysi	s:		
			frequen	cy			
MOI	DE	EIGENVALUE	RADIANS	CYCLES	PERIOD		
	1 2	2.211891E-09	4.703075E-05	7.485176E-06	1.335974E+05		
	2 7	7.014023E-09	8.374977E-05	1.332919E-05	7.502332E+04		
	3 1	L.121953E-08	1.059223E-04	1.685805E-05	5.931883E+04		
	4 8	3.428602E+04	2.903205E+02	4.620595E+01	2.164223E-02		
	5 4	4.713255E+05	6.865315E+02	1.092649E+02	9.152072E-03		
	6 4	4.713255E+05	6.865315E+02	1.092649E+02	9.152072E-03		
	7 1	L.061078E+06	1.030087E+03	1.639434E+02	6.099667E-03		
	8 1	L.478558E+06	1.215960E+03	1.935260E+02	5.167264E-03		
	9 1	L.581923E+06	1.257745E+03	2.001764E+02	4.995595E-03		
:	10 1	L.581923E+06	1.257745E+03	2.001764E+02	4.995595E-03		

Fully integrated S/R solid - for poor aspect ratio (acc) - (elform=-2)

Fully integrated quadratic 8 node element with nodal rotations (elform=3)

```
Simply Supported Square Plate
results of eigenvalue analysis:
                           |----- frequency -----|
              EIGENVALUE
       MODE
                               RADIANS
                                              CYCLES
                                                            PERIOD
              1.877197E-08 1.370108E-04
          1
                                          2.180595E-05 4.585904E+04
          2 2.239540E-08 1.496509E-04 2.381768E-05 4.198561E+04
             2.614979E-08 1.617090E-04 2.573678E-05
7.425572E+04 2.724990E+02 4.336957E+01
                                                         3.885490E+04
          3
                                                          2.305764E-02
          4
             4.327897E+05 6.578675E+02 1.047029E+02
          5
                                                         9.550837E-03
          6
             4.327897E+05 6.578675E+02 1.047029E+02
                                                         9.550837E-03
              9.345968E+05 9.667455E+02 1.538623E+02
1.473997E+06 1.214083E+03 1.932273E+02
          7
              9.345968E+05
                                                         6.499317E-03
          8
                                                         5.175253E-03
             1.524111E+06 1.234549E+03 1.964846E+02 5.089458E-03
          9
         10 1.536475E+06 1.239546E+03 1.972799E+02 5.068940E-03
```

Mode Shapes:

The constant stress solid (elform=1) mode shapes are shown in Figure 12.2, the fully integrated S/R solid (elform=2) in Figure 12.3, the fully integrated quadratic 8 node element with nodal rotations (elform=3) in Figure 12.4, the fully integrated S/R solid (the so-called efficient formulation choice) intended to address poor aspect ratios (elform=-1) in Figure 12.5, and the fully integrated S/R solid (the so-called accurate formulation choice) intended to address poor aspect ratios (elform=-2) in Figure 12.6.

The first three modes are not shown (rigid body translations). Modes 4 through 10 are shown for the selected results. Modes 5 and 6 are identical for all element formulations; modes 9 and 10 are also identical for all, with the exception of the fully integrated quadratic 8 node element with nodal rotations (elform=3) formulation.



Figure 12.2 - Mode shapes for constant stress solid (elform=1).



Figure 12.3 - Mode shapes for fully integrated S/R solid (elform=2).



Figure 12.4 - Mode shapes for fully integrated quadratic 8 node element with nodal rotations (elform=3).



Figure 12.5 - Mode shapes for fully integrated S/R solid (elform=-1).



Figure 12.6 - Mode shapes fully integrated S/R solid (elform=-2).

Input deck:

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*C0	NTROL_IM	PLICIT_GE	NERAI	_						
\$#	imflag	dt0	i	imform	nsbs	igs	s cn	stn	form	
	1	0.0								
*C0	NTROL_TE	RMINATION								
\$#	endtim	endcyc		dtmin	endeng	endmas	5			
1	.000000	0		0.0	0.0	0.0)			
*DA	TABASE_B	INARY_D3P	LOT							
\$#	dt/cycl	lcdt/nr		beam	npltc	psetic	1			
1	.000000									
*EL	EMENT_SO	LID	_				_			
\$#	eid	pid	nl	n	2 n3	n4	n5	n6	n7	n8
	1	1	1	3.	7 41	5	2	38	8 42	6
	192	1	283	31	9 323	287	284	320	324	288
*NO	DE									
\$#	nid		х		У		Z	to	e rc	
	1		0.0		0.0		0.0	3	5	
	204	10 0000		1.0		1 0/				
+ -	324		0000	10	.00000000	1.00	1000000			
	id/paid	PC_SEI		dofr	dofr	dof	- do	from	dofmi	dofwr
9 #11	10/11510	010		uorx 0	doly	u012	2 ao	TTX	dolly	GOLLZ
*D7	т рт	0		0	0	-	L			
¢#	+i+l_									
mat	erial tvi	pe # 1 (1	Elast	ic)						
\$#	pid	secid		mid	eosid	hgio	d g	rav	adpopt	tmid

	1	1	1	0	1				
*SEC	*SECTION_SOLID								
\$#	secid	elform	aet						
	1	1	1						
\$	1	2	1						
\$	1	-1	1						
\$	1	-2	1						
\$	1	3	1						
*MAT_	_ELASTIC	2							
\$#	mid	ro	е	pr	da	db	not used		
	1	8000.0002.	.0000e+11	0.300000	0.0	0.0	0.0		
*HOUI	RGLASS								
\$#	hgid	ihq	qm	ibq	q1	q2	db	đM	
	1	6	1.0	0	0.0	0.0	0.0	0.0	
*SET	_NODE_LI	IST							
\$#	sid	da1	da2	da3	da4	solver			
	1	0.0	0.0	0.0	0.0				
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8	
	1	37	73	109	145	181	217	253	
	289	293	297	301	305	309	313	317	
	321	285	249	213	177	141	105	69	
	33	29	25	21	17	13	9	5	
*END									

Notes:

13. Simply Supported Square Plate: Out-of-Plane Vibration (thick shell mesh)

Keywords:

*CONTROL_IMPLICIT_EIGENVALUE *CONTROL_IMPLICIT_GENERAL

Description:

Determine the first 10 natural frequencies of a solid simply-supported plate of thickness t=1.0 m. Each side of the plate measure 10.0 m. The plate is meshed with solid hexahedra element with an 8 x 8 x 3 density. On the lower surface, outer boundary nodes, $U_z = 0$.

The finite element model is shown in Figure 13.1.





Figure 13.1 – Finite element model with simply supported boundary conditions on lower surface, outer nodes.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Modal	-	Linear	Linear	-	Implicit	Block Shift and Inverted Lanczos

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

Rectangular dimensions of square plate: 10.0 m x 10.0 m x 1.00 m.

Material Data:

Mass Density	$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.00 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3

Element Types:

S/R 2x2 IPI thick shell (elform=2) Assumed strain 2x2 IPI thick shell (elform=3) Assumed strain RI thick shell (elform=5)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA natural frequencies, first 10 (frequency in *Hertz*), and mode shapes (4 through 10) are compared with *NAFEMS Free Vibration Benchmarks*, Test FV52.

Mode(s)	NAFEMS FV52 (Hz)	elform=2 (Hz)	elform=3 (Hz)	elform=5 (Hz)
1, 2, 3	rigid body	rigid body	rigid body	rigid body
4	45.897	43.480	42.556	44.656
5, 6	109.440	105.363	102.983	107.290
7	167.890	152.764	150.187	157.397
8	193.590	185.301	193.378	193.583
9	206.190	193.708	197.247	200.136
10	206.190	200.997	197.295	200.136

The assumed strain 2x2 IPI thick shell (elform=3) and the assumed strain RI thick shell (elform=5) use a full three-dimensional stress update rather than the two-dimensional plane stress update of the one point reduced integration (elform=1) and the selectively reduced 2x2 IPI thick shell (elform=2).

The selectively reduced 2x2 IPI thick shell (elform=2), the assumed strain 2x2 IPI thick shell (elform=3), and the assumed strain RI thick shell (elform=5) all provide similar frequency results for this analysis.

The selectively reduced 2x2 IPI thick shell (elform=2) formulation appears to have identified (added) an unexpected result for mode 8 (an anomaly, a low energy warping mode which is believed will not cause solution troubles) due to the calculation of the out-of-plane shear stiffness terms. Code inspection indicated that the out-of-plane shear stress and stiffness is calculated at the mid-point rather than the 2x2 integration points in order to prevent shear locking in bending. Modes 9 and 10 (elform=2) are however, similar to modes 8 and 9, respectively, for the assumed strain RI thick shell (elform=5) formulation.

The assumed strain 2x2 IPI thick shell (elform=3) formulation appears to have identified two different modes and frequencies for modes 9 and 10. This is possibly due to the employment of a corotational system that rotates with the elements, which suppresses the element locking.

Eigenvalue Results:

From the *eigout* file, generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword:

S/R 2x2 IPI thick shell (elform=2)

Simply Supported Square Plate: Out-of-Plane Vibration (thick shell mesh) results of eigenvalue analysis:

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.

		freque	ncy	
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD
1	-8.469215E-09	9.202834E-05	1.464676E-05	6.827446E+04
2	-3.463356E-09	5.885028E-05	9.366313E-06	1.067656E+05
3	-1.746230E-09	4.178791E-05	6.650753E-06	1.503589E+05
4	7.449258E+04	2.729333E+02	4.343868E+01	2.302096E-02
5	4.377512E+05	6.616277E+02	1.053013E+02	9.496558E-03
6	4.377512E+05	6.616277E+02	1.053013E+02	9.496558E-03
7	9.193747E+05	9.588403E+02	1.526042E+02	6.552901E-03
8	1.255008E+06	1.120271E+03	1.782967E+02	5.608628E-03
9	1.481334E+06	1.217101E+03	1.937076E+02	5.162420E-03
10	1.594925E+06	1.262903E+03	2.009973E+02	4.975191E-03

Assumed strain 2x2 IPI thick shell (elform=3)

Simply Supported Square Plate: Out-of-Plane Vibration (thick shell mesh) results of eigenvalue analysis:

		frequer	ncy	
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD
1	-8.338247E-09	9.131400E-05	1.453308E-05	6.880856E+04
2	-4.802132E-09	6.929742E-05	1.102903E-05	9.066983E+04
3	-3.012246E-09	5.488394E-05	8.735050E-06	1.144813E+05
4	7.149598E+04	2.673873E+02	4.255601E+01	2.349844E-02
5	4.186854E+05	6.470590E+02	1.029826E+02	9.710374E-03
6	4.186854E+05	6.470590E+02	1.029826E+02	9.710374E-03
7	8.904817E+05	9.436534E+02	1.501871E+02	6.658361E-03
8	1.476299E+06	1.215030E+03	1.933781E+02	5.171217E-03
9	1.535967E+06	1.239341E+03	1.972473E+02	5.069778E-03
10	1.536702E+06	1.239638E+03	1.972945E+02	5.068565E-03

Assumed strain RI thick shell (elform=5)

Simply Supported Square Plate: Out-of-Plane Vibration (thick shell mesh) results of eigenvalue analysis:

		frequer	ncy	
MODE	EIGENVALUE	RADIANS	CYCLES	PERIOD
1	5.966285E-10	2.442598E-05	3.887516E-06	2.572337E+05
2	2.881279E-09	5.367755E-05	8.543047E-06	1.170543E+05
3	5.762558E-09	7.591152E-05	1.208169E-05	8.276986E+04
4	7.872674E+04	2.805829E+02	4.465615E+01	2.239333E-02
5	4.544420E+05	6.741231E+02	1.072900E+02	9.320531E-03
6	4.544420E+05	6.741231E+02	1.072900E+02	9.320531E-03
7	9.780258E+05	9.889519E+02	1.573966E+02	6.353378E-03
8	1.479432E+06	1.216319E+03	1.935832E+02	5.165738E-03
9	1.581277E+06	1.257488E+03	2.001355E+02	4.996615E-03
10	1.581277E+06	1.257488E+03	2.001355E+02	4.996615E-03

Mode Shapes:

The first three modes are not shown (rigid body translations). Modes 4 through 10 are shown for the selected results. Modes 5 and 6 are identical for all element formulations. Modes 9 and 10 offer three different sets of results, depending on element formulation; (1) for selectively reduced 2x2 IPI thick shell (elform=2), there is a distinct difference in natural frequencies (Figure 13.2), (2) for assumed strain 2x2 IPI thick shell (elform=3), there is a very slight difference (Figure 13.3), and (3) for assumed strain RI thick shell (elform=5), the results are identical (Figure 13.4).



Figure 13.2 - Mode shapes for S/R 2x2 IPI thick shell (elform=2).



Figure 13.3 - Mode shapes for assumed strain 2x2 IPI thick shell (elform=3).



Figure 13.4 - Mode shapes for assumed strain RI thick shell (elform=5).

Input deck:

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Simp	ly Suppo	orted Squa	re Plate:	Out-of-Pla	ane Vibrati	on (thick	shell mesh)	
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Ş#	neig	center	lilag	litend	rilag	rhtend	eigmth	shiscl
	10	0.0	0	0.0	0	0.0	0	0.0
*CON	TROL_IMP	LICIT_GEN	ERAL				-	
\$# :	imtlag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.0						
*CON	TROL_SHE	LL						
\$# 1	wrpang	esort	irnxx	istupd	theory	bwc	miter	proj
20	.00000	0	0	0	2	2	1	
\$# ro	otascl	intgrd	lamsht	cstyp6	tshell	nfail1	nfail4	
	0.0	1						
*CON	FROL_TER	MINATION						
\$# 0	endtim	endcyc	dtmin	endeng	endmas			
1.0	000000	0	0.0	0.0	0.0			
*DAT	ABASE_BI	NARY_D3PL	TO					
\$# d1	t/cycl	lcdt/nr	beam	npltc	psetid			
1.0	000000							
*ELEI	MENT_TSH	IELL						
\$#	eid	pid	nl n	.2 n3	n4	n5	n6 n7	n8
	1	1	1 3	7 41	5	2	38 42	6
	192	1	283 31	9 323	287	284	320 324	288
*NODI	E							
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	324	T0.00000	000 T0	.000000000	1.000	00000		
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*BOUI \$#nio	324 NDARY_SP d/nsid 1	C_SET cid	dofx 0	.00000000 dofy 0	l.000 dofz 1	dofrx	dofry	dofrz
*BOUI \$#nic	324 NDARY_SF d/nsid 1 T	10.00000 PC_SET cid 0	dofx 0	.00000000 dofy 0	1.000 dofz 1	dofrx	dofry	dofrz
*BOUI \$#nic *PAR	NDARY_SF d/nsid 1 T	10.00000 PC_SET cid 0	dofx 0	.00000000 dofy 0	1.000 dofz 1	dofrx	dofry	dofrz
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*BOUI \$#nic *PAR \$# t: mate: \$#	NDARY_SF d/nsid 1 f itle rial typ pid 1	D.00000 C_SET cid 0 pe # 1 (E secid	dofx 0 lastic) mid 1	dofy eosid	l.000 dofz l hgid	dofrx grav	dofry adpopt	dofrz tmid
*BOUI \$#nic *PAR \$# t: mate: \$# \$#	NDARY_SP d/nsid 1 r itle rial typ pid 1 rION TSP	D.00000 C_SET cid 0 pe # 1 (E secid 1	dofx 0 lastic) 1	dofy 0 eosid	l.000 dofz 1 hgid 1	dofrx dofrx grav	dofry adpopt	dofrz tmid
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*BOUI \$#nic *PAR \$# t: mate: \$# *SEC \$#	NDARY_SF d/nsid I T itle rial typ pid 1 TION_TSH secid	DE H 1 (E secid 1 IELL elform	dofx dofx 0 lastic) mid 1 shrf	dofy o eosid 0 nip	l.000 dofz l hgid l propt	dofrx grav qr/irid	dofry adpopt icomp	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mate: \$# *SEC' \$# \$#	NDARY_SF d/nsid I r itle rial typ pid 1 FION_TSF secid 1	De # 1 (E secid 1 ELL elform 2	dofx dofx 0 lastic) mid 1 shrf 0.0	dofy eosid 0 nip 5	l.000 dofz l hgid l propt 0	dofrx grav qr/irid 0.0	dofry adpopt icomp	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mate: \$# *SEC' \$# \$# \$	NDARY_SF d/nsid I r itle rial typ pid 1 FION_TSF secid 1 1	De # 1 (E secid 1 IELL elform 2 3	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0	dofy eosid 0 nip 5	l.000 dofz l hgid l propt 0 5 0	dofrx dofrx grav qr/irid 0.0	dofry adpopt icomp	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mate: \$# *SEC' \$# \$ \$	NDARY_SF d/nsid 1 r itle pid 1 rION_TSF secid 1 1 r	De # 1 (E secid 1 IELL elform 2 3	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0	dofy eosid 0 nip 5	1.000 dofz 1 hgid 1 propt 0 5 0	dofrx dofrx grav qr/irid 0.0 0.0	dofry adpopt icomp	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mate: \$# *SEC' \$# \$ \$ \$ *MAT_ **	NDARY_SF d/nsid 1 r itle rial typ pid 1 TION_TSH secid 1 1 _ELASTIC 	De # 1 (E secid 1 IELL elform 2 3 5	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0	eosid ofy eosid 0 nip 5	1.000 dofz 1 hgid 1 propt 0 5 0 0	dofrx dofrx grav qr/irid 0.0 0.0 0.0	dofry adpopt icomp	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mate: \$# *SEC' \$# \$ \$ *MAT_ \$#	NDARY_SF d/nsid itle rial typ pid 1 TION_TSH secid 1 _ ELASTIC mid	De # 1 (E secid 1 IELL elform 3 5 r ro	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0 0.0	dofy eosid 0 nip 5	1.000 dofz 1 hgid 1 propt 0 5 0 0 5 0	dofrx dofrx grav qr/irid 0.0 0.0 0.0 0.0	dofry adpopt icomp not used	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mate: \$# *SEC' \$# \$ \$ \$ *MAT_ \$# *UOU	NDARY_SF d/nsid 1 r itle rial typ pid 1 TION_TSH secid 1 1 _ELASTIC mid	De # 1 (E secid 0 De # 1 (E secid 1 IELL elform 2 3 5 5 7 70 8000.0002	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0 0.0 0.0	.00000000 dofy eosid 0 nip 5	1.000 dofz 1 hgid 1 propt 0 5 0 0 5 0 0	dofrx dofrx grav qr/irid 0.0 0.0 0.0 0.0	dofry adpopt icomp not used 0.0	dofrz tmid tshear
*BOUI \$#nic *PAR' \$# t: mater \$# *SEC' \$# \$ \$ *MAT \$# *HOUI	NDARY_SF d/nsid itle rial typ pid 1 TION_TSH secid 1 _ ELASTIC mid 1 RGLASS	10.00000 C_SET cid 0 0 0 0 0 0 0 0 0 0 0 0 0	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0 0.0 0.0 0.0	dofy eosid 0 nip 5 0.300000	1.000 dofz 1 hgid 1 propt 0 5 0 0 5 0 0 5 0 0	dofrx dofrx grav qr/irid 0.0 0.0 0.0 0.0	dofry adpopt icomp not used 0.0	dofrz tmid tshear
*BOUI \$#nic *PAR \$# t: mate: \$# *SEC \$# \$ \$ *MAT \$# *HOUI \$#	NDARY_SF d/nsid I r itle rial typ pid 1 TION_TSH secid 1 1 _ELASTIC mid 1 RGLASS hgid	10.00000 C_SET cid 0 0 0 0 0 0 0 0 0 0 0 0 0	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0 0.0 0.0 0.0	dofy eosid 0 nip 5 0.300000 ibq	1.000 dofz 1 hgid 1 propt 0 5 0 0 5 0 0 5 0 0 5 0 0 0 0 0 0 0	dofrx dofrx grav qr/irid 0.0 0.0 0.0 0.0 0.0 0.0 0.0	dofry adpopt icomp not used 0.0 qb	dofrz tmid tshear
*BOUI \$#nic *PAR \$# t: mate: \$# *SEC \$# \$ \$ *MAT \$# *HOUI \$#	NDARY_SF d/nsid 1 r itle rial typ pid 1 TION_TSH secid 1 1 _ELASTIC mid 1 RGLASS hgid 1	10.00000 C_SET cid 0 0 0 0 0 0 0 0 0 0 0 0 0	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0	dofy eosid 0 nip 5 0.300000 ibq 0	1.000 dofz 1 hgid 1 propt 0 5 0 0 5 0 0 0 1 0.0	dofrx dofrx grav qr/irid 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0	dofry adpopt icomp not used 0.0 qb 0.0	dofrz tmid tshear qw 0.0
*BOUI \$#nic *PAR \$# t: mate: \$# *SEC \$# \$ \$ *MAT \$# *HOUI \$# *SET_ **	NDARY_SF d/nsid 1 r itle rial typ pid 1 FION_TSF secid 1 1 _ELASTIC mid 1 RGLASS hgid 1	10.00000 C_SET cid 0 0 0 0 0 0 0 0 0 0 0 0 0	dofx dofx 0 lastic) mid 1 shrf 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0	.00000000 dofy 0 eosid 0 nip 5 9 0.300000 ibq 0	1.000 dofz 1 hgid 1 propt 0 5 0 0 5 0 0 0 0 0 1 0.0	dofrx dofrx grav qr/irid 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0	dofry adpopt icomp not used 0.0 qb 0.0	dofrz tmid tshear qw 0.0
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Notes:

1. The assumed strain 2x2 IPI thick shell (elform=3) and the assumed strain RI thick shell (elform=5) are distortion sensitive and should not be used in situations where the elements are badly shaped.

- 2. With the one point reduced integration (elform=1) and the selectively reduced 2x2 IPI thick shell (elform=2), a single element through the thickness will capture bending response, but with the assumed strain 2x2 IPI thick shell (elform=3) and the assumed strain RI thick shell (elform=5), two elements are recommended to avoid excessive softness.
- 3. Only the selectively reduced 2x2 IPI thick shell (elform=2), the assumed strain 2x2 IPI thick shell (elform=3), and the assumed strain RI thick shell (elform=5) are available for implicit applications. If one point reduced integration (elform=1) is specified in an implicit analysis, it is internally switched to selectively reduced 2x2 IPI thick shell (elform=2).

14. Simply Supported Square Plate: Transient Forced Vibration (solid mesh)

Keywords:

*CONTROL_IMPLICIT_AUTO *CONTROL_IMPLICIT_DYNAMICS *CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLVER *CONTROL_IMPLICIT_SOLUTION

Description:

A plate is subjected to a suddenly applied pressure on its top. A transient analysis is performed in order to obtain the response of the plate. Damping is present. On the lower surface, outer boundary nodes, $U_z = 0$.

The finite element model is shown in Figure 14.1.

Simply Supported Square Plate: Transient Forced Vibration (solid mesh)



Figure 14.1 – Finite element model with applied pressure on upper surface and simply supported boundary conditions on lower surface, outer nodes.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Pressure Damping	Linear	Linear	-	Implicit	1 - Linear

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

Rectangular dimensions of square plate: 10.0 m x 10.0 m x 1.00 m.

Material Data:

Mass Density	$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
Young's Modulus	$E = 2.00 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3
Damping ratio	$\zeta = 2\%$

Load:

Pressure	$P = 1.0 \times 10^{6}$	Pa

Element Types:

Constant stress solid (elform=1) Fully integrated S/R solid (elform=2) Fully integrated S/R solid - for poor aspect ratio (eff) - (elform=-1) Fully integrated S/R solid - for poor aspect ratio (acc) - (elform=-2) Fully integrated quadratic 8 node element with nodal rotations (elform=3)

Material Models:

*MAT_001 or *MAT_ELASTIC

Damping:

The damping factor d is easily found from the natural frequency of the system:

$$d_i = \zeta 2\omega_i$$

As the excited mode is the first, corresponding to $f_1 = 45.897 \ Hz$ ($\omega_1 = 288.380 \ rad / s$) (that from NAFEMS Benchmark Test FV52), we choose the damping factor relative to the first frequency:

$$d_1 = 11.535 Hz$$

Results Comparison:

LS-DYNA X-direction bending stress, σ_{xx} , at (Node 161) on bottom surface plus its Zdisplacement, U_z , are compared with *NAFEMS Selected Benchmarks for Forced Vibration*, Test 21T.

Reference Condition - Center (Node 161)	Peak Bending Stress σ_{xx} (Pa)	Peak $U_{z}(m)$	Steady-State U _z (m)
NAFEMS Benchmark Test 21T	6.211×10 ⁷	-4.524×10 ⁻³	-2.333×10 ⁻³
Constant stress solid (elform=1)	4.638×10 ⁷	-5.438×10 ⁻³	-2.778×10^{-3}
Fully integrated S/R solid (elform=2)	3.732×10 ⁷	-3.925×10^{-3}	-2.019×10 ⁻³
Fully integrated S/R solid (elform=-1)	4.611×10 ⁷	-4.435×10 ⁻³	-2.242×10^{-3}
Fully integrated S/R solid (elform=-2)	4.221×10 ⁷	-4.365×10 ⁻³	-2.203×10 ⁻³
Fully integrated quadratic element with nodal rotations (elform=3)	$x.xxx \times 10^7$	$-x.xxx \times 10^{-3}$	$-x.xxx \times 10^{-3}$

The constant stress solid (elform=1) result of 4.638×10^7 Pa is an element centroid value.

These nodal displacement results were generated by *DATABASE_NODOUT keyword while the axial stress (nodal) results were generated by *DATABASE_ELOUT (*elout* file) and *DATABASE_EXTENT_BINARY (*eloutdet* file provides detailed element output at integration points and connectivity nodes) keyword entries.

You can set intout=stress or intout=all (*DATABASE_EXTENT_BINARY) and have stresses output for all the integration points to a file called *eloutdet*

(*DATABASE_ELOUT governs the output interval and *DATABASE_HISTORY_ SOLID governs which elements are output). Setting nodout=stress or nodout=all in *DATABASE_EXTENT_BINARY will write the extrapolated nodal stresses to *eloutdet*.

LS-DYNA stress and strain output corresponds to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different post-processors).

For this coarse mesh, the one-point quadrature (low order) constant stress solid (elform=1) element formulation (the LS-DYNA default) provides a less stiff, stress and displacement comparison. Refinement of the mesh should provide a better comparison.

The higher order, fully integrated selectively reduced solid (elform=2) provides a comparatively stiff result, both in stress and displacement, probably due to the coarse mesh.

The aspect ratio of these elements is 3.75 (ratio of side to depth length). Available options are the higher order, fully integrated S/R solid (both the so-called efficient and the so-called accurate formulation choices) intended to address poor aspect ratios (elform=-1 and -2, respectively). These formulations provide a good comparison of displacements (peak and steady-state) for this coarse mesh. Unfortunately, the stress comparison is not very good; it is not understood why at this time.

The fully integrated S/R solid (the so-called efficient formulation choice) intended to address poor aspect ratios (elform=-1), can provide a slightly less stiff solution than the so-called accurate formulation choice (elform=2). This formulation (elform=-1) involves a slight modification of the Jacobian matrix which can lead to a stiffness reduction for certain modes, in particular the out-of-plane hourglass mode (according to Borrvall [2009]).

The higher order, fully integrated quadratic 8 node element with nodal rotations (elform=3) formulation provides a (????) results. Waiting for LSTC LS-DYNA code fix to remark on this.

For the fully integrated S/R solid accurate formulation (elform=-2), the contour plot of the X-direction bending stress (Figure 14.2) and the Z-displacement (Figure 14.3) were obtained from the *d3plot* file at peak displacement time which were generated by the *DATABASE_BINARY_D3PLOT keyword.



Figure 14.2 – Contour plot of the X-stress (elform=-2) at peak displacement time.



Figure 14.3 - Contour plot of Z-displacement (elform=-2) at peak displacement time.
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Notes:

1. One should remember that the constant stress solid (elform=1), the fully integrated S/R solid (elform=2), and the fully integrated S/R solid (both the so-called efficient and the so-called accurate formulation choices) intended to address poor aspect ratios (elform=-1 and -2, respectively) were originally developed for performing highly nonlinear, dynamic deformation simulations.

15. Simply Supported Square Plate: Transient Forced Vibration (thick shell mesh)

Keywords:

*CONTROL_IMPLICIT_AUTO *CONTROL_IMPLICIT_DYNAMICS *CONTROL_IMPLICIT_GENERAL *CONTROL_IMPLICIT_SOLVER *CONTROL_IMPLICIT_SOLUTION

Description:

A plate is subjected to a suddenly applied pressure on its top. A transient analysis is performed in order to obtain the response of the plate. Damping is present. On the lower surface, outer boundary nodes, $U_z = 0$.

The finite element model is shown in Figure 15.1.

Simply Supported Square Plate: Transient Forced Vibration (thick shell mesh)



Figure 15.1 – Finite element model with applied pressure on upper surface and simply supported boundary conditions on lower surface, outer nodes.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Pressure Damping	Linear	Linear	-	Implicit	1 - Linear

Units:

kg, m, s, N, Pa, N-m (kilogram, meter, second, Newton, Pascal, Newton-meter)

Dimensional Data:

Rectangular dimensions of square plate: 10.0 m x 10.0 m x 1.00 m.

Material Data:

$\rho = 8.00 \times 10^3 \ kg \ / \ m^3$
$E = 2.00 \times 10^{11} Pa$
v = 0.3
$\zeta = 2\%$

Load:

Pressure	$P = 1.0 \times 10^{6}$	Pa

Element Types:

S/R 2x2 IPI thick shell (elform=2) Assumed strain 2x2 IPI thick shell (elform=3) Assumed strain RI thick shell (elform=5)

Material Models:

*MAT_001 or *MAT_ELASTIC

Damping:

The damping factor d is easily found from the natural frequency of the system:

$$d_i = \zeta 2\omega_i$$

As the excited mode is the first, corresponding to $f_1 = 45.897 \ Hz$ ($\omega_1 = 288.380 \ rad / s$) (that from NAFEMS Benchmark Test FV52), we choose the damping factor relative to the first frequency:

$$d_1 = 11.535 Hz$$

Results Comparison:

LS-DYNA X-direction bending stress, σ_{xx} , at (Node 161) on bottom surface plus its Zdisplacement, U_z , are compared with *NAFEMS Selected Benchmarks for Forced Vibration*, Test 21T.

Reference Condition - Center (Node 161)	Peak Bending Stress σ_{xx} (Pa)	Peak $U_{z}(m)$	Steady-State $U_{z}(m)$
NAFEMS Benchmark Test 21T	6.211×10 ⁷	-4.524×10 ⁻³	-2.333×10 ⁻³
S/R 2x2 IPI thick shell (elform=2)	6.398×10 ⁷	-4.937×10^{-3}	-2.537×10^{-3}
Assumed strain 2x2 IPI thick shell (elform=3)	6.350×10 ⁷	-5.090×10 ⁻³	-2.616×10 ⁻³
Assumed strain RI thick shell (elform=5)	6.319×10 ⁷	-5.022×10^{-3}	-2.557×10^{-3}

These nodal displacement results were generated by *DATABASE_NODOUT keyword while the axial stress (nodal) results were generated by *DATABASE_ELOUT (*elout* file) and *DATABASE_EXTENT_BINARY (*eloutdet* file provides detailed element output at integration points and connectivity nodes) keyword entries.

You can set intout=stress or intout=all (*DATABASE_EXTENT_BINARY) and have stresses output for all the integration points to a file called *eloutdet* (*DATABASE_ELOUT governs the output interval and *DATABASE_HISTORY_ TSHELL governs which elements are output). Setting nodout=stress or nodout=all in *DATABASE_EXTENT_BINARY will write the extrapolated nodal stresses to *eloutdet*.

LS-DYNA stress and strain output corresponds to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different post-processors).

Lobatto integration (intgrd=1 - *CONTROL_SHELL) was employed since it has an advantage in that the inner and outer integration points are on the shell surfaces. Gauss integration is the default through thickness integration rule (the default number of through thickness integration points is nip=2 - *SECTION_TSHELL) in LS-DYNA, where 1-10 integration points may be specified, whereas, with Lobatto integration, 3-10 integration points may be specified (for 2 point integration, the Lobatto rule is very inaccurate).

The selectively reduced 2x2 IPI thick shell (elform=2), the assumed strain 2x2 IPI thick shell (elform=3), and the assumed strain RI thick shell (elform=5) all provide similar results for this transient forced vibration example, though slightly less stiff in comparison, both in stress and displacement.

Remember that (a) only the higher order, selectively reduced 2x2 IPI thick shell (elform=2) provides a reasonable stress comparison for a single element through the thickness, although with a comparatively stiff result, and (b) the higher order, assumed strain 2x2 IPI thick shell (elform=3) and assumed strain RI thick shell (elform=5) formulations do provide acceptable results with at least two elements through the thickness (recommended) to capture the bending response.

For this transient forced vibration example, with an element aspect ratio of 3.75, it is seen that the thick shell formulations, on the whole, compare better than the solid element formulation results. The exception to this would be the displacement comparison provided by the higher order, fully integrated S/R solid (both so-called efficient and accurate formulation choices) intended to address poor aspect ratios (elform=-1 and -2, respectively).

For the selectively reduced 2x2 IPI thick shell (elform=2), the contour plot of the X-direction bending stress (Figure 15.2) and the Z-displacement (Figure 15.3) were obtained from the *d3plot* file at peak displacement time which were generated by the *DATABASE_BINARY_D3PLOT keyword.



Figure 15.2 – Contour plot of the X-stress (elform=2) at peak displacement time.



Figure 15.3 - Contour plot of Z-displacement (elform=2) at peak displacement time.

Input deck:

*KEYWORD *TITLE Simply Supported Square Plate: Transient Forced Vibration (thick shell mesh) *CONTROL_IMPLICIT_AUTO itewin dtmin dtmax dtexp kfail kcycle \$# iauto iteopt 0 11 5 0.0 0.0 *CONTROL_IMPLICIT_DYNAMICS \$# imass gamma beta 1 0.500000 0.250000 *CONTROL_IMPLICIT_GENERAL \$# imflag dt0 imform nsbs igs cnstn form zero_v 1 0.000100 1 2 2 *CONTROL_IMPLICIT_SOLVER \$# lsolvr lprint negev order drcm drcprm autospc autotol 2 2 4 0 1 0.0 0.0 1 \$# lcpack 2 *CONTROL_IMPLICIT_SOLUTION \$# nsolvr ilimit maxref dctol ectol rctol lstol abstol 0.0100 1.00e+10 0.900000 1.000000 2 11 15 0.0010 istif nlprint \$# dnorm diverg 2 1 1 2 \$# arcctl arcdir arclen arcmth arcdmp 0 1 *CONTROL_SHELL 0.0 1 2 \$# wrpang esort irnxx istupd bwc miter theorv proj 20.00000 0 0 0 2 2 1 intgrd 1 \$# rotascl lamsht tshell nfail1 nfail4 cstyp6 0.0 *CONTROL_TERMINATION \$# endtim endcyc endeng endmas dtmin 0 0.020000 0.0 0.0 0.0 *DATABASE_ELOUT \$# dt binary lcur ioopt 1.0000e-06 1 *DATABASE_NODFOR \$# dt binary lcur ioopt 1.0000e-06 1 *DATABASE_NODOUT \$# dt binary 1.0000e-06 1 lcur ioopt 1.0000e-06 *DATABASE_BINARY_D3PLOT \$# dt/cycl 0.001000 *DATABASE EXTENT BINARY \$# neiph neips maxint strflg sigflg epsflg rtflg engflg stssz n3thdt \$# cmpflg ieverp beamip dcomp shge ialemat \$# nintsld pkp_sen msscl sclp hydro therm intout nodout 8 1.0 stress stress *DATABASE_HISTORY_TSHELL \$# id1 id2 id3 id4 id5 id6 id7 id8 28 29 36 37 *DATABASE_NODAL_FORCE_GROUP \$# nsid cid 164 *DATABASE_HISTORY_NODE \$# nidl nid2 nid3 nid4 nid5 nid6 nid7 nid8 161 164 *SET NODE LIST da1 sid \$# da2 da3 da4 solver 164 0.0 0.0 0.0 0.0 nid2 nid3 nid4 nid5 nid6 nid7 nid8 \$# nid1 164 *DAMPING_GLOBAL \$# lcid valdmp stx sty stz srx sry srz 0.0 0 11.53500 0.0 0.0 0.0 0.0 0.0

*DEFI	INE_CUR	VE								
\$#	lcid	sdir	sfa	a	sfo	offa	offo		dattyp	
	1	0	0.	0	0.0	0.0	0.0			
\$#		al			01					
		0.0	1.	000	0000e+06					
	0	.10000000	1.	000	0000e+06					
*ELEN	MENT_TSI	HELL								
\$#	eid	pid	nl	n2	n3	n4	n5	n6	n7	n8
	1	1	1	37	41	5	2	38	42	6
	192	1	283	319	323	287	284	320	324	288
*NODE	C									
\$#	nid		x		У		Z	tc	rc	
	1		0.0		0.0		0.0	3		
*0010	324	10.0000	0000	10.0	00000000	1.000	00000			
° BOUR	VDARI_SI	PC_SEI	dof		dofu	dofr	dofry		dofry	dofra
\$#111C	1/11510 1	CIU	001.	<u>∧</u> ∩	dory	1	UULLX		dolly	uoliz
*PART		0		0	0	1				
\$# τ1	LTIE									
mater	riai tyj	pe # 1 (H	slastic)	a		h a i d			due europe	بر ان میر م
Ş₩	1	secia	III LO	1	eosia	ngra	grav	a	αρορι	Unita
*	⊥ וסיד זא∩די	עד דיקט נוסיד		L						
4#	secid	alform	chr	f	nin	propt	ar/irid		icomp	tchoor
ŶΠ	1	2	0	0	5	01010	Q1 / 11 10		rcomp	concar
Ċ	1	2	2 O.	0	5	0	0.0	0		
Ś	1		5 0	0	5	0	0.	0		
~ *МАТ	FLASTI	r .		••	5	0		0		
\$#	mid	ro		e	pr	da	db	not	used	
	1	8000.0002	2.0000e+1	1 (0.300000	0.0	0.0		0.0	
*LOAD		NT								
\$#	lcid	sf	a	t	nl	n2	n3		n4	
	1	1.000000	0.	0	4	40	44		8	
	1	1.000000	0.	0	284	320	324		288	
*SET_	NODE_L	IST								
\$# _	sid	da1	da	2	da3	da4	solver			
	1	0.0	0.	0	0.0	0.0				
\$#	nid1	nid2	nid	3	nid4	nid5	nid6		nid7	nid8
	1	37	7	3	109	145	181		217	253
	289	293	29	7	301	305	309		313	317
	321	285	24	9	213	177	141		105	69
	33	29	2	5	21	17	13		9	5

*END

Notes:

16. Transient Response of a Cylindrical Disk Impacting a Deformable Surface

Keywords:

```
*CONTROL_IMPLICIT_DYNAMICS
*CONTROL_IMPLICIT_AUTO
*CONTROL_IMPLICIT_GENERAL
*CONTROL_IMPLICIT_SOLVER
*CONTROL_IMPLICIT_SOLUTION
*CONTACT_2D_AUTOMATIC_NODE_TO_SURFACE
```

Description:

A rigid cylindrical disk of given mass (m) is released from 1.0 inch height (h) and, accelerated by gravity (g), hits a deformable surface of given stiffness (k). Plot the velocity, displacement, and kinetic energy of the disk, plus identify the time of impact. The maximum displacement of the cylindrical disk is also to be determined.

This simulation (Figure 16.1) is 2D plane strain (a choice of the available LS-DYNA options to address the impact - with the objects being rigid, this can be seen to negate the need for any planar type definition). The cylindrical disk is modeled with shell elements, x-y plane, which do not need additional constraints to ensure in-plane behavior. The deformable surface is modeled with rigid beam elements, x-y plane, to address contact, and a 1D translational spring element of a finite length (l), x-y plane, to address the deformation.

As a geometric convenience, LS-DYNA employs the *SECTION_SHELL and *ELEMENT_SHELL entries to describe 2D plane stress, plane strain, and axisymmetric solids, and the *SECTION_BEAM and *ELEMENT_BEAM entries to describe 2D axisymmetric shells, and 2D plane strain beam elements.

A suitable contact algorithm for this problem is the *CONTACT_2D_ AUTOMATIC_NODE_TO_SURFACE. For this algorithm, the contact stiffness is activated when a node nears a segment at some given tolerance. The stiffness is increased as the node moves closer with the full stiffness being used when the nodal point finally makes contact. Understanding LS-DYNA contact considerations adds to the focus of this example. A plot of the contact force history is sought. Transient Response of a Cylindrical Disk Impacting a Flexible Surface





Figure 16.1 – Finite element model with selected parts, elements, and nodes identified.

Analysis Summary:

ž_x

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
2D	Dynamic	Gravity	Linear	Linear	2D	Implicit	2-Nonlinear w/BFGS

Units:

 $lbf-s^2/in$, in, s, lbf, psi, lbf-in (blob, inch, second, pound force, pound force/inch², pound force-inch)

Dimensional Data:

 $h = 1.0 \times 10^{\circ}$ in, $l = 1.0 \times 10^{\circ}$ in

Material Data:

Mass Density	$\rho = 3.995281 \times 10^0 \ lbf - s^2 / in^4$
Nodal Mass	$m = 0.50 \ lbf - s^2 \ / \ in$
Spring Stiffness	$k = 1.97392 \times 10^3 \ lb / in$

Load:

```
Body Force g = 0.0 \text{ in } / s^2 varied linearly to 3.86 \times 10^2 \text{ in } / s^2, then held constant

g = 0.0 \text{ in } / s^2, t = 0.0 \text{ s}

g = 3.86 \times 10^2 \text{ in } / s^2, t = 1.0 \times 10^{-3} \text{ s}

g = 3.86 \times 10^2 \text{ in } / s^2, t = 1.0 \times 10^0 \text{ s}
```

Element Types:

2D plane strain shell element (xy plane) - *SECTION_BEAM entry (elform=7)

Plane strain (x-y plane) - *SECTION_SHELL entry - (elform=13)

Translational spring - (SECTION_DISCRETE (dro=0)

Material Models:

*MAT_001 or *MAT_ELASTIC

*MAT_020 or *MAT_RIGID

Results Comparison:

LS-DYNA results for velocity V_Y and displacement U_Y of the cylindrical disk at impact (plus the time) and maximum displacement of the cylindrical disk U_{Ymax} , are compared with W.T. Thomson's studies in *Vibration Theory and Applications*, 1965 (pg. 110).

	Impact Time (s)	Velocity V _Y (in/s)	Displacement U_{Y} (in)	Max. Disp. U _{Ymax} (in)
Thomson [1965]	0.07198	-27.7900	-1.0000	-1.5506
Cylindrical Disk - Part 1	0.07240	-27.7728	-0.9978	-1.5524

LS-DYNA results are reported at the closet time point for the displacement value designated as full stiffness contact (i.e. $U_y = -1.0000$ in).

The nodal results were generated by *DATABASE_NODOUT keyword, the kinetic energy by *DATABASE_MATSUM keyword, and the contact force by *DATABASE_RCFORC keyword.

Figure 16.2 provides the velocity V_Y history, Figure 16.3 the vertical displacement U_Y history, and Figure 16.4 the kinetic energy history, all of the cylindrical disk.

Figure 16.5 gives the contact force between the cylindrical disk (slave) and the flexible surface (master).



Figure 16.2 – Velocity V_y of the cylindrical disk.



Figure 16.3 – Vertical displacement U_y of the cylindrical disk.



Figure 16.4 – Kinetic energy of the cylindrical disk.



Figure 16.5 – Contact force between cylindrical disk (slave) and flexible surface (master).

Input deck:

*KE	YWORD							
*TI	TLE							
Tra	nsient R	esponse of	a Cylindr	ical Disk	Impacting	a Flexible	Surface	
*CO	NTROL_IM	PLICIT_DYN	AMICS					
\$#	imass	gamma	beta					
	1	0.500000	0.250000					
*CO	NTROL_IM	PLICIT_AUT	0					
\$#	iauto	iteopt	itewin	dtmin	dtmax			
	1	11	5	1.00e-06	1.00e-04			
*CO	NTROL_IM	PLICIT_GEN	ERAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.000100	0	0	0			
*C0	NTROL_IM	PLICIT_SOL	VER					
\$#	lsolvr	prntflg	negeig	order	drcm	drcprm	autospc	aspctl
	4	2	2	0	1	0	1	0
\$#	lcpack							
	2							
*CO	NTROL_IM	PLICIT_SOL	UTION		_	_		
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	2	11	15	0.0010	0.0100	1.00e+10	0.900000	1.00e-10
Ş#	dnorm	diverg	istif	nlprint	nlnorm	d3itctl	cpchk	
	2			2				
Ş#	arcctl	arcdir	arclen	arcmth	arcdmp			
	0	1	0.0	1	2			
*C0	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
	0.1100							
*DA	TABASE_G	LSTAT						
\$#	dt	binary						

1.0000e-04 1 *DATABASE_MATSUM *DATABASE_FMILE_1 \$# dt binary 1.0000e-04 1 *DATABASE_RCFORC \$# dt binary 1.0000e-04 1 *DATABASE_NODOUT \$# dt binary 1.0000e-04 1 *DATABASE_BINARY_D3PLOT \$# dt/cycl lcdt/nr beam npltc psetid 1.0000e-03 *DATABASE_HISTORY_NODE \$# nid1 nid2 nid3 nid4 ni5 nid6 nid7 nid8 166 1001 *PART \$# title rigid cylindrical disk \$# pid secid mid 1 1 1 *SECTION_SHELL eosid hgid grav adpopt tmid \$# secid elform shrf
 1 13 0
\$# t1 t2 t3
 1.0000 1.0000 1.0000
*MAT_RIGID nip propt qr/irid icomp setyp -4 t4 nloc marea 1.0000 *MAT_RIGID \$ mid ro e n 0.0 couple m 0.0 0.0 alias pr 1 3.9952831 3.00e+07 0.3000 0.0 \$ cmo conl con2 1.0000 6.0 \$lco_or_al a2 0.0 0.0 *PART 7.0 a3 0.0 v1 v2 v3 0.0 0.0 0.0 discrete spring - flexible surface \$# pid secid mid eosid
 2 2 2
*SECTION_DISCRETE hgid grav adpopt tmid dro 0 \$# secid dro kd v0 cl fd 2 0 \$# cdl tdl 0.0 0.0 0.0 0 0.0 0 *MAT_SPRING_ELASTIC \$# mid k .- k 2 1973.9200 *PART rigid plane beam (wall) \$# pid secid mid eosid hgid grav adpopt tmid
3 3 3
*SECTION_BEAM cst nsm scoor 0.0000 *MAT_RIGID \$ mid ro e pr 3 1.00e-07 3.00e+07 0.3000 con1 con2 n couple m 0.0 0.0 0.0 alias 0.0 \$ Cmo con1 con2 \$ cmo con1 con2 1.0000 6.0 7.0 \$lco_or_a1 a2 a3 0.0 0.0 0.0 *ELEMENT_BEAM v1 v2 v3 0.0 0.0 0.0 ELEMENT_BEAM 1002 3 1003 1001 1003 3 1001 1004 *ELEMENT_DISCRETE n2 \$ eid pid 1001 2 s pf offset 1.0 0 0.0 nl n2 1001 1002 vid 2 0 *ELEMENT_SHELL \$# eid pid n1 n2 1 1 1 12 n3 n4 n5 n6 n7 n8 13 2 0 0 0 0 0

	500	1	512 52	1 431	430	0	0	0	0
*NC	DE								
\$#	nid 1	-0.14142	x 136 0	у .05857864		z 0.0	tc	rc	
	521	0.16180	338 0	.31755710		0.0			
	1001		0.0	-1.050		0.0	6	7	
	1002		0.0	-2.050		0.0	7	7	
	1003	-	0.5	-1.050		0.0	6	7	
	1004		0.5	-1.050		0.0	6	7	
Ś	1001		0.0	-1 0		0.0	6	7	
ŝ	1002		0.0	-2.0		0.0	7	7	
ŝ	1003		-0.5	-1.0		0.0	6	7	
ŝ	1004		0.5	-1.0		0.0	6	7	
*CC	NTACT 2D	AUTOMATIC	NODE TO S	URFACE		0.0	Ū		
\$#	ssid	msid	sfact	freq	fs	fd	de		membs
Υ II	-2	-1	0.10	0	0	0	0		0
\$#	tbirth	tdeath	505	som	nds	ndm	cof		init
Υ II	0	0	0	0	0	0	0		0
\$#	vc	vdc	ipf	slide	istiff	tiedgap	Ū		Ū
<u>+</u> +					2				
2°C	CONTACT_2D	_PENALIY	+ biw+ b	tdooth					
># α	ssia	IIISIO 1	LDIFLI	Ldealli					
с т	aut pag	+boto1	+boto)	tolia	202	toloff	fmagal		
с 4	ext_pas	liletai	LIIELAZ	LOT_IG	0 10	0 00010	IICSCI		Ulleway
ч 4 ОТ	ד די די די	CTT			0.10	0.00010			
		.51 do1	4-2	4-2	da 4				
фĦ	1	uai	uaz	uas	uat				
¢#	nid1	nid2	nida	nid4	nid5	nid6	nid7		nid8
ŶΠ	1003	1001	1004	mai	mus	mido	iiiu,		mao
*SF	T NODE LT	ST	1001						
S#	sid	dal	da2	da3	da4				
Υ II	2	aar	aan	aab					
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7		nid8
4.0	155	166	177						
*LC	AD BODY Y								
Ś	lcid	sf	lciddr	xc	VC	ZC	cid		
	1	1.0			1-				
*DE	FINE CURV	Е							
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp		
	1	0	1.0000	1.0000	0.0	0.0			
\$#		al		01					
		0.000		0.000					
		0.001		386.00					
		1.000		386.00					
*EN	ID								

Notes:

- 1. From the LS-DYNA User's Manual: Note that the 2D and 3D element types must not be mixed, and different types of 2D elements, i.e. plane strain, plane stress, and axisymmetric, must not be used together. The discrete (spring) 1D element can be used with either 2D or 3D elements.
- 2. Consider the two surfaces comprising a contact. It is necessary to designate one as a slave surface and the other as a master surface. Nodal points defining the slave surface are called slave nodes, and similarly, nodes defining the master surface are called master nodes. Each slave-master surface combination is referred to as a contact surface. If one surface is more finely zoned, it should be defined as the slave surface.

3. By default, the true thickness of 2D shell elements is taken into account for *CONTACT_2D_AUTOMATIC_SURFACE_TO_SURFACE and _AUTOMATIC_ NODE_TO_SURFACE contacts. The user can override the true thickness by using the sos and som parameters on the contact entry.

input example:

*CO	NTACT_2D_	AUTOMATIC_N	IODE_TO_SU	RFACE					
\$#	ssid	msid	sfact	freq	fs	fd		dc	membs
	-2	-1	0.10	0	0	0		0	0
\$#	tbirth	tdeath	SOS	som	nds	ndm		cof	init
	0	0	0	0	0	0		0	0
\$#	VC	vdc	ipf	slide	istiff 2	tiedgap			
\$									
*SE	T_NODE_LI	ST							
\$#	sid	dal	da2	da3	da4				
<i></i>		. 10				. 16		. 16	. 10
ŞĦ	n1d1 1002	n1d2	n1d3	nia4	nid5	nidb		nid/	nias
* 0 T	TUU3	TUUT	1004						
^SE	I_NODE_LI	51							
Ş#	sıd 2	dal	da2	da3	da4				
\$#	nid1	nid2	nid3	nid4	nid5	nid6		nid7	nid8
	155	166	177						
\$									
*NO	DE								
\$#	nid		x	У		Z	tc	rc	
	1001	0.	0	-1.050		0.0	6	7	
	1002	0.	0	-2.050		0.0	7	7	
	1003	-0.	5	-1.050		0.0	6	7	
	1004	0.	5	-1.050		0.0	6	7	

There is a stiffness control variable available, cof, which allows the full stiffness to be gradually applied as a node approaches a segment. The tolerance for the stiffness appears to be hardwired internally in the LS-DYNA software. cof offers only two options: on (cof=0 - LS-DYNA default) or off (cof=1); no tolerance adjusting.

Using cof=0 activates the contact stiffness as a node approaches a segment at some unknown value; the stiffness is increased as the node moves closer with the full stiffness being used when the nodal point finally makes contact. Using cof=1 does not turn on any contact stiffness until the nodal point makes full stiffness contact.

It is observed that the contact output force calculation (*DATABASE_RCFORC) is not made (delayed) until full stiffness contact is made for either cof option.

For cof=1, the contact force is calculated without a delay (due to full stiffness being applied without the gradual increase), but is nosier (oscillatory) than the other solution cof=0, at least for this example problem.

4. If the older penalty contact algorithms are used, *CONTACT_2D _PENALTY and _PENALTY_ FRICTION, the slave-master distinction is irrelevant. These contacts use the mid-surface of the 2D shell elements; thus, the shell thickness is not taken into account. This sometimes may make it necessary to modify various physical

coordinates to achieve reasonable results, e.g., the arrival time of a dropped rigid sphere onto a 2D shell plate of a moderate thickness.

input example:

*C(DNTACT_2D	_PENALTY							
\$#	ssid	msid	tbirth	tdeath					
	2	1							
\$#	ext_pas	thetal	theta2	tol_ig	pen	toloff	f	rcscl	oneway
					0.10	0.00010			
	155	166	177						
\$									
*SI	T_NODE_L	IST							
\$#	sid	dal	da2	da3	da4				
	1								
\$#	nid1	nid2	nid3	nid4	nid5	nid6		nid7	nid8
	1003	1001	1004						
*SI	ET_NODE_L	IST							
\$#	sid	dal	da2	da3	da4				
	2								
\$#	nidl	nid2	nid3	nid4	nid5	nid6		nid7	nid8
	155	166	177						
\$									
*N(DDE								
\$#	nid		х	У		Z	tc	rc	
	1001		0.0	-1.0		0.0	6	7	
	1002		0.0	-2.0		0.0	7	7	
	1003	-	0.5	-1.0		0.0	6	7	
	1004		0.5	-1.0		0.0	6	7	

There is an adjustable, stiffness control variable available, toloff, which allows the full stiffness to be gradually applied as a node approaches a segment.

From the LS-DYNA User's Manual: toloff - Tolerance for stiffness activation for implicit solution only. The contact stiffness is activated when a node approaches a segment at a distance equal to the segment length multiplied by toloff. The stiffness is increased as the node moves closer with the full stiffness being applied when the nodal point finally makes contact.

It is observed that the contact output force calculation (*DATABASE_RCFORC) is delayed until full stiffness contact is made.

17. Natural Frequency of a Linear Spring-Mass System

Keywords:

*CONTROL_TIMESTEP *ELEMENT_DISCRETE *ELEMENT_MASS *MAT_SPRING_ELASTIC

Description:

A mass (*m*) is attached to a linear spring, as shown in Figure 17.1. The mass is initially displaced d = -1.0 in from its equilibrium position and released. Determine the period of vibration τ .

The spring is modeled by one discrete element (*ELEMENT_DISCRETE) using a linear elastic spring material (*MAT_S01/*MAT_SPRING_ELASTIC). The lumped mass is modeled by an *ELEMENT_MASS entry.



Figure 17.1 – Sketch representing the model.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Initially Displace	Linear	Linear	-	Explicit	-

Units:

lbf-s²/in, in, s, lbf, psi, lbf-in (blob, inch, second, pound force, pound force/inch², pound force-inch)

Dimensional Data:

 $d = 1.0 \times 10^{\circ}$ in

Material Data:

Nodal Mass	$m = 2.588 \times 10^{-3} \ lbf - s^2 \ / \ in$
Spring Stiffness	$k = 5.0 \ lbf \ / in$

Element Types:

Translational spring - *SECTION_DISCRETE (dro=0)

Lumped mass (*ELEMENT_MASS entry)

Material Models:

*MAT_S01 or *MAT_SPRING_ELASTIC

Results Comparison:

LS-DYNA results for the period of vibration related to this linear spring-mass system are compared with S.P. Timoshenko and D.H. Young studies in *Vibration Problems in Engineering*, 1955 (pg. 1).

	Period of Vibration $ au$ (s)
Timoshenko and Young [1955]	0.14295
Linear spring-mass system	0.14295

This nodal displacement result and the computer period of vibration was generated by *DATABASE_NODOUT keyword (also see Figures 17.2 and 17.3).



Figure 17.2 – Node 1 displacement U_{y} .



Figure 17.3 – Node 1 displacement U_y (detailed time capture).

Input deck:

*KEYWORD *TITLE Natural Frequency of a Linear Spring-Mass System *CONTROL_TERMINATION \$# endtim endcyc dtmin endeng endmas 0.150000 0.0 0.0 0.0 0 *CONTROL_TIMESTEP tssfac \$# dtinit isdo tslimt dt2ms lctm erode ms1st 1.000e-04 1.000e-04 0 0.0 0.0 0 0 0 \$# dt2msf dt2mslc imscl 0.0 0 0 *DATABASE_NODOUT \$# dt binary 0.000100 *DATABASE_BINARY_D3PLOT \$# dt/cycl lcdt/nr beam npltc psetid 0.001000 *DATABASE_HISTORY_NODE nid1 nid2 nid3 nid4 ni5 nid6 nid7 nid8 \$# 1 *PART \$# title linear elastic spring pid hgid \$# secid mid eosid adpopt tmid grav 1 1 1 *SECTION_DISCRETE v0 \$# secid dro kd cl fd 1 0 0.0 0.0 0.0 0.0 cd1 \$# td1 0.0 0.0 *MAT_SPRING_ELASTIC \$# mid k 1 5.00 *ELEMENT_DISCRETE \$# eid pid n1 n2 vid pf offset s 1.00000 1.00000 1 1 1 2 0 0 *ELEMENT_MASS eid id pid \$# mass 2 1 0.00258800 3 2 0.00258800 *NODE \$# nid х z tc rc У 0.0 0.0 0.0 1 2 0.0 1.0000000 0.0 *BOUNDARY_SPC_NODE \$#nid/nsid cid dofx dofy dofz dofrx dofry dofrz 2 0 1 1 1 1 1 1 *END

Notes:

1. As an alternative, it is possible to model the linear spring with a *BEAM_ELEMENT (with the option discrete) and *MAT_066/*MAT_ LINEAR_ELASTIC_ DISCRETE_BEAM material behavior.

2. For simulations with linear stiffness, one could use the following implicit entries and perform a simple eigenvalue analysis:

*CO	NTROL_IM	IPLICIT_EIGE	INVALUE					
\$#	neig	center	lflag	lftend	rflag	rhtend	eigmth	shfscl
	3	11.000	0 -1	L.00e+29	0	1.00e+29	2	0.0
*CO	NTROL_IM	PLICIT_GENE	RAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	6	1.00e-04	2	1	2			

The period could be obtained directly from the *eigout* results file generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword as shown here:

18. Natural Frequency of a Nonlinear Spring-Mass System

Keywords:

*CONTROL_TIMESTEP *ELEMENT_DISCRETE *ELEMENT_MASS *MAT_SPRING_NONLINEAR_ELASTIC

Description:

A mass (*m*) is attached to a nonlinear spring: $k = k_0 + k_1 \delta^2$, as shown in Figure 18.1. The mass is initially displaced d = -1.0 in from its equilibrium position and released. Determine the period of vibration τ .

The spring is modeled by one discrete element (*ELEMENT_DISCRETE) using a nonlinear elastic spring material (*MAT_S04/*MAT_SPRING_NONLINEAR_ELASTIC). The lumped mass is modeled by an *ELEMENT_MASS entry.

To input the data for the *MAT_S04/*MAT_SPRING_NONLINEAR_ELASTIC it is necessary to convert the stiffness-deflection curve $k(\delta)$ to a force-deflection $F(\delta)$: $F = k\delta = k_0\delta + k_1\delta^3$, using a *DEFINE_CURVE entry. This curve is converted to eleven points of force-deflection points in the range $\delta = [0,1]$.



Figure 18.1 – Sketch representing the model.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Initially Displace	Non- linear	Linear	-	Explicit	-

Units:

 $lbf-s^2/in$, in, s, lbf, psi, lbf-in (blob, inch, second, pound force, pound force/inch², pound force-inch)

Dimensional Data:

 $d = 1.0 \times 10^{\circ}$ in

Material Data:

Nodal Mass	$m = 2.588 \times 10^{-3} \ lbf - s^2 / in$
Spring Stiffness	$k = k_0 + k_1 \delta^2$, $k_0 = 2.0$ <i>lbf</i> / <i>in</i> , $k_1 = 4.0$ <i>lbf</i> / <i>in</i> ³

Element Types:

Translational spring - *SECTION_DISCRETE (dro=0)

Lumped mass (*ELEMENT_MASS entry)

Material Models:

*MAT_S04 or *MAT_SPRING_NONLINEAR_ELASTIC

Results Comparison:

LS-DYNA results for the period of vibration related to this nonlinear spring-mass system are compared with S.P. Timoshenko and D.H. Young studies in *Vibration Problems in Engineering*, 1955 (pg. 141).

	Period of Vibration $ au$ (s)
Timoshenko and Young [1955]	0.14470
Nonlinear spring-mass system	0.14380

This nodal displacement result and the computer period of vibration was generated by *DATABASE_NODOUT keyword (also see Figures 18.2 and 18.3).



Figure 18.2 – Node 1 displacement U_y .



Figure 18.3 – Node 1 displacement U_y (detailed time capture).

Input deck:

*KEYWORD *TITLE Natural Frequency of a Nonlinear Spring-Mass System *CONTROL_TERMINATION \$# endtim endcyc dtmin endeng endmas 0.150000 0 0.0 0.0 0.0 *CONTROL_TIMESTEP \$# dtinit tssfac isdo tslimt dt2ms lctm erode ms1st 1.000e-04 1.000e-04 0 0.0 0.0 0 0 0 \$# dt2msf dt2mslc imscl 0.0 0 0 *DATABASE_NODOUT binary \$# dt 0.000100 *DATABASE_BINARY_D3PLOT \$# dt/cycl lcdt/nr beam npltc psetid 0.001000 *DATABASE_HISTORY_NODE \$# nid1 nid2 nid3 nid4 ni5 nid6 nid7 nid8 1 *PART \$# title nonlinear elastic spring \$# pid secid mid eosid hgid adpopt tmid grav 1 1 1 *SECTION_DISCRETE \$# secid dro kd v0 cl fd 1 0 0.0 0.0 0.0 0.0 cd] td1 \$# 0.0 0.0 *MAT_SPRING_NONLINEAR_ELASTIC \$# mid lcd lcr 1 1 *DEFINE_CURVE \$# lcid sdir sfa sfo offa offo dattyp 0 1.000000 1.000000 1 0.0 0.0 \$# al 01 0.00 0.0000 0.10 0.2040 0.20 0.4320 0.30 0.7080 0.40 1.0240 0.50 1.5000 0.60 2.0640 0.70 2.7720 0.80 3.6480 0.90 4.7160 6.0000 1.00 *ELEMENT_DISCRETE n2 offset \$# eid pid n1 vid s pf 1 1 1 2 0 1.00000 0 1.00000 *ELEMENT_MASS \$# eid id mass pid 0.00258800 2 1 0.00258800 3 2 *NODE \$# nid х У z tc rc 0.0 0.0 0.0 1 2 0.0 1.00000000 0.0 *BOUNDARY_SPC_NODE dofry \$#nid/nsid cid dofx dofy dofz dofrx dofrz 2 0 1 1 1 1 1 1 *END

Notes:

- 1. A somewhat better comparison could be achieved with a more detailed representation of the nonlinear spring stiffness.
- 2. As an alternative, it is possible to model the nonlinear spring with a *BEAM_ELEMENT (with the option discrete) and *MAT_067/*MAT_NONLINEAR_ELASTIC_DISCRETE_BEAM material behavior.
- 3. For simulations with linear stiffness, one would use the following implicit entries and perform a simple eigenvalue analysis:

*C01	TROL_IM	PLICIT_EIGE	NVALUE					
\$#	neig	center	lflag	lftend	rflag	rhtend	eigmth	shfscl
	3	11.000	0 -1	L.00e+29	0	1.00e+29	2	0.0
*COI	JTROL_IM	PLICIT_GENE	RAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	6	1.00e-04	2	1	2			

The period could be obtained directly from the *eigout* results file generated by the *CONTROL_IMPLICIT_EIGENVALUE keyword as shown here:

```
Natural Frequency of a Nonlinear Spring-Mass System

r e s u l t s o f e i g e n v a l u e a n a l y s i s:

|------ frequency -----|

MODE EIGENVALUE RADIANS CYCLES PERIOD

1 -9.094947E-11 9.536743E-06 1.517820E-06 6.588397E+05

2 4.547474E-12 2.132481E-06 3.393948E-07 2.946421E+06

3 4.961360E+03 7.043692E+01 1.121038E+01 8.920301E-02
```

However, for this example, the spring stiffness is nonlinear, represented by a piecewise linear curve. LS-DYNA will make a stiffness, from two forcedisplacement pairs, to compute an eigenvalue. Which pairs used will depend on whether there is an initial offset or not (provided via *ELEMENT_DISCRETE). If there is zero initial offset, the first two force-displacement pairs are used; if there is an initial offset, the two pairs on either side of the offset would be used; if the offset and displacement value are equal, LS-DYNA uses this as the upper pair. Using this stiffness value will not yield a correct period of vibration.

It is not recommended to use the eigenvalue solver for nonlinear simulations

19. Buckling of a Axially Loaded Thin Walled Cylinder

Keywords:

```
*CONTROL_IMPLICIT_GENERAL
*CONTROL_IMPLICIT_SOLUTION
*CONTROL_IMPLICIT_BUCKLE
*CONTROL_IMPLICIT_EIGENVALUE
```

Description:

A cylinder is loaded with a uniform distributed (equal value to each node) line load of P = 1000 lbs (compressive) along the top edge. Determine the critical buckling load.

The lower end of the cylinder is clamped, i.e. fixed for all translational and rotational directions (z = 0: $U_x = U_y = U_z = R_x = R_y = R_z = 0$, the upper end of the cylinder is only fixed in x and y direction (z = L: $U_x = U_y = 0$).

The finite element model is shown in Figure 19.1.

Buckling of a Thin Walled Cylinder Under Compression



Figure 19.1 – Finite element model with applied axial load and boundary nodes (marked with []'s). There are 29 elements axially and 76 elements circumferentially.

Analysis Type:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Force	Linear	Linear	-	Implicit	2-Nonlinear w/BFGS

Units:

 $lbf-s^2/in$, in, s, lbf, psi, lbf-in (blob, inch, second, pound force, pound force/inch², pound force-inch)

Dimensional Data:

 $L = 1.20 \times 10^{2}$ in, $r_{c} = 4.8 \times 10^{1}$ in, $t = 1.0 \times 10^{-1}$ in

Material Data:

Mass Density	$\rho = 1.00 \times 10^{-2} \ lbf - s^2 / in^4$
Young's Modulus	$E = 1.00 \times 10^7 \ lbf \ / \ in^2$
Poisson's Ratio	v = 0.3

Load:

Axial Load	$P = 1.00 \times 10^3$	lbs

Element Types:

Belytschko-Tsay shell (elform=2) S/R Hughes-Liu shell (elform=6) Belytschko-Wong-Chiang shell (elform=10) Fully integrated shell (elform=16)

Material Models:

*MAT_001 or *MAT_ELASTIC

Results Comparison:

LS-DYNA results for the critical buckling load of a thin walled cylinder under axial compression are compared with S.P. Timoshenko and J.M. Gere studies in *Theory of Elastic Stability*, 1961 (pg. 457).

	Critical Buckling Load P _{cr} (lbf)	Critical Axial Stress & (psi)
Timoshenko and Gere [1961]	3.8025×10 ⁵	1.2608×10^{4}
Belytschko-Tsay shell (elform=2)	3.8763×10 ⁵	1.2853×10 ⁴
S/R Hughes-Liu shell (elform=6)	4.6226×10 ⁵	1.5327×10 ⁴
Belytschko-Wong-Chiang shell (elform=10)	3.8763×10 ⁵	1.2853×10 ⁴
Fully integrated shell (elform=16)	4.6086×10 ⁵	1.5281×10^4

The analytical solution for this problem, from Timoshenko and Gere [1961], is:

$$\sigma_{cr}^* = E \frac{t}{R} \frac{1}{\sqrt{3(1-\nu^2)}} = 1.2608 \times 10^4 \text{ psi}$$

with a mode shape that is sinusoidal both axially and circumferentially.

The LS-DYNA critical load P_{cr} is computed from the first eigenvalue and the applied axial load:

$$P_{cr} = \lambda_1 P = 3.8762 \times 10^2 \times 1.0000 \times 10^3 \, lbf = 3.8762 \times 10^5 \, lbf$$

while the critical axial stress $\sigma_{\scriptscriptstyle cr}$ is given by

$$\sigma_{cr} = \frac{P_{cr}}{A} = \frac{3.8763 \times 10^5 \, lbf}{3.0159 \times 10^1 \, in^2} = 1.2853 \times 10^4 \, psi$$

This result, for the one point quadrature shell elements (elform=2 and elform=10), is in good agreement with the analytical solution.

The critical load and axial stress for the fully integrated shell elements (elform=6 and elform=16) is greater than the one point quadrature shell elements. The difference is not understood.

Eigenvalue Results:

From the *eigout* file, generated by the *CONTROL_IMPLICIT_BUCKLE keyword:

Belytschko-Tsay shell (elform=2):

Buckling of a Thin Walled Cylinder Under Compression results of eigenvalue analysis: |----- frequency -----| MODE EIGENVALUE RADIANS CYCLES PERIOD 1 3.876317E+02 2 3.882852E+02 3 3.882852E+02

S/R Hughes-Liu shell (elform=6)

Buckling of a Thin Walled Cylinder Under Compression results of eigenvalue analysis: |----- frequency -----| MODE EIGENVALUE RADIANS CYCLES PERIOD 1 4.622332E+02 2 4.622523E+02 3 4.700114E+02

Belytschko-Wong-Chiang shell (elform=10)

Buckling of a Thin Walled Cylinder Under Compression results of eigenvalue analysis: |------ frequency -----| MODE EIGENVALUE RADIANS CYCLES PERIOD 1 3.876317E+02 2 3.882852E+02 3 3.882852E+02

Fully integrated shell (elform=16)

```
Buckling of a Thin Walled Cylinder Under Compression
results of eigenvalue analysis:
|------ frequency -----|
MODE EIGENVALUE RADIANS CYCLES PERIOD
1 4.608551E+02
2 4.608564E+02
3 4.681777E+02
```

The one point quadrature shell elements (elform =2 and elform=10) only provide the axial sinusoidal mode shape (10 half sine waves) as can be seen in Figure 19.2.

The fully integrated shell elements (elform=6 and elform=16) provide both the axial and circumferential sinusoidal mode shapes (2 half sine waves axially and 20 half sine waves circumferentially) as can be seen in Figure 19.3.

The eigenmodes for the one point quadrature elements and the fully integrated shell elements do not compare well. Again, this difference not understood.



Figure 19.2 – First eigenmode with the 1000 lbf load applied (elform=10).



Figure 19.3 – First eigenmode with the 1000 lbf load applied (elform=16).

Input deck:

*KEYWORD *TITLE Buckling of a Thin Walled Cylinder Under Compression *CONTROL_IMPLICIT_GENERAL \$# imflag dt0 nsbs imform igs cnstn form 1 0.100000 2 1 2 *CONTROL_IMPLICIT_SOLUTION \$# nsolvr ilimit maxref dctol ectol not used lstol rssf 15 0.001000 0.010000 0.0 0.900000 1.000000 2 11 \$# dnorm diverg istif nlprint 2 1 1 2 arcdmp \$# arcctl arcdir arclen arcmth 0 1 0.0 1 2 *CONTROL_IMPLICIT_BUCKLE \$# nmode 3 *CONTROL_IMPLICIT_EIGENVALUE \$# neig center lflag lftend rflag rhtend eigmth shfscl 300.0 *CONTROL_TERMINATION \$# endtim endcyc dtmin endeng endmas 1.000000 0 0.0 0.0 0.0 *CONTROL_SHELL \$# wrpang esort 20.00000 0 irnxx istupd bwc miter theorv proj 0 0 2 1 1 1 \$# rotascl intgrd lamsht tshell nfail1 nfail4 сstypб 0 0.0 *DATABASE_BINARY_D3PLOT \$# dt/cycl 0.010000 *PART \$# title material type # 1 (Elastic) \$# pid secid eosid hgid mid grav adpopt tmid 1 1 1 0 1 *SECTION_SHELL \$# secid elform shrf nip propt qr/irid icomp setyp 0.0 0 0 0 0 0.0 2 Ō 1 1 1 0 \$ 1 б 0.0 1 0.0 1 -1 1 \$ 1 10 0.0 0.0 0 1 1 16 t1 t2 0.0 0.0 \$ 0 1 \$# t1 t3 t4 nloc marea 0.100000 0.100000 0.100000 0.100000 0.0 0 *MAT_ELASTIC \$# mid ro e pr 1 0.0100001.0000e+07 0.300000 da db not used 0.0 0.0 0.0 *HOURGLASS \$# hgid ihq qm ibq ql q2 qb qw 0.0 0.0 0.0 0.0 0.0 1 4 0 *SET_NODE_LIST_TITLE bottom nodes \$# sid da1 da2 da3 da4 solver 1 0.0 0.0 0.0 0.0 \$# nid1 nid2 nid3 nod4 nid5 nid6 nid7 nid8 2 3 4 5 6 7 8 1 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 27 28 29 30 26 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 69 70 65 66 67 68 71 72 74 75 76 73 0 0 0 0 *SET_NODE_LIST_TITLE top nodes \$# sid da1 da2 da3 da4 solver 0.0 2 0.0 0.0 0.0

\$#	nidl	ni	d2	nid3	nod4	nid5	nid6		nid7	nid8
	2205	22	06	2207	2208	2209	2210		2211	2212
	2213	22	14	2215	2216	2217	2218		2219	2220
	2221	22	22	2223	2224	2225	2226		2227	2228
	2229	22	30	2231	2232	2233	2234		2235	2236
	2237	22	38	2239	2240	2241	2242		2243	2244
	2245	22	46	2247	2248	2249	2250		2251	2252
	2253	22	54	2255	2256	2257	2258		2259	2260
	2261	22	62	2263	2264	2265	2266		2267	2268
	2269	22	70	2271	2272	2273	2274		2275	2276
	2277	22	78	2279	2280	0	0		0	0
*BOI	UNDARY_S	PC_SET								
\$#n:	id/nsid	C	id	dofx	dofy	dofz	dofrx		dofry	dofrz
	1		0	1	1	1	1		1	1
*BOI	UNDARY_S	PC_SET								
\$#n:	id/nsid	C	id	dofx	dofy	dofz	dofrx		dofry	dofrz
	2		0	1	1	0	0		0	0
*DEI	FINE_CUR	VE								
\$#	lcid	sd	ir	sfa	sfo	offa	offo		dattyp	
	1		0	0.0	0.0	0.0	0.0			
\$#			al		01					
0.0 1.00000000		.0	0.0							
		13.15789474								
*LO	AD_NODE_	SET								
\$#	nsid	d	of	lcid	sf	cid	m1		m2	m3
2			3	1 -1	.000000					
*ELI	EMENT_SH	ELL								
\$#	eid	pid	n1	n2	n3	n4	n5	nб	n7	n8
	1	1	1	2	78	77	0	0	0	0
	2204	1	2204	2129	2205	2280	0	0	0	0
*NOI	DE									
\$#	nid		х		У		Z	tc	rc	
	1	0.000		48.00000		0.000		0	0	
	2280	3.	963804	47	.836056	120.	00000	0	0	
*ENI	D									

Notes:

- 1. The keyword entry *CONTROL_IMPLICIT_BUCKLE allows for buckling analysis at the end of the static implicit simulation.
- 2. The fully integrated and one point quadrature shell elements are formulated for nonlinear analysis. Although this analysis is linear, it was solved with a nonlinear solution method to demonstrate the use of these elements.
Mesh Convergence Study:

Seven different mesh refinements were studied for this simulation:

original mesh 29 axial by 76 circumferential elements (Figure 19.4a)

• $4.1379 \times 10^{\circ}$ in by $3.9671 \times 10^{\circ}$ in

1st mesh refinement 58 axial by 152 circumferential elements

• $2.0690 \times 10^{\circ}$ in by $1.9840 \times 10^{\circ}$ in

2nd mesh refinement 87 axial by 228 circumferential elements (Figure 19.4b)

• $1.3793 \times 10^{\circ}$ in by $1.3227 \times 10^{\circ}$ in

3rd mesh refinement 116 axial by 304 circumferential elements

• $1.0345 \times 10^{\circ}$ in by $0.9921 \times 10^{\circ}$ in

4th mesh refinement 145 axial by 380 circumferential elements (Figure 19.4c)

• $0.8276 \times 10^{\circ}$ in by $0.7937 \times 10^{\circ}$ in

5th mesh refinement 174 axial by 456 circumferential elements

• $0.6897 \times 10^{\circ}$ in by $0.6614 \times 10^{\circ}$ in

6th mesh refinement 203 axial by 532 circumferential elements

• $0.5911 \times 10^{\circ}$ in by $0.5669 \times 10^{\circ}$ in

Buckling of a Thin Walled Cylinder Under Compression



Figure 19.4a - Finite element model for the original mesh (29 axial by 76 circumferential) discretization with applied axial load.

Buckling of a Thin Walled Cylinder Under Compression



Figure 19.4b - Finite element model for the 2nd mesh refinement (87 axial by 228 circumferential) discretization with applied axial load.

Buckling of a Thin Walled Cylinder Under Compression



Figure 19.4c - Finite element model for the 4th mesh refinement (145 axial by 380 circumferential) discretization with applied axial load.

Mesh Convergence Results Comparison:

LS-DYNA results for the critical buckling load of a thin walled cylinder under axial compression are compared for seven different mesh discretizations.

analytical solution - 3.8025×10^5 <i>lbf</i>	Belytschko-Wong- Chiang (elform=10)	fully integrated shell (elform=16)
original mesh (29 axial by 76 circumferential elements)	3.8763×10 ⁵	4.6086×10 ⁵
1st mesh refinement (58 axial by 152 circumferential elements)	3.8115×10 ⁵	4.0616×10 ⁵
2nd mesh refinement (87 axial by 228 circumferential elements)	3.8052×10 ⁵	3.9313×10 ⁵
3rd mesh refinement (116 axial by 304 circumferential elements)	3.8033×10 ⁵	3.8758×10 ⁵
4th mesh refinement (145 axial by 380 circumferential elements)	3.8026×10 ⁵	3.8480×10 ⁵
5th mesh refinement (174 axial by 456 circumferential elements)	3.8022×10 ⁵	3.8312×10 ⁵
6th mesh refinement (203 axial by 532 circumferential elements)	3.8019×10 ⁵	3.8207×10 ⁵

For the Belytschko-Wong-Chiang (one point quadrature) shell element (elform=10), the 29 axial by 76 circumferential element mesh (original) critical buckling load result was in good agreement with the analytical solution. This element/mesh converged rapidly. The 29 axial by 76 circumferential element mesh only differed by less than 2% from the analytical solution while the 203 axial by 532 circumferential element mesh differed by less than 0.02%.

For the fully integrated shell element (elform=16), the 29 axial by 76 circumferential element mesh (original) critical buckling load result was greater (over 21%) than the analytical solution. It is not known why. Doubling the number of elements axially and circumferentially reduces the critical buckling by about 10%; however, still not in good agreement with the analytical solution, especially considering the level of mesh refinement. Two further mesh refinements (116 axial by 304 circumferential element elements) were required to reach a similar good agreement (the 2% difference) with the one point quadrature shell element (elform=10) and the original mesh discretization. The 203 axial by 532 circumferential element mesh refinement for the fully integrated shell differed by less than 0.5%.

The one point quadrature shell element (elform=10) only provides the axial sinusoidal mode shape (Figures 19.5a and 19.5b):

- 10 half sine waves in 29 axial by 76 circumferential element mesh (original),
- 13 half sine waves in 58 axial by 152 circumferential element mesh,
- 14 half sine waves in 87 axial by 228 circumferential element mesh,
- 15 half sine waves in 116 axial by 304 circumferential element mesh,
- 15 half sine waves in 145 axial by 380 circumferential element mesh,
- 16 half sine waves in 174 axial by 456 circumferential element mesh,
- 16 half sine waves in 203 axial by 532 circumferential element mesh,

estimated from the eigenmode figures. The number of half sine waves is the number of buckles. It is not known why this element formulation only provides the axial sinusoidal modes.



Figure 19.5a - First eigenmode with the 1000 lbf load applied for the six refined mesh discretizations (elform=10) - no contouring.



Figure 19.5b - First eigenmode with the 1000 lbf load applied for the six refined mesh discretizations (elform=10) - resultant displacement contouring.

The fully integrated shell element (elform=16) provides both the axial and circumferential sinusoidal mode shapes (Figures 19.6a and 19.6b):

- 2 half sine waves axially and 20 half sine waves circumferentially in 29 axial by 76 circumferential element mesh (original),
- 3 half sine waves axially and 24 half sine waves circumferentially in 58 axial by 152 circumferential element mesh,
- 4 half sine waves axially and 28 half sine waves circumferentially in 87 axial by 228 circumferential element mesh,
- 5 half sine waves axially and 30 half sine waves circumferentially in 116 axial by 304 circumferential element mesh,
- 6 half sine waves axially and 32 half sine waves circumferentially in 145 axial by 380 circumferential element mesh,
- 6 half sine waves axially and 32 half sine waves circumferentially in 174 axial by 456 circumferential element mesh,
- 7 half sine waves axially and 34 half sine waves circumferentially in 203 axial by 532 circumferential element mesh.

estimated from the eigenmode figures.



Figure 19.6a - First eigenmode with the 1000 lbf load applied for the six refined mesh discretizations (elform=16) - no contouring.



Figure 19.6b - First eigenmode with the 1000 lbf load applied for the six refined mesh discretizations (elform=16) - resultant displacement contouring.

Notes:

- 1. Solution problems may exist:
 - because LS-DYNA buckling solutions assume the first buckling mode will be around 1.0 and/or
 - if numerous eigenvalues are clustered around that smallest bucking frequencies.

For the refined meshes, for this problem, it was necessary to override the internal heuristic for picking a starting point for Lanczos shift strategy, which is the initial Eigen frequency shift. In these cases, the user must specify the initial shift via the parameter shfscl. shfscl should be close to the first nonzero frequency.

20. Membrane with a Hot Spot

Keywords:

*LOAD_THERMAL_LOAD_CURVE *MAT_ELASTIC_PLASTIC_THERMAL

Description:

This benchmark analyzes the behavior of shell elements subjected to a thermal load. Two distinct regions are modeled: the central hot-spot region (radius equal to *r*), subjected to the thermal strain $\varepsilon = \alpha T$, and the rest of the plate, which is at constant temperature with $\varepsilon = 0.0$. Due to symmetry, only ¹/₄ of the plate (side lengths 2*L* and thickness *t*) is modeled (Figures 20.1a and 20.1b).

The material defining the hot spot is *MAT_ELASTIC_PLASTIC_THERMAL (*MAT_004), sensitive to temperature changes. The rest of the plate is defined with material *MAT_ELASTIC (*MAT_001).

The temperature is uniformly applied to the whole model by means of the *LOAD_THERMAL_LOAD_CURVE keyword.

Determine the y-component of the stress tensor along the edge y=0, just outside the hot spot. A fine mesh is required in the region of interest.

To possibly achieve better accuracy, the value at the integration point is considered (values at nodes are interpolated from neighboring integration points).

Membrane with a Hot Spot



Figure 20.1b - Finite element model of hot spot (blue region) and refined surrounding mesh, with selected nodes, element, and dimensions identified.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Thermal	Linear	Linear	-	Implicit	1-Linear

Units:

ton ,mm, s, N, MPa, N-m, $^{\circ}C$ (tonne, millmeter, second, Newton, MegaPascal, Newton-millimeter, degree Centigrade)

Dimensional Data:

L = 10.0 mm, r = 1.00 mm, t = 1.00 mm

Material Data:

Young's Modulus	$E = 1.00 \times 10^5 MPa$
Poisson's Ratio	v = 0.3
Linear Expansion	$\alpha = 1.00 \times 10^{-5} mm / mm /^{\circ} C$

Load:

Thermal	T = 0.0 varied linearly to	$100 \degree C$
---------	----------------------------	-----------------

Element Types:

Fully integrated shell (elform=16)

Material Models:

*MAT_001 or *MAT_ELASTIC

*MAT_004 or *MAT_ELASTIC_PLASTIC_THERMAL

Results Comparison:

LS-DYNA global stress σ_{yy} at point just outside the hot spot (Node 18) is compared with *NAFEMS Background to Benchmark*, Test T1.

Reference Condition - Point Just Outside Hot Spot (Node 18)	Global Stress - σ_{yy} (M <i>Pa</i>)
NAFEMS Benchmark Test T1	5.0000×10^{1}
Element 1148 (average value)	4.7528×10^{1}
First in-plane integration point (2x2 quadrature) - element 1148	4.5476×10 ¹
Node 18	4.3974×10 ¹

The global stress σ_{yy} results were generated from *DATABASE_ELOUT (*elout* file) and *DATABASE_EXTENT_BINARY (*eloutdet* file provides detailed element output at integration points and connectivity nodes) keyword entries.

You can set intout=stress or intout=all (*DATABASE_EXTENT_BINARY) and have stresses output for all the integration points to a file called *eloutdet* (*DATABASE_ELOUT governs the output interval and *DATABASE_HISTORY_ SHELL governs which elements are output). Setting nodout=stress or nodout=all all in *DATABASE_EXTENT_BINARY will write the extrapolated nodal stresses to *eloutdet*.

LS-DYNA stress and strain outputs correspond to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different postprocessors).

Shell element stresses are reported at through-thickness integration points. The location of those integration points depends on the number of integration points and the type of integration rule used, e. g., Gaussian, Lobatto, trapezoidal, user-defined rule (see nip and qr/irid in *SECTION_SHELL). Fully-integrated shell formulations have 4 in-plane integration points at each through-thickness location. For these formulations, the 4 values of each stress component are averaged before being written to *elout* (except for the case of linear analysis when nsolvr=1 in *CONTROL_IMPLICIT_SOLUTION, in which case all 4 stress components are written to *elout*).

Shell element stresses can be shown in the global, element, or material coordinate system. By default, shell element stresses/strains written to *d3plot* are global; shell stresses/strains written to *elout* are in the element local coordinate system (except for the case of linear analysis when nsolvr=1 in *CONTROL_IMPLICIT_SOLUTION, in which case stresses are in the global system). Shell element stresses/strains from *d3plot* are converted by LS-PrePost to the shell element coordinate system.

Even with this fine mesh in the region of interest, the large gradient temperature profile makes it difficult to capture the global stress σ_{YY} along the line of symmetry. The average global stress of the element (Figure 20.2) provides the best comparative value (~5% difference), a few percent better than the nearest element integration point (~9% difference) and the extrapolated nodal (~12% difference) results.



Figure 20.2 -Contour plot of global stress σ_{yy} . Maximum value at element 1148.

Input deck:

*KE	YWORD							
*TI	TLE							
Men	brane wi	th a Hot Sp	ot					
*CC	NTROL_IM	PLICIT_GENE	ERAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	zer0_v
	1	0.100000	2	1	2	0	0	0
*CC	NTROL_IM	PLICIT_SOLU	JTION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	1	11	15	0.001000	0.010000	1.0e+10	0.900000	1.000000
\$#	dnorm	diverg	istif	nlprint	nlnorm	d3itcl	cpchk	
	2	1	1	0	2	0	0	
\$#	arcctl	arcdir	arclen	arcmth	arcdmp	arcpsi	arcalf	arctim
	0	0	0.0	1	2	0.0	0.0	0.0
*CC	NTROL_SH	ELL						
\$#	wrpang	esort	irnxx	istupd	theory	bwc	miter	proj
2	0.00000	0	-1	0	16	2	1	0
\$#	rotascl	intgrd	lamsht	cstyp6	tshell			
1	.000000	0	0	1	0			
*CC	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
1	.000000	0	0.0	0.0	0.0			
*DA	TABASE_E	XTENT_BINAF	RY					

\$#	neiph 0	neips O	maxint 4	strflg 1	sigflg 1	epsflg 1	rtflg 1	engflg 1
\$#	cmpflg	ieverp	beamip	dcomp	shge	stssz	n3thdt	ialemat
\$#	nintsld 1	pkp_sen	sclp 1.0	hydro	msscl	therm	intout stress	nodout stress
*DA \$#	TABASE_E dt/cycl	LOUT						
0	.100000							
*DA \$#	eid1 1148	eid2	eid3	eid4	ei5	eid6	eid7	eid8
*DA	TABASE G	TISTAT						
\$# C	dt/cycl .100000							
*DA \$#	TABASE_M dt/cycl	IATSUM						
C	.100000							
*DA	TABASE_E	SINARY_D3PI	TOT					
\$# ^								
*D7	.100000							
\$#	title							
Par	t	1 for M	lat	1 and Elem	ı Type	16		
\$#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
	1	1	1					
*SE	CTION_SH	IELL						
\$#	secid	elform	shrf	nip	propt	qr/irid	icomp	setyp
ĊШ	⊥ ⊬1	16	0.830000		1 m] n m	0.0	0	T
Ş# 1		1 000000	1 000000	1 000000	01100	marea 0 0		
*MA	T ELASTI	1.000000	1.000000	1.000000	0	0.0		
\$#	mid	ro	е	pr	da	db	not used	
	1	1	L.0000e+05	0.300000	0.0	0.0	0.0	
*PA	ART							
\$#	title							
Par	t	2 for N	lat	2 and Elem	ITVDE	16		
1 011					. 1720	10		
\$#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
\$#	pid 2	secid 2	mid 2	eosid	hgid	grav	adpopt	tmid
\$# *SE \$#	pid 2 CTION_SH secid	secid 2 HELL elform	mid 2 shrf	eosid	hgid	grav	adpopt	tmid
\$# *SE \$#	pid 2 CTION_SH secid 2	secid 2 HELL elform 16	mid 2 shrf 0.830000	eosid nip 1	hgid propt	grav qr/irid 0.0	adpopt icomp 0	tmid setyp 1
\$# *SE \$# \$#	pid 2 CTION_SH secid 2 t1	secid 2 HELL elform 16 t2	mid 2 shrf 0.830000 t3	eosid nip 1 t4	propt 1 nloc	grav qr/irid 0.0 marea	adpopt icomp 0	tmid setyp 1
\$# *SE \$# \$#	pid 2 CTION_SH secid 2 t1	secid 2 HELL elform 16 t2 1.000000	mid 2 shrf 0.830000 t3 1.000000	eosid nip 1 t4 1.000000	propt 1 1 0	grav qr/irid 0.0 marea 0.0	adpopt icomp 0	tmid setyp 1
\$# *SE \$# \$# 1 *MA	pid 2 CTION_SH secid 2 t1 000000 T_ELASTI	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_	mid 2 shrf 0.830000 t3 1.000000 _THERMAL	eosid nip 1 t4 1.000000	hgid propt 1 nloc 0	grav qr/irid 0.0 marea 0.0	adpopt icomp 0	tmid setyp 1
\$# *SE \$# \$# *MA \$#	pid 2 CTION_SH secid 2 tl 000000 NT_ELASTI mid	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro	mid 2 shrf 0.830000 t3 1.000000 _THERMAL	eosid nip 1 t4 1.000000	hgid propt 1 nloc 0	grav qr/irid 0.0 marea 0.0	adpopt icomp 0	tmid setyp 1
\$# *SE \$# \$# 1 *MA \$# \$#	pid 2 CCTION_SH secid 2 t1 000000 NT_ELASTI mid 2 t1	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro	mid 2 shrf 0.830000 t3 1.000000 _THERMAL	eosid nip 1 t4 1.000000	hgid propt 1 nloc 0	grav qr/irid 0.0 marea 0.0	adpopt icomp 0	tmid setyp 1
\$# *SE \$# \$# *MA \$# \$#	pid 2 CCTION_SH secid 2 t1 000000 NT_ELASTI mid 2 t1 0 0	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0	eosid nip 1 t4 1.000000	hgid propt 1 nloc 0 t5	grav qr/irid 0.0 marea 0.0	adpopt icomp 0 t7	tmid setyp 1 t8
\$# \$# \$# \$# \$# \$# \$#	pid 2 CCTION_SH secid 2 t1 000000 ST_ELASTI mid 2 t1 0.0 e1	secid 2 HELL elform 16 t2 1.000000 CC_PLASTIC_ ro t2 1000.000 e2	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3	eosid nip 1 t4 1.000000 t4 0.0 e4	hgid propt 1 nloc 0 t5 0.0 e5	grav qr/irid 0.0 marea 0.0 t6 0.0 e6	adpopt icomp 0 t7 0.0 e7	tmid setyp 1 t8 0.0 e8
\$# \$# \$# \$# \$# \$# \$# \$# \$# \$#	pid 2 CTION_SH secid 2 t1 000000 T_ELASTI mid 2 t1 0.0 e1 000e+05	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2 1000.000 e2 1.000e+05	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3 0.0	eosid nip 1 t4 1.000000 t4 0.0 e4 0.0	t5 0.0 0	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0	adpopt icomp 0 t7 0.0 e7 0.0	tmid setyp 1 t8 0.0 e8 0.0
\$# \$SE \$# \$# \$# \$# \$# \$# \$# \$#	pid 2 CTION_SH secid 2 t1 000000 T_ELASTI mid 2 t1 0.0 e1 000e+05 pr1	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3 0.0 pr3	eosid nip 1 t4 1.000000 t4 0.0 e4 0.0 pr4	t5 0.0 propt 1 nloc 0	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6	adpopt icomp 0 t7 0.0 e7 0.0 pr7	tmid setyp 1 t8 0.0 e8 0.0 pr8
\$# *SE \$# \$# \$# \$# \$# \$# \$# \$# 0	pid 2 CTION_SH secid 2 t1 000000 T_ELASTI mid 2 t1 0.0 e1 000e+05 pr1 0.300000	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.300000	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3 0.0 pr3 0.0	eosid nip 1 t4 1.000000 t4 0.0 e4 0.0 pr4 0.0	t5 0.0 propt 1 nloc 0 t5 0.0 pr5 0.0	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6 0.0	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0
\$# \$SE \$# \$# \$# \$# \$# \$# \$# \$#	pid 2 CTION_SH secid 2 t1 000000 T_ELASTI mid 2 t1 0.0 el 000e+05 pr1 0.300000 alphal	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.300000 alpha2	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3	eosid nip 1 t4 1.000000 t4 0.0 e4 0.0 pr4 0.0 alpha4	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6 0.0 alpha6	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8
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\$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SF secid 2 t1 000000 ST_ELASTI mid 2 t1 0.00 el 000e+05 pr1 0.300000 alphal 000e-05 sigy1	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.300000 alpha2 1.000e-05 sigy2 0.0	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0	eosid nip 1 t4 1.000000 t4 0.0 e4 0.0 pr4 0.0 alpha4 0.0 sigy4 0.0	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0 0	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8
\$# \$\$ \$\$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SE secid 2 t1 000000 ST_ELASTI mid 2 t1 0.00 el 0.00e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etanl	secid 2 HELL elform 16 t2 1.000000 cC_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.300000 alpha2 1.000e-05 sigy2 0.0 etan2	mid 2 shrf 0.830000 t3 1.000000 _THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3	eosid nip 1 t4 1.000000 t4 0.0 e4 0.0 pr4 0.0 alpha4 0.0 sigy4 0.0 etan4	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5	grav gr/irid 0.0 marea 0.0 t6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 etan6	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0.0 etan7	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 sigy8 0.0
\$# \$\$ \$\$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SF secid 2 t1 000000 NT_ELASTI mid 2 t1 0.00 el 0.00e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etanl 0.0	secid 2 HELL elform 16 t2 1.000000 CC_PLASTIC_ ro t2 1.000e+05 pr2 0.300000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0	mid 2 shrf 0.830000 1.00000 THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0	eosid nip 1 t4 1.000000 e4 0.0 e4 0.0 pr4 0.0 alpha4 0.0 sigy4 0.0 etan4 0.0	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0	grav gr/irid 0.0 marea 0.0 t6 0.0 pr6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 etan6 0.0	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 sigy8 0.0
\$# \$\$ \$# \$# \$# \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SF secid 2 t1 000000 NT_ELASTI mid 2 t1 0.00 el 000e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etan1 0.0 .EMENT_SF	secid 2 HELL elform 16 t2 1.00000 CC_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.300000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0 NELL	mid 2 shrf 0.830000 1 1.00000 THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0	eosid nip 1 t4 1.000000 e4 0.0 e4 0.0 pr4 0.0 alpha4 0.0 sigy4 0.0 etan4 0.0	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0	grav gr/irid 0.0 marea 0.0 t6 0.0 pr6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 etan6 0.0	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0
\$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SF secid 2 t1 000000 NT_ELASTI mid 2 t1 0.00 el 0.00e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etan1 0.0 EMENT_SF eid	secid 2 HELL elform 16 t2 1.00000 CC_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.30000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0 UELL pid	mid 2 shrf 0.830000 t3 1.00000 THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0 n1 m	eosid nip 1 t4 1.000000 eta 0.0 eta 0.0 gr4 0.0 sigy4 0.0 sigy4 0.0 etan4 0.0	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0 n4	grav gr/irid 0.0 marea 0.0 t6 0.0 pr6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 sigy6 0.0 n5	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0 n6 n7	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0
\$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SF secid 2 t1 000000 NT_ELASTI mid 2 t1 0.00 el 0.00e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etanl 0.0 SEMENT_SF eid 1	secid 2 HELL elform 16 t2 1.000000 CC_PLASTIC_ ro t2 1.000e+05 pr2 0.300000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0 HELL pid 2	mid 2 shrf 0.830000 t3 1.00000 THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0 n1 n 344 36	eosid nip 1 t4 1.000000 e4 0.0 e4 0.0 pr4 0.0 alpha4 0.0 sigy4 0.0 sigy4 0.0 etan4 0.0	hgid hgid propt 1 nloc 0 t5 0.0 e5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0 etan5 0.0	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 sigy6 0.0 pr5	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0 n6 n7	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0 n8
\$ * \$ # 12 \$ * \$ # 12 \$ * \$ # \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 CCTION_SF secid 2 t1 000000 NT_ELASTI mid 2 t1 0.00 el 0.00e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etanl 0.0 EMENT_SF eid 1	secid 2 HELL elform 16 t2 1.00000 C_PLASTIC_ ro t2 1.000e+05 pr2 0.30000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0 HELL pid 2	mid 2 shrf 0.830000 t3 1.00000 THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0 sigy3 0.0 etan3 0.0	eosid nip 1 t4 1.000000 eta 1.000000 eta 1.000000 eta 1.000 eta 1.000 eta 1.000 eta 1.00000 eta 1.00000 eta 1.000000 eta 1.00000 eta 1.00000 eta 1.00000 eta 1.00000 eta 1.00000 eta 1.00000 eta 1.00000 eta 1.00000 eta 1.000000 eta 1.000000 eta 1.000000000000000000000000000000000000	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0 etan5 0.0 n4 345	grav gr/irid 0.0 marea 0.0 t6 0.0 pr6 0.0 sigy6 0.0 sigy6 0.0 etan6 0.0 n5	adpopt icomp 0 t7 0.0 e7 0.0 pr7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0 n6 n7	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0 r8
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\$# E# \$\$ \$\$ \$\$ \$\$ \$\$ \$\$ \$\$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 2CTION_SF secid 2 11 000000 NT_ELASTI mid 2 t1 0.00 el 0.00e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etanl 0.0 etanl 0.0 .EMENT_SF eid 1 1456 DDE nid 1 541	secid 2 HELL elform 16 t2 1.000000 CC_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.30000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0 etan2 0.0 UELL pid 2 2 1.00000 2.98852	mid 2 shrf 0.830000 t3 1.00000 THERMAL t3 0.0 e3 0.0 pr3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0 etan3 0.0 figy3 0.0 etan4 2 etan4 2 etan3 0.0 etan4 2 etan4 2 etan4 2 etan4 2 etan4 2 etan4 2 etan4 2 etan4 2 etan4 2 etan4 2 etan4 etan4 2 etan4 2 etan4 et	eosid nip 1 t4 1.000000 eta 0.0 eta 0.0 eta 0.0 eta 4 0.0 sigy4 0.0 eta 4 0.0 eta 4 0.0 eta 4 0.0 eta 4 1 2 4 33 8 4 41 2 2 4 33 9 0.0 0 2.92062616	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0 sigy5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 etan6 0.0 n5 z 0.0 0.0	adpopt icomp 0 t7 0.0 e7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0 n6 n7	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0 rn8
\$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$ \$	pid 2 2CTION_SF secid 2 11 000000 NT_ELASTI mid 2 1000e+05 pr1 0.000e+05 pr1 0.300000 alphal 000e-05 sigy1 0.0 etanl 0.0 etanl 1 1456 DDE nid 1 1541 UUNDARY_S	secid 2 HELL elform 16 t2 1.000000 C_PLASTIC_ ro t2 1000.000 e2 1.000e+05 pr2 0.30000 alpha2 1.000e-05 sigy2 0.0 etan2 0.0 etan2 0.0 UELL pid 2 2 1.00000 EELL pid 2 2.98852 SPC_SET	mid 2 shrf 0.830000 t3 1.000000 THERMAL t3 0.0 e3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0 etan3 0.0 alpha3 0.0 sigy3 0.0 etan3 0.0 2444 7	eosid nip 1 t4 1.000000 eta 0.0 eta 0.0 eta 4 0.0 sigy4 0.0 eta 4 0.0 eta 4 0.0 eta 4 0.0 eta 4 1 2 4 33 8 4 41 2 2 4 33 9 0.0 0	hgid propt 1 nloc 0 t5 0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0 sigy5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0	grav gr/irid 0.0 marea 0.0 t6 0.0 e6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 etan6 0.0 n5 z 0.0	adpopt icomp 0 t7 0.0 e7 0.0 alpha7 0.0 sigy7 0.0 etan7 0.0 n6 n7 tc rc	tmid setyp 1 t8 0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0 n8

	3	0	0	1	1	1	1	1
	4	0	1	0	1	1	1	1
	5	0	1	1	1	1	1	1
	6	0	0	0	1	1	1	0
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xsym		3 - 1	-1 - 0	1-2	-1 - 4	1		
ŞĦ	sia	dal	daz	da3	da4	solver		
	3	0.0	0.0	0.0	0.0			
\$#	nidl	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	19	20	181	182	183	185	323	324
	325	326	327	328	330	332	17	306
	304	302	301	300	299	297	296	172
	170	169	167	48	47	46	45	18
	1	11	12	186	187	188	222	334
	225	226	227	220	107	270	270	254
	335	330	337	330	0	272	270	200
	267	265	264	155	153	151	82	81
*SET	_NODE_LIS	ST_TITLE						
ysym	m							
\$#	sid	dal	da2	da3	da4	solver		
	4	0.0	0.0	0.0	0.0			
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
4.0	36	83	84	147	150	154	257	258
	261	262	269	271	216	215	210	211
	201	202	209	2/1	310	313	212	21
	308	307	1//	1/5	1/3	38	37	33
	89	90	91	130	135	137	140	221
	224	231	232	235	236	239	242	65
	282	279	278	277	276	275	274	273
	160	158	157	156	68	67	66	61
*SET	NODE LIS	T TITLE						
xv		_						
¢#	eid	dal	da2	da3	da4	golver		
γπ	51G	0 0	0.0	0.0	0 0	SOLVCI		
a u	5	0.0	0.0	0.0	0.0			
ŞĦ	nial	nid2	nid3	n1d4	nias	nido	nid/	nias
	80							
*SET	_NODE_LIS	T_TITLE						
whol	e							
\$#	sid	da1	da2	da3	da4	solver		
	6	0.0	0.0	0.0	0.0			
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
4.0	1	2	3	4	5	6	7	8
	-	2	5	1	5	0	/	0
	1 5 9 5	1 = 2 0	1 5 3 0	1 = 4 0	1 - 41			
	1537	1538	1539	1540	1541			
*LOA	D_THERMAL	_LOAD_CU	IRVE					
\$#	lcid	lciddr						
	1	0						
*DEF	INE_CURVE]						
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp	,
	1	0	1.000000	1.000000	0.0	0.0		
\$#		a1		01				
		0 0		0 0				
	1 0	0.00	1 0	0.00				
* 17117	1.0		10	0.0000000				
ГЛИД								

Notes:

1. The fully integrated and one point quadrature shell elements are formulated for nonlinear analysis. Although this analysis is linear, it could have been solved with a nonlinear solution method (nsolvr=2) which provides some slightly different stress output options (see Notes 2 and 3). This was done for this simulation and it was found to yield results with small differences from the linear analysis (nsolvr=1).

21. 1D Heat Transfer with Radiation

Keywords:

```
*CONTROL_SOLUTION
*CONTROL_THERMAL_SOLVER
*CONTROL_THERMAL_NONLINEAR
*CONTROL_THERMAL_TIMESTEP
*BOUNDARY_TEMPERATURE_SET
*BOUNDARY_RADIATION_SET
*MAT_THERMAL_ISOTROPIC
```

Description:

A 0.10 *m* long bar (L_z) , with square 0.01 m $(L_x) \ge 0.01$ m (L_y) cross-section (Figure 21.1), radiates (steady state) to an ambient temperature of $T = 300^{\circ}K$ at one end (node 11). The other end (node 1) is maintained at constant temperature $T = 1000^{\circ}K$. The bar is perfectly insulated along its length. There is zero internal heat generation.

Find the temperature at node 33 (x=0.000 *m*, y=0.010 *m*, z=0.100 *m*).

The bar is meshed with 40 elements: ten elements along the length and four elements in the cross section.

1D Heat Transfer with Radiation



Figure 21.1 - Finite element model with selected nodes and dimensions identified.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Steady State	Thermal	Linear	Linear	-	Thermal Non-Linear	3-Diagonal scaled conjugate gradient

Units:

kg, m, s, N, Pa, N-m, °C (kilogram, meter, second, Newton, Pascal, Newton-meter, degree Centigrade) - Joule (J) is a N-m, Watt (W) is a J/s, 1° $\Delta C = 1° \Delta K$

Dimensional Data:

 $L_x = 0.01 \ m$, $L_y = 0.01 \ m$, $L_z = 0.10 \ m$

Material Data:

Mass Density	$\rho = 7.850 \times 10^3 \ kg \ / \ m^3$
Heat Capacity	$C_p = 4.600 \times 10^2 \ J / kg \ ^{\circ}C$
Thermal Conductivity	$k = 5.560 \times 10^1 W / m^{\circ}C$
Emissivity	$\varepsilon = 0.980$
Stefan-Boltzman	$\sigma = 5.670 \times 10^{-8} W / m^2 ° K^4$

Load:

Thermal	T = 1000.0 °K (constant)
Convection	$h = 7.500 \times 10^2 W / m^2 °C$
Initial Temperature	$T_0 = 300.0 ^{\circ} K$ (all nodes)

Element Types:

Fully integrated S/R solid (elform=2)

Material Models:

*MAT_T01 or *MAT_THERMAL_ISOTROPIC

Results Comparison:

LS-DYNA bar temperature at x=0.000 m, y=0.010 m, z= 0.100 m (Node 33) is compared with *NAFEMS Background to Benchmark*, Test T2.

Reference Condition - Point Along Bar 0.1 m (Node 33) from Hot End (Node 23)	Temperature ($^{\circ}K$)
NAFEMS Benchmark Test T2	9.2700×10^2
Node 33	9.2700×10^{2}

The fully integrated selectively reduced solid element (elform=2) model (also see Figure 21.2) provides an exact temperature comparison for this coarse mesh.

The problem is 1D, although solved in 3D. The results are, as expected, the same for all x-y planes along the Z-direction.



Figure 21.2 -Contour plot of temperatures with nodes 23 and 33 specifications.

Input deck:

*KEYWORD *TITLE 1D Heat Transfer with Radiation *CONTROL_SOLUTION \$# soln nlq 1 0 isnan lcint ± 0 0 *CONTROL_THERMAL_SOLVER \$# at voe 0 \$# atype ptype solver cgtol gpt eqheat fwork sbc 0 1 3 1.00e-06 8 1.000000 1.0000005.6700e-08 *CONTROL_THERMAL_NONLINEAR \$# refmax tol dcp 20 1.000e-06 0.500000 *CONTROL_THERMAL_TIMESTEP \$# ts tip its tmin tmax dtemp tscp 1 0.500000 1.000e-04 1.000e-04 0.100000 1.000000 0.500000 *CONTROL_TERMINATION \$# endtim endcyc 1.00000 0 *DATABASE_TPRINT endeng dtmin endmas 0.0 0.0 0.0 \$# dt binary 1.000000 0 ioopt lcur 0 1 *DATABASE HISTORY NODE nid4 nid5 nid6 nid7 nid8 66 77 88 \$# nid1 nid2 nid3 22 33 44 11 99 55 77 *DATABASE_BINARY_D3PLOT psetid \$# dt lcdt npltc beam 0 0 1.000000 0 0 *PART \$# title Part 1 for TMat 1 and Elem Type \$# pid secid mid eosid hgi 1 1 0 0 *SECTION_SOLID 2 eosid hgid grav adpopt tmid 0 0 0 1 \$# secid elform 1 2 aet 1 *MAT_THERMAL_ISOTROPIC \$# tmid tro tgrlc 1 7850.000 0.0 \$# hc tc 460.0000 55.6000 tgmult 0.0 *BOUNDARY_TEMPERATURE_SET \$# nsid lcid cmult 1 0 1000.000 loc 0 *BOUNDARY RADIATION SET \$# ssid type 2 1 \$# flcid fmult tilcid timult 05.5566e-08 0 300.0000 loc 0 *INITIAL_TEMPERATURE_SET \$# nsid temp loc 3 300.0000 0 *SET_NODE_LIST_TITLE А da2 0.0 nid? sid dal 1 0.0 nid1 nid2 da3 da4 \$# 0.0 0.0 nid2 nid4 nid5 nid6 nid7 nid8 \$# nid1 1 23 34 45 56 12 67 78 89 *SET_SEGMENT_TITLE В \$# sid da1 da2 da3 da4 da3 0.000 2 0.000 0.000 0.000 \$# n2 n4 a2 a3 n1 n3 al a4 0.000 0.000 55 22 11 44 0.000 0.000 11 22 0.000 55 77 0.000 0.000 66 33 0.000 88 55 44 0.000 0.000 88 0.000 55 0.000 0.000 66 0.000 99

*SE	T_NODE_L	IST_TITL	E							
cent	tral									
\$#	sid	da	1	da2	da3	da4				
	3	0.	0	0.0	0.0	0.0				
\$#	nid1	nid	2	nid3	nid4	nid5	nid6		nid7	nid8
	1		2	3	4	5	6		7	8
	9	1	0	11	12	13	14		15	16
	17	1	8	19	20	21	22		23	24
	25	2	6	27	28	29	30		31	32
	33	3	4	35	36	37	38		39	40
	41	4	2	43	44	45	46		47	48
	49	5	0	51	52	53	54		55	56
	57	5	8	59	60	61	62		63	64
	65	6	6	67	68	69	70		71	72
	73	7	4	75	76	77	78		79	80
	81	8	2	83	84	85	86		87	88
	89	9	0	91	92	93	94		95	96
	97	9	8	99						
*ELI	EMENT_SO	LID								
\$#	eid	pid	nl	n2	n3	n4	n5	nб	n7	n8
	1	1	1	34	45	12	2	35	46	13
	40	1	54	87	98	65	55	88	99	66
*NOI	DE									
\$#	nid		x		У		z	tc	rc	
	1		0.0		0.0		0.0			
	99	0.010	00000	0.01	000000	0.1000	0000			
*ENI	D									

Notes:

- 1. The problem must be flagged as nonlinear if any boundary condition parameter is a function of temperature. This includes a linear (i.e., straight line) relationship. Iterations are needed to obtain the correct solution. Radiation is a T^4 boundary condition.
- 2. The *CONTROL_THERMAL_NONLINEAR keyword is optional. For example, the default values for remax (maximum number of iterations allowed per time step), tol (temperature convergence tolerance), and dcp (divergence control tolerance) will be used, if the nonlinear keyword is omitted, with ptype>0 on *CONTROL_THERMAL_SOLUTION keyword.

22. 1D Transient Heat Transfer in a Bar

Keywords:

```
*CONTROL_SOLUTION
*CONTROL_THERMAL_SOLVER
*CONTROL_THERMAL_TIMESTEP
*BOUNDARY_TEMPERATURE_SET
*INITIAL_TEMPERATURE_SET
*MAT_THERMAL_ISOTROPIC
```

Description:

A 0.1 *m* long bar (L_z) , with square 0.01 m $(L_x) \ge 0.01$ m (L_y) cross-section (Figure 22.1), is subjected at one end (node 6) to a varying thermal with the following law: $T = 100 \sin(\pi t/40)^{\circ}C$. The other end (node 1) is maintained at constant temperature $T = 0^{\circ}C$. The bar is perfectly insulated along its length.

Determine the temperature at 0.02 m from the hot end after 32 seconds.

The bar is meshed with 20 elements: five elements along the length and four elements in the cross section.

1D Transient Heat Transfer in a Bar



Figure 22.1 - Finite element model with selected nodes and dimensions identified.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Thermal Transient	Thermal	Linear	Linear	-	Thermal Linear	3-Diagonal scaled conjugate gradient

Units:

kg, m, s, N, Pa, N-m, $^{\circ}C$ (kilogram, meter, second, Newton, Pascal, Newton-meter, degree Centigrade) - Joule (J) is a N-m, Watt (W) is a J/s

Dimensional Data:

 $L_x = 0.01 m$, $L_y = 0.01 m$, $L_z = 0.10 m$

Material Data:

Mass Density	$\rho = 7.200 \times 10^3 \ kg \ / \ m^3$
Heat Capacity	$C_p = 4.405 \times 10^2 \ J / kg^{\circ}C$
Thermal Conductivity	$K = 3.500 \times 10^1 W / m^{\circ}C$

Load:

Thermal	$T = 100 \degree C$ (constant)
Convection	$h = 7.500 \times 10^2 W / m^2 °C$
Initial Temperature	$T_0 = 0 ^{\circ}C$ (all nodes)

Element Types:

Fully integrated S/R solid (elform=2)

Material Models:

*MAT_T01 or *MAT_THERMAL_ISOTROPIC

Results Comparison:

LS-DYNA bar temperature at x=0.0050 m, y=0.005 m, z=0.080 m (Node 35) is compared with *NAFEMS Background to Benchmark*, Test T3.

Reference Condition - Point Along Bar 0.2 m (Node 35) from Hot End (Node 36)	Temperature (°C)
NAFEMS Benchmark Test T3	3.6600×10^{1}
Node 35	3.4861×10 ¹

The fully integrated selectively reduced solid element (elform=2) model (Figure 22.2) provides a reasonable temperature comparison for this coarse mesh.

The problem is 1D, although done in 3D. The results are, as expected, the same for all x-y planes along the Z-direction.



Figure 22.2 -Contour plot of temperatures at time =32.0 seconds with nodes 35 and 36 specifications.

The histories of temperature for two nodes (35 and 36) used in the comparison are shown in Figure 22.3.



Figure 22.3 - Temperature histories for nodes 35 and 56.

Input deck:

*KE	YWORD							
*TT	TLE .							
1D '	Transien	t Heat Tra	anster in a	a Bar				
*C0	NTROL_SO	LUTION						
\$#	soln	nlq	isnan	lcint				
	1	0	0	0				
*C0	NTROL_TH	ERMAL_SOL	/ER					
\$#	atype	ptype	solver	cgtol	gpt	eqheat	fwork	sbo
	1	0	3	1.00e-06	8	1.000000	1.000000	0.0
*C0	NTROL_TH	ERMAL_TIME	ESTEP					
\$#	ts	tip	its	tmin	tmax	dtemp	tscp	
	1	0.500000	1.000e-04	1.000e-04	0.100000	1.000000	0.500000	
*C0	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
3	2.00000	0	0.0	0.0	0.0			
*DA	TABASE_T	PRINT						
\$#	dt	binary	lcur	ioopt				
1	.000000	0	0	1				
*DA	TABASE_H	ISTORY_NOI	DE					
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	5	6	11	12	17	18	23	24
	29	30	35	36	41	42	47	48
	53	54						
*DA	TABASE_B	INARY_D3PI	LOT					
\$#	dt	lcdt	beam	npltc	psetid			
1	.000000	0	0	0	0			
*PA	RT							
\$#	title							
Par	t	1 for 7	ſMat	1 and Ele	em Type	2		

\$#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
* ୧୮୯	1 TION SC	1 תד.דו	0	0	0	0	0	1
\$#	secid	elform	aet					
*MAT	THERMA	L ISOTROPI	C					
\$#	tmid	- tro	tgrlc	tgmult				
\$#	1 hc	7200.000 tc	0.0	0.0				
44	0.5000	35.0000						
*BOU	NDARY_T	'EMPERATURE	SET	1.0.7				
ŞĦ	11510 1	1010		100				
*BOU	NDARY_T	EMPERATURE	_SET	0				
\$#	nsid	lcid	cmult	loc				
	2	1	1.000000	0				
*INI	TIAL_TE	MPERATURE	SET					
ŞĦ	nsia 3	cemp 0 0	TOC					
*SET	NODE L	IST TITLE	0					
A		_						
\$#	sid	dal	da2	da3	da4			
±	1	0.0	0.0	0.0	0.0			
\$#	nidl	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	⊥ 49	/	13	19	25	31	37	43
*SET	_NODE_L	IST_TITLE						
В								
\$#	sid	dal	da2	da3	da4			
чщ	2	0.0	0.0	0.0	0.0	nide	nid7	2140
ъ н	6	12	11103	24	30	36	42	48
	54	12	10	21	50	50	12	10
*SET	_NODE_L	IST_TITLE						
cent	ral							
\$#	sid	da1	da2	da3	da4			
¢#	3 nid1	0.0 nid2	0.0 nid3	0.0 nid4	0.0 nid5	nid6	nid7	nid8
γĦ	1	2	3	4	5	6	7	8
	9	10	11	12	13	14	15	16
	17	18	19	20	21	22	23	24
	25	26	27	28	29	30	31	32
	33	34	35	36	37	38	39	40
	41 49	42	43	44 52	45	46 54	47	48
*DEF	INE CUR	VE	51	52	55	51		
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp	
	1	0	0.0	0.0	0.0	0.0		
\$#		al		01				
	1	0.0	7	0.0				
	2	.00000000	15	63558102				
	3	.00000000	23	.33292198				
	4	.00000000	30	.88655281				
	5	.00000000	38	.24995041				
	6	.00000000	45	.37776184				
	7	.00000000	52	.22608948				
	8	.00000000	58	01754012				
	10	00000000	70	68251801				
	11	.00000000	76	.01214600				
	12	.00000000	80	.87360382				
	13	.00000000	85	.23696136				
	14	.00000000	89	.07533264				
	15	.00000000	92	.36508179				
	10 17	.00000000	95	.08594513				
	1 A	.00000000	97	75759888				
	19	.00000000	99	.68576813				
	20	.00000000	99	.99996948				
	21	.00000000	99	.69825745				

		22.000000 23.000000 24.000000	0 0 0	98.782 97.258 95.135	250122 333130 513947					
		25.0000000	0	92.426	00250					
		27.0000000	0	85.320	13702					
		28.0000000	0	80.967	17834					
		29.000000	0	76.115	53955					
		30.000000	0	70.795	508972					
		31.000000	0	65.038	861237					
		32.000000	0	58.881	55746					
*ELE	MENT_	SOLID								
\$#	eid	pid	nl	n2	n3	n4	n5	n6	n7	n8
	1	1	1	19	25	7	2	20	26	8
	20	1	29	47	53	35	30	48	54	36
*NOD	E									
\$#	nid		x		У		Z	tc	rc	
	1		0.0		0.0		0.0			
*END	54	0.010	00000	0.010	00000	0.100	00000			

Notes:

23. 2D Heat Transfer with Convection

Keywords:

```
*CONTROL_SOLUTION
*CONTROL_THERMAL_SOLVER
*CONTROL_THERMAL_TIMESTEP
*MAT_THERMAL_ISOTROPIC
*BOUNDARY_CONVECTION_SET
*BOUNDARY_TEMPERATURE_SET
*INITIAL_TEMPERATURE_SET
```

Description:

A slab ($L_y = 1.00 \ m$ in depth) of rectangular cross-section ($L_x = 0.60 \ m$ by $L_z = 1.00 \ m$) shown in Figure 23.1 is subjected to the following thermal loads for a steady state simulation:

- constant Temperature $T_0 = 100^{\circ} C$ on the face defined by nodes 78-79-85-95,
- natural convection to the ambient temperature $T_a = 0^{\circ}C$ on the faces defined by nodes 12-85-79-18 and 1-12-18-2 (convection coefficient $h = 7.50 \times 10^2 W / m^2 {}^{\circ}C$),
- face defined by nodes 1-2-78-95 is adiabatically insulated.

Find the temperature at node 225 (x=0.60 m, y=1.00 m, z=-0.20 m).



Figure 23.1 - Finite element model with selected nodes and dimensions identified.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Steady State	Thermal	Linear	Linear	-	Thermal	3-Diagonal scaled conjugate gradient

Units:

kg, m, s, N, Pa, N-m, $^{\circ}C$ (kilogram, meter, second, Newton, Pascal, Newton-meter, degree Centigrade) - Joule (J) is a N-m, Watt (W) is a J/s

Dimensional Data:

 $L_x = 0.60 m$, $L_y = 1.00 m$, $L_z = 1.00 m$

Material Data:

Mass Density	$\rho = 8.000 \times 10^3 \ kg \ / \ m^3$
Heat Capacity	$C_p = 1.000 \times 10^0 J / kg^{\circ}C$
Thermal Conductivity	$k = 5.200 \times 10^1 W / m^{\circ}C$

Load:

Thermal	$T = 100 \degree C$ (constant)
Convection	$h = 7.500 \times 10^2 W / m^2 °C$

Element Types:

Fully integrated S/R solid (elform=2)

Material Models:

*MAT_T01 or *MAT_THERMAL_ISOTROPIC

Results Comparison:

LS-DYNA slab edge temperature at x=0.60 m, y=1.00 m, z=-0.20 m (Node 225) are compared with *NAFEMS Background to Benchmark*, Test T4.

Reference Condition - Point Along Slab Edge (Node 225)	Temperature (°C)
NAFEMS Benchmark Test T4	1.8300×10^{1}
Node 225	1.7954×10^{1}

The fully integrated selectively reduced solid element (elform=2) model (Figure 23.2) provides a reasonable temperature comparison for this relatively coarse mesh.

The problem is 2D, although solved in 3D. The results are, as expected, the same for all planes in the 3rd dimension (Y-direction in this case).



Figure 23.2 -Contour plot of temperatures with node 225 specification.

Input deck:

*KEY	WORD							
*TIT	LE 		a					
*CON	eat Tra TROL_SO	nsier with LUTION	Convectio	n 				
\$#	soln 1	nlq 0	isnan 0	lcint 0				
*CON	TROL_TH	ERMAL_SOLVE	R					
\$#	atype 0	ptype 0	solver 3	cgtol 1.00e-06	gpt 8	eqheat 1.000000	fwork 1.000000	sbc 0.000000
*CON	TROL_TE	RMINATION						
\$# (endtim	endcyc	dtmin	endeng	endmas			
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*DAT	ABASE_T	PRINT						
\$#	dt	binary	lcur	ioopt				
1.	000000	0	0	1				
*DAT	ABASE_H	ISTORY_NODE						
\$#	nid1	nid2	nid3	nid4	nid5	nid6	nid7	nid8
	225	171	371	372	373	374	375	376
	377	378	379					
*DAT	ABASE_B	INARY_D3PLO	Т					
\$#	dt	lcdt	beam	npltc	psetid			
1.	000000	0	0	0	0			
*PAR	Т							
\$# t	itle							
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Ş#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
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¥ 11	0	750.00000	0	0.000	0			
*BOUI	NDARY C	ONVECTION S	ET					
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	2							
\$#	hlcid	hmult	tlcid	tmult	loc			
	0	750.00000	0	0.000	0			
*BOUI	NDARY_T	EMPERATURE_	SET					
\$#	nsid	lcid	cmult	loc				
	1	0 1	00.00000	0				
*INI	TIAL_TE	MPERATURE_S	ET					
\$#	nsid	temp	loc					
	1	100.00000	0					
*SET	_NODE_L	IST_TITLE						
fron	tt100							
\$#	sid	dal	da2	da3	da4			
.	1	0.000	0.000	0.000	0.000			
Ş#	nial	nid2	nid3	nid4	nid5	nido	nid7	nid8
	78	79	80	81	82	83	84	85
	86	87	88	89	90	91	92	93
	100	95 100	90 104	9/ 10E	98 10 <i>6</i>	99 107	100	101
	110	1U3	110	1U5 110	100 11 <i>1</i>	10/ 115	108 11 <i>6</i>	109 117
	110	110	1 2 0	101	1 2 2	100	104	105
	126	1 0 7	120	1 2 L	120	⊥∠3 121	⊥⊿4 120	120 120
	120	125	126	127	128	120	140	1 <u>4</u> 1
	142	142	144	145	146	147	149	140
	150	151	152	153	154	1 I /	10	0
*SET	SEGMEN	T TITLE	194	100	101	5	5	0
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\$#	sid	da1	<u>_</u>	da2	da3	da4	1			
	1	0.000)	0.000	0.000	0.000)			
\$#	nl	n2	2	n3	n4	al	L	a2	a3	a4
	18	19	9	434	164	0.000	0.0.	000	0.000	0.000
	370	226	5	85	94	0.000	0.	000	0.000	0.000
*SET	_SEGMENT	_TITLE								
back	t0									
\$#	sid	da1	L	da2	da3	da4	1			
	2	0.000)	0.000	0.000	0.000)			
\$#	nl	n2	2	n3	n4	al	L	a2	a3	a4
	2	11	L	33	28	0.000	0.	000	0.000	0.000
	77	13	3	12	27	0.000) 0.	000	0.000	0.000
*ELE	MENT SOI	JD								
\$#	eid	pid	n1	n2	n3	n4	n5	n6	n7	n8
* 11	1	1	2	28	33	11	163	213	443	289
	600	1	017	270	226	272	154	0.4	95	96
*NOD	5000 F	T	01/	570	220	212	134	71	05	50
с. 4	nid nid							t a	ra	
<i>ү</i> #	1110	(1	2000000	1 (2		IC	
	T	(0.000	1.	0000000	-1.0	000000	0	0	
	847	0.500	00000	0.	9000000	-0.1	L000000	0	0	
*END										

Notes:

1. A two-dimensional model simulation could be made using the plane stress (x-y plane) element formulation (elform=12 given on *SECTION_SHELL keyword) with a constant temperature through the thickness. Under the keyword *CONTROL_SHELL, the option tshell allows the user to choose between a constant temperature through the thickness and a 20 node brick formulation which allows heat conduction through the thickness.

24. 3D Thermal Load

Keywords:

```
*CONTROL_IMPLICIT_GENERAL
*CONTROL_IMPLICIT_SOLUTION
*LOAD_THERMAL_VARIABLE_NODE
*MAT_ELASTIC_PLASTIC_THERMAL
```

Description:

The solid cylinder tapering to a sphere geometry depicted below (Figure 24.1) is subjected to a prescribed temperature gradient. Two analyses, one with a coarse mesh 5 x 1 x 3 and one with a fine mesh 10 x 2 x 3 (Figure 24.2), are made. The model represents ¹/₄ of the total geometry. Symmetry conditions on the plane x-z and y-z are enforced. The faces parallel to the plane x-y are simply supported in the Z-direction.

The linear temperature loading (radial and axial direction) (Figure 24.3) is applied by means of temperature dependent material with thermal expansion coefficient and resulting thermal strain $\varepsilon = \alpha T$.

Determine the stress in the Z-direction at Node A (Node 10 for the coarse-mesh model and Node 16 for the fine-mesh model).



Figure 24.1 - Schematic of ¼ model and cross-section dimensions (all dimensions are in meters).



Figure 24.2 - Finite element models with selected node and element identified.



Figure 24.3 - Finite element models with temperature loading.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Static	Thermal	Linear	Linear	-	Implicit	1-Linear

Units:

kg, m, s, N, Pa, N-m, °C (kilogram, meter, second, Newton, Pascal, Newton-meter, degree Centigrade) - Joule (J) is a N-m, Watt (W) is a J/s

Dimensional Data:

See Figure 24.1 (all dimensions in meters).

Material Data:

Young's Modulus	$E = 2.10 \times 10^{11} Pa$
Poisson's Ratio	v = 0.3
Linear Expansion	$\alpha = 2.30 \times 10^{-4} \ m / m / ^{\circ} C$

Load:

Thermal

$$T(^{\circ}C) = \sqrt{x^2 + y^2} + z$$

Element Types:

Fully integrated S/R solid (elform=2)

Material Models:

*MAT_004 or *MAT_ELASTIC_PLASTIC_THERMAL

Results Comparison:

LS-DYNA global stress σ_{zz} at inner point on yz-symmetry plane (coarse mesh - Node 10, Element 3: fine mesh - Node 16, Element 5) is compared with *NAFEMS Background* to *Benchmarks*, Test LE11.

Reference Condition - Inner Point on Symmetry Plane	Coarse Mesh - Global Stress - σ_{zz} (MPa)	Fine Mesh - Global Stress - σ_{zz} (MPa)		
NAFEMS Benchmark Test LE11	-1.0500×10^{2}	-1.0500×10^{2}		
Element 3 (coarse) - Element 5 (fine) - (an averaged value)	-6.1849×10^{1}	-7.5890×10^{1}		
First in-plane integration point - Elem 3 (coarse) - Element 5 (fine)	-7.9107×10^{1}	-8.5217×10^{1}		
Node 10 (coarse) - Node 16 (fine)	-9.2071×10^{1}	-9.2102×10^{1}		

The global stress σ_{zz} results were generated from *DATABASE_ELOUT (*elout* file) and *DATABASE_EXTENT_BINARY (*eloutdet* file provides detailed element output at integration points and connectivity nodes) keyword entries.
By default, stresses/strains for solids are written to d3plot and *elout* in the global coordinate system. The *elout* file contains only the values at the element centroid (average of 8 integration points).

You can set intout=stress or intout=all (*DATABASE_EXTENT_BINARY) and have stresses output for all the integration points to a file called *eloutdet* (*DATABASE_ELOUT governs the output interval and *DATABASE_HISTORY_ SOLID governs which elements are output). Setting nodout=stress or nodout=all in *DATABASE_EXTENT_BINARY will write the extrapolated nodal stresses to *eloutdet*.

LS-DYNA stress and strain outputs correspond to integration point locations. Stress at a node is an artifact of the post-processor and represents an average of the surrounding integration point stresses (the value will likely be different with different postprocessors).

Both meshes are rather coarse which makes it difficult to capture the global stress σ_{zz} at the inner point along yz-symmetry plane. The extrapolated nodal stress results (also see Figures 24.4 and 24.5) provide the best comparative value (~12% difference for both meshes), primarily due to the nodal location. As expected, the fine mesh does a better job of capturing the overall contouring. The average global stress of the element provides the least acceptable comparative value results (~40% and ~25% differences), again not unexpected, while the nearest element integration point results provided significant improvement (~25% and ~15% differences) due to the larger integration sample (8 points as compared to 1) and nodal location.



Figure 24.4 - Coarse mesh contour plots of global stress σ_{zz} with average value given for Element 3. On the left is in-plane integration point contouring while on the right is extrapolated nodal stress contouring with specification values given at Node 10.



Figure 24.5 - Fine mesh contour plots of global stress σ_{zz} with average value given for Element 5. On the left is in-plane integration point contouring while on the right is extrapolated nodal stress contouring with specification values given at Node 16.

Input deck:

*KE	EYWORD							
*T]	TLE							
3D	Thermal	Load (coarse	e mesh)					
*CC	ONTROL_IM	PLICIT_GENER	AL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	1.000000	2	1	2			
*CC	ONTROL_IM	PLICIT_SOLU	FION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	lstol	abstol
	1	11	15	0.001000	0.010000	1.0e+10	0.900000	1.000000
\$#	dnorm	diverg	istif	nlprint	nlnorm	d3itcl	cpchk	
	2	1	_ 1	2	2	0	0	
\$#	arcctl	arcdir	arclen	arcmth	arcdmp	arcpsi	arcalf	arctim
	0	1	0.0	1	2	0.0	0.0	0.0
*C0	ONTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
]	1.000000	0	0.0	0.0	0.0			
*D#	ATABASE_E	XTENT_BINARY	<u> </u>			6.7		C 1
Ş#	neiph	neips	maxint	strilg	sigilg	epsilg	rtilg	engilg
àн	0		0	1	1	1	1 	1
Ş#	cmpilg	leverp	beamip	dcomp	shge	stssz	n3thdt	ialemat
\$#	nintsld	pkp_sen	sclp	hydro	msscl	therm	intout	nodout
	8		1.0				stress	stress
*DA	ATABASE_E	LOUT						
\$#	dt/cycl							
(0.100000							
*DF	ATABASE_H	ISTORY_SOLII	0					
\$#	eid1 3	eid2	eid3	eid4	ei5	eid6	eid7	eid8
*DA	ATABASE_B	INARY_D3PLO	Г					
\$#	dt/cycl							
1	L.000000							
*P#	ART							
\$#	title							
Par	rt	1 for Mat	-	1 and Ele	m Type	2		
\$#	pid	secid	mid	eosid	hgid	grav	adpopt	tmid
	- 1	1	1		_	-		
*SE	CTION_SO	LID						
\$#	secid	elform	aet					
	1	2	1					
*MZ	AT_ELASTI	C_PLASTIC_TH	HERMAL					
\$#	mid	ro						
	1	1.000000						
\$#	t1	t2	t3	t4	t5	tб	t7	t8

\$# 2.3 \$# 2.3 \$# \$# \$#	0.0 e1 100e+11 pr1 .300000 alpha1 300e-04 sigy1 0.0 etan1 0.0	1000.000 e2 2.100e+11 pr2 0.300000 alpha2 2.300e-04 sigy2 0.0 etan2 0.0	а	0.0 e3 0.0 pr3 0.0 llpha3 0.0 sigy3 0.0 etan3 0.0	0.0 e4 0.0 pr4 0.0 alpha4 0.0 sigy4 0.0 etan4 0.0	0.0 e5 0.0 pr5 0.0 alpha5 0.0 sigy5 0.0 etan5 0.0	0.0 e6 0.0 pr6 0.0 alpha6 0.0 sigy6 0.0 etan6 0.0	a	0.0 e7 0.0 pr7 0.0 lpha7 0.0 sigy7 0.0 etan7 0.0	0.0 e8 0.0 pr8 0.0 alpha8 0.0 sigy8 0.0 etan8 0.0
\$#	eid 1	pid 1	n1 1	n2 13	n3 16	n4 4	n5 2	n6 14	n7 17	n8 5
*NOI	15 DE	1	35	39	40	36	43	47	48	44
\$#	nid 1	1.0000	x 0000		у 0.000		z 0.000	tc 0	rc 0	
*९┲੶	48 5 NODE I	0.	000	1	.0000000	1.7	900000	0	0	
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\$#	sid	dal		da2	da3	da4	solver			
	1	0.0		0.0	0.0	0.0				
\$#	nidl	nid2		nid3	nid4	nid5	nid6		nid7	nid8
*SET	19 _NODE_I	JIST_TITLE		4	16					
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	43	47		46	42					
*SEI	r_node_i	LIST_TITLE								
xsyr	nm						_			
\$#	sid	dal		da2	da3	da4	solver			
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ysyt	nm —									
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	4	0.0		0.0	0.0	0.0				
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^SEI	L_NODE_I	TST_LTTTE								
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	1	13		41	45					
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\$#	nid1	nid2		nid3	nid4	nid5	nid6		nid7	nid8
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*BOU	JNDARY_S	SPC_SET								
\$#n:	id/nsid	cid		dofx	dofy	dofz	dofrx		dofry	dofrz
	1	0		0	0	1				
*BOU	JNDARY_S	SPC_SET		dofr	dofr	dofe	dofm		dofar	dofwa
φ#11	2 v / 115 10	CIG		U U	uory U	1	UOLIX	(ασττλ	uorrz
*BOI	JNDARY S	SPC_SET		5	0	±				
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	3	0		1	0	0			-	
*BOU	JNDARY_S	SPC_SET								
\$#n:	id/nsid	cid		dofx	dofy	dofz	dofrx	(dofry	dofrz
+	4	0		0	1	0				
	JNDARY_S	SPC_SET		dofy	dofr	dofa	dofre		dofry	dofra
Υ#111	La / IID TU	CIU		UULA	uory	uorz	UOLIX		GOTTÀ	GOLLZ

	5	0	0	1	1			
*BOUNI	DARY_S	PC_SET						
\$#nid/	/nsid	cid	dofx	dofy	dofz	dofrx	dofry	dofrz
	б	0	1	0	1			
*LOAD_	_THERM	AL_VARIABLE_	NODE					
\$#	nid	ts	tb	lcid				
	1	1.000000	0.0	1				
	48	2.790000	0.0	1				
*DEFIN	NE_CUR	VE						
\$#	lcid	sdir	sfa	sfo	offa	offo	dattyp	
	1	0	0.0	0.0	0.0	0.0		
\$#		al		01				
		0.0	1.0	0000000				
	1	.00000000	1.0	0000000				
*END								

Notes:

- 1. The fully integrated solid elements are formulated for nonlinear analysis. Although this analysis is linear, it could have been solved with a nonlinear solution method (nsolvr=2). This was done for this simulation and it was found to yield results with some differences (less than 0.01% for coarse mesh and ~4% for fine mesh) from the linear analysis (nsolvr=1). It is not understood why the fine mesh offers this difference.
- 2. Fully-integrated solid formulations have 8 in-volume integration points for each element. For these formulations, the 8 values of each stress component are averaged at the element centroid before being written to *elout*.
- 3. If setting nintsld=8 on *DATABASE_EXTENT_BINARY, LS-DYNA will write stresses at all integration points for solid elements (also given in *eloutdet*) to the d3plot file. When this option is set, LS-PrePost applies the stress values to the nodes from the closest integration point and after that, the average value from the contributions are computed and presented in the stress fringe plot.
- 4. A command line option (extrapolate 1) is added to LS-PrePost, which will linearly extrapolate the values from integration points to the nodes (the extrapolated nodal stresses are also given in *eloutdet*).
- 5. For elastic bending, two integrations points through the thickness is the minimum number. For plastic bending, three integrations points through the thickness is the minimum.

25. Cooling of a Billet via Radiation

Keywords:

```
*CONTROL_SOLUTION
*CONTROL_THERMAL_SOLVER
*CONTROL_THERMAL_NONLINEAR
*CONTROL_THERMAL_TIMESTEP
*BOUNDARY_TEMPERATURE_SET
*BOUNDARY_RADIATION_SET
*MAT_THERMAL_ISOTROPIC
```

Description:

A billet $(L_z = 4.00 \ ft$ in height) of rectangular cross-section $(L_x = 2.00 \ ft$ by $L_7 = 2.00 \ ft$) shown in Figure 25.1 is initially at temperature $T_0 = 2000 \ R$ loses heat by radiation (transient) from all its surfaces to its surroundings at a temperature of $T_e = 530 \ R$. There is zero internal heat generation.

Determine the temperature of the billet (e.g. Node 625) after 3.7 hours $(1.3320 \times 10^4 \text{ sec})$.

The bar is meshed with 432 elements: 12 elements along the height and 36 elements in the cross section.



Figure 25.1 - Finite element model with selected nodes and dimensions identified.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Thermal Transient	Thermal	Linear	Linear	-	Thermal Nonlinear	3-Diagonal scaled conjugate gradient

Units:

*lbf-s*²/*ft, ft, s, lbf, psf, lbf-ft (slug, foot, second, pound force, pound force/foot*², *pound force-foot*) - *Thermal Energy* is a *Btu, Power* is a *Btu/s,* 1° $\Delta F = 1^{\circ} \Delta R$

Dimensional Data:

 $L_x = 2.00 \ ft$, $L_y = 2.00 \ ft$, $L_z = 2.00 \ ft$

Material Data:

Mass Density	$\rho = 4.875 \times 10^2 \ lbf - s^2 \ / \ ft^4$
Heat Capacity	$C_p = 1.100 \times 10^{-1} Btu / lbf - R$
Thermal Conductivity	$k = 1.000 \times 10^4 Btu / ft - R$ (arbitrary value)
Emissivity	$\varepsilon = 1.000$
Stefan-Boltzman	$\sigma = 4.750 \times 10^{-13} Btu / s - ft^2 - R^4$

Load:

Thermal (billet)	$T_0 = 2000.0 \ ^\circ R$ (constant)
Thermal (outside)	$T_e = 530.0$ °R (surroundings)

Element Types:

Fully integrated S/R solid (elform=2)

Material Models:

*MAT_T01 or *MAT_THERMAL_ISOTROPIC

Results Comparison:

LS-DYNA temperature of the billet at selected point x=2.000 ft, y=2.010 ft, z=0.000 ft (Node 625) is compared with R. Siegal and J.R. Howell studies in *Thermal Radiation Heat Transfer*, 1981, pg. 229.

Reference Condition - Billet	Temperature ($^{\circ}R$) at t=13320 sec
Siegal and Howell [1981]	1.0000×10^{3}
Node 625	1.0080×10^{3}

The fully integrated selectively reduced solid element (elform=2) model provides a reasonable temperature comparison for this mesh (less that 1% difference).

With the given simple geometry plus the initial temperature and radiation heat losses being space invariant, the temperature is uniform throughout the mesh.

The temperature history of Node 625 used in the comparison is shown in Figure 25.2.



Figure 25.2 - Temperature history for Node 625.

Input deck:

*KEYWORD *TITLE Cooling of a Billet Via Radiation *CONTROL_SOLUTION \$# soln nlq isnan 1 0 0 *CONTROL_THERMAL_SOLVER lcint 0 CONTROL_INERMAL_SOLVER \$# atype ptype solver cgtol gpt eqheat fwork sbc 1 2 3 1.00e-06 1 1.000000 1.0000004.7500e-13 *CONTROL THERMAL NONLINEAR *CONTROL_THERMAL_NONLINEAR \$# refmax tol dcp 20 1.000e-06 0.500000 *CONTROL_THERMAL_TIMESTEP \$# ts tip its tmin tmax dtemp tscp 1 0.500000 0.100000 0.100000 100.0000 1.000000 0.500000 *CONTROL_TERMINATION dtmin \$# endtim endcyc
1.3320e+04 0
*DATABASE_TPRINT endeng endmas 0.0 0.0 0.0 \$# dt binary 10.00000 0 ioopt lcur 0 1 *DATABASE HISTORY NODE nid4 nid5 nid6 nid7 nid8 \$# nid1 nid2 1 13 nid3 79 265 *DATABASE_BINARY_D3PLOT beam npltc 0 0 \$# dt lcdt 100.0000 0 psetid 0 0 *PART \$# title Part 1 for TMat 1 and Elem Type eosid hgid 2 mid eosid hgid grav 0 0 0 0 0 \$# pid secid 1 1 *SECTION_SOLID adpopt tmid 0 0 1 \$# secid elform 1 2 aet 1 *MAT_THERMAL_ISOTROPIC \$# tmid tro 1 487.5000 \$# hc tc tgrlc tgmult 0.0 0.0 0.11000 1.000e+04 *BOUNDARY_RADIATION_0000 \$# ssid type 1 1 \$# flcid fmult tilcid timult loc 04.7500e-13 0 530.0000 0 \$# nsid temp 1 2000.0000 loc 0 *SET_NODE_LIST_TITLE all sid dal da2 1 0.0 0.0 nid1 nid2 nid3 1 2 3 da3 da4 \$# 0.0 0.0 nid6 \$# nid1 nid4 nid5 nid7 nid8 4 1 5 2 3 6 7 8 633 634 635 636 637 *SET_SEGMENT_TITLE rad_surf \$# sid da1 da2 da3 da4 1 0.0 0.0 0.0 0.0 \$# n1 n2 n3 n4 a1 a2 a3 a4 2 0.0 0.0 0.0 92 93 1 0.0 637 545 546 636 0.0 0.0 0.0 0.0 *ELEMENT_SOLID n4 n5 n6 14 2 93 \$# eid pid n1 n2 1 1 1 92 n3 n7 n8 106 105 15

	432	1	532	623	636	545	533	624	637	546
*NOI	DE									
\$#	nid		x		У		z	tc	rc	
	1		0.0		0.0		0.0			
	637	2.000	00000	2.00	000000	4.00	000000			
*ENI	2									

Notes:

1. The problem must be flagged as nonlinear if any boundary condition parameter is a function of temperature *T*. This includes a linear (i.e., straight line) relationship. Iterations are needed to obtain the correct solution. Radiation (*q*) is a T^4 temperature boundary condition (usually *F*=1):

$$q = \sigma \varepsilon F (T_{surface}^4 - T_{\infty}^4)$$

- The *CONTROL_THERMAL_NONLINEAR keyword is optional. For example, the default values for remax (maximum number of iterations allowed per time step), tol (temperature convergence tolerance), and dcp (divergence control tolerance) will be used, if the nonlinear keyword is omitted, with ptype>0 on *CONTROL_ THERMAL_SOLUTION keyword.
- 3. This study could have been performed using a single element with the following modifications to the above input deck:

*SEI	_NODE_LI	IST_TITL	Ε							
all										
	1	0.	0	0.0	0.0	0.0				
	1		2	3	4	5	6		7	8
*SEI	_SEGMENT	I_TITLE								
	ext_surf									
\$#	sid	da	1	da2	da3	da4				
	1	0.	0	0.0	0.0	0.0				
\$#	nl	n	2	n3	n4	al	a2		a3	a4
	2		3	7	6	0.0	0.0		0.0	0.0
	7		8	5	6	0.0	0.0		0.0	0.0
	4		8	7	3	0.0	0.0		0.0	0.0
	2		6	5	1	0.0	0.0		0.0	0.0
	4		1	5	8	0.0	0.0		0.0	0.0
	1		4	3	2	0.0	0.0		0.0	0.0
*ELE	MENT_SOI	LID								
\$#	eid	pid	n1	n2	n3	n4	n5	n6	n7	n8
	1	1	1	2	3	4	5	6	7	8
*NOI	ЭE									
	1		0.0		0.0		0.0			
	2	2.000	00000		0.0		0.0			
	3	2.000	00000	2.00	000000		0.0			
	4		0.0	2.00	000000		0.0			
	5		0.0		0.0	4.0000	0000			
	б	2.000	00000		0.0	4.0000	0000			
	7	2.000	00000	2.00	000000	4.0000	0000			
	8		0.0	2.00	000000	4.0000	0000			

This single element representation provides the identical temperature result at t=3.7 hours $(1.3320 \times 10^4 \text{ sec})$ as the 432 element mesh.

26. Pipe Whip

Keywords:

```
*CONTROL_CONTACT
*CONTACT_AUTOMATIC_SURFACE_TO_SURFACE_TITLE
*MAT_PLASTIC_KINEMATIC
*MAT_RIGID
*INITIAL_VELOCITY_GENERATION
*CONSTRAINED_EXTRA_NODES_SET
```

Description:

This problem illustrates the capabilities of LS-DYNA in a high speed, large deformation event with complex contact conditions, e.g. a pipe-on-pipe impact.

The pipes are modeled using fully integrated shell elements.

The impacted pipe is fully restrained, translationally and rotationally, at both ends (x = 0 and $x = L_1$: $U_x = U_y = U_z = R_x = R_y = R_z = 0$), while the impacting pipe is rotating at an initial angular speed of 75 *rad/s* about a fixed point at one end (Figures 26.1 and 26.2).

The pipe material is elastic-perfectly plastic, and the material model *MAT_PLASTIC_ KINEMATIC with zero tangent modulus is appropriate.

The initial rotational velocity is imposed through the keyword *INITIAL_VELOCITY_ GENERATION. A rigid, rotational end joint is defined using the pipe's end ring of nodes which are made rigid using the *CONSTRAINED_EXTRA_NODES_SET and *MAT_RIGID keywords.

The contact is ***CONTACT_AUTOMATIC_SURFACE_TO_SURFACE**. This contact has the following characteristics:

- it is a two-way contact, in that user-specified slave nodes are checked for penetration of the master segments and then a second time, to check the master-side nodes for penetration through the slave segments,
- the treatment is thus symmetric and the definition of the slave surface and master surface is arbitrary,
- AUTOMATIC contacts check for penetration on either side of a shell element,
- this is a recommended contact type in large deformation application, e.g. in crash simulations, since the orientation of parts relative to each other cannot always be anticipated.

Shell thickness is considered with option shlthk=1. The soft=2 option (segment based contact) is used to distribute the contact forces over the elements.





Figure 26.1 – Finite element model with boundary condition nodes (marked with \Box 's). There are 100 elements axially and 40 elements circumferentially.



Figure 26.2 – Half-symmetry finite element model of a 50 in pipe with initial angular velocity (75 *rad/s*).

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Velocity	Non- linear	Linear	3D	Explicit	

Units:

 $lbf-s^2/in$, in, s, lbf, psi, lbf-in (blob, inch, second, pound force, pound force/inch², pound force-inch)

Dimensional Data:

 $L_1 = L_2 = 5.000 \times 10^2$ in, $t = 4.320 \times 10^{-1}$ in

Material Data:

Mass Density	$\rho = 7.324 \times 10^{-4} \ lbf - s^2 \ / \ in^4$
Young's Modulus	$E = 3.000 \times 10^7 \ lbf \ / \ in^2$
Poisson's Ratio	v = 0.3
Yield Stress	$\sigma_{y} = 4.500 \times 10^{4} \ lbf \ / \ in^{2}$
Tangent Modulus	$E_t = 0.000 \times 10^0 \ lbf \ / \ in^2$

Load:

Velocity $\omega = 7.500 \times 10^1 \ rad \ / s$

Element Types:

Fully integrated shell (elform=16)

Material Models:

*MAT_003 or *MAT_PLASTIC_KINEMATIC

*MAT_020 or *MAT_RIGID

Results Comparison:

The results for deformed shapes taken from R.M. Ferencz studies on *Element-by-Element Preconditioning Techniques for Large-Scale, Vectorized Finite Element Analysis in Nonlinear Solid and Structural Mechanic,* March, 1989 (pg. 142) are reproduced here in Figure 26.3. The LS-DYNA results for deformed shapes at selected times in the simulation (Figures 26.4a and 26.4b) are in good agreement.



Figure 26.3 – Deformed shapes (Ferencz [1989]) at 0.0, 2.5, 5.0, and 10.0 ms.



Figure 26.4a – Half-symmetry deformed shapes at 0.0025, 0.0050, 0.0100, and 0.0150 sec (hidden line view).



Figure 26.4b – Half-symmetry deformed shapes at 0.0025, 0.0050, 0.0100, and 0.0150 sec (shaded view).

The histories of the kinetic energy, internal energy, sliding energy, and the total energy are given in Figure 26.5.

Nearly all of the initial kinetic energy has been converted into plastic deformation (internal energy) due to the pipe deformation.

There is a small amount of energy dissipated in the contact (sliding energy) between the pipes, which, when included in the output computation, makes for an energy balance.



Figure 26.5 – Histories of the kinetic energy, internal energy, sliding energy, and the total energy.

Input Deck:

YWORD							
TLE							
e Whip							
NTROL_TE	RMINATION						
endtim	endcyc	dtmin	endeng	endmas			
.015000	0	0.0	0.0	0.0			
NTROL_TI	MESTEP						
dtinit	tssfac	isdo	tslimt	dt2ms	lctm	erode	ms1st
0.0	0.900000						
dt2msf	dt2mslc	imscl					
0.0	0	0					
NTROL_CO	NTACT						
slsfac	rwpnal	islchk	shlthk	penopt	thkchg	orien	enmass
.000000	0.0	2	2	0	0	1	
usrstr	usrfrc	nsbcs	interm	xpene	ssthk	ecdt	tiedprj
	YWORD TLE pe Whip NTROL_TE endtim 0.015000 NTROL_TI dtinit 0.0 dt2msf 0.0 NTROL_CC slsfac 000000 usrstr	YWORD TILE De Whip NTROL_TERMINATION endtim endcyc 0.015000 0 NTROL_TIMESTEP dtinit tssfac 0.0 0.900000 dt2msf dt2mslc 0.0 0 NTROL_CONTACT slsfac rwpnal .000000 0.0 usrstr usrfrc	YWORD TILE De Whip NTROL_TERMINATION endtim endcyc dtmin 0.015000 0 0.0 NTROL_TIMESTEP dtinit tssfac isdo 0.0 0.900000 dt2msf dt2mslc imscl 0.0 0 0 0 NTROL_CONTACT slsfac rwpnal islchk 000000 0.0 2 usrstr usrfrc nsbcs	YWORD TILE De Whip NTROL_TERMINATION endtim endcyc dtmin endeng 0.015000 0 0.0 0.0 NTROL_TIMESTEP dtinit tssfac isdo tslimt 0.0 0.900000 dt2msf dt2mslc imscl 0.0 0 0 0 NTROL_CONTACT slsfac rwpnal islchk shlthk 000000 0.0 2 2 usrstr usrfrc nsbcs interm	YWORD TILE be Whip NTROL_TERMINATION endtim endcyc dtmin endeng endmas 0.015000 0 0.0 0.0 0.0 NTROL_TIMESTEP dtinit tssfac isdo tslimt dt2ms 0.0 0.900000 dt2msf dt2mslc imscl 0.0 0 0 0 NTROL_CONTACT slsfac rwpnal islchk shlthk penopt .000000 0.0 2 2 0 usrstr usrfrc nsbcs interm xpene	YWORD TLE De Whip NTROL_TERMINATION endtim endcyc dtmin endeng endmas 0.015000 0 0.0 0.0 0.0 NTROL_TIMESTEP dtinit tssfac isdo tslimt dt2ms lctm 0.0 0.900000 dt2msf dt2mslc imscl 0.0 0 0 NTROL_CONTACT slsfac rwpnal islchk shlthk penopt thkchg 000000 0.0 2 2 0 0 usrstr usrfrc nsbcs interm xpene ssthk	YWORD TILE De Whip NTROL_TERMINATION endtim endcyc dtmin endeng endmas 0.015000 0 0.0 0.0 0.0 NTROL_TIMESTEP dtinit tssfac isdo tslimt dt2ms lctm erode 0.0 0.900000 dt2msf dt2mslc imscl 0.0 0 0 NTROL_CONTACT slsfac rwpnal islchk shlthk penopt thkchg orien 000000 0.0 2 2 0 0 1 usrstr usrfrc nsbcs interm xpene ssthk ecdt

0 0 0 0 4.000000 \$# sfric dfric edc vfc th th_sf pen_sf 0.0 0.0 0.0 0.0 0.0 0.0 0.0 0.0 \$# ignore frceng skiprwg 0 0 0 0 *CONTROL_ENERGY \$# bccp slnten rylen 2 ° \$# hgen rwen 2 2 slnten *DATABASE_GLSTAT \$# dt binary 1.0000e-05 1 *DATABASE_MATSUM \$# dt binary 1.0000e-05 1 \$# dt binary *DATABASE_SLEOUT 1.0000e-05 *DATABASE BINARY D3PLOT \$# dt/cycl lcdt/nr beam npltc psetid 2.5000e-04 *PART \$# title material type # 3 (Kinematic/Isotropic Elastic-Plastic) \$# pid secid mid eosid hgid grav 1 1 1 0 1 adpopt tmid *SECTION_SHELL \$# secid elform shrf nip propt 1 16 0.83333 5.0 1 \$# t1 t2 t3 t4 nloc qr/irid icomp setyp 0.0 marea 0 1 marea 0.432000 0.432000 0.432000 0.432000 0.0 0 *MAT_PLASTIC_KINEMATIC \$# mid ro e pr sigy е beta etan 1 7.324e-04 3.000e+07 0.300000 4.500e+04 0.0 0.0 \$# src srp fs vp 0.0 0.0 0.0 0.0 *HOURGLASS
 \$#
 hgid
 ihq
 qm
 ibq
 q1
 q2

 1
 0
 0.0
 0
 0.0
 0.0
 qb -ی^ر 0.0 qw 0.0 *PART \$# title material type # 3 (Kinematic/Isotropic Elastic-Plastic) \$# pid secid mid eosid hgid grav 2 2 2 2 0 2 0 *SECTION SHELL adpopt tmid 0 1 *SECTION_SHELL \$# secid elform shrf nip propt qr/irid 2 16 0.83333 5.0 1 0.0 \$# t1 t2 t3 t4 nloc marea 0.432000 0.432000 0.432000 0.432000 0 0.0 icomp setyp 0 1 *MAT_PLASTIC_KINEMATIC pr sigy etan 000 4.500e+04 0.0 beta \$# mid ro е 2 7.324e-04 3.000e+07 0.300000 4.500e+04 0.0 \$# src srp fs 0.0 0.0 0.0 vp 0 0 0.0 *HOURGLASS \$# hgid ihq qm ibq 2 0 0.0 0 ي∠ 0.0 dp 0.0 q1 qw ے۔م 2 *PART 0.0 0.0 0 \$# title material type # 20 (Rigid) \$# pid secid mid eosid 99 99 99 99 hgid grav adpopt tmid icomp setyp 0 *MAT_RIGID *# mid ro

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 7.324e-04
 3.000e+07
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 \$#
 cmo
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 1.000000
 7
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 m alias

\$#lcc	or al	a2	a3	v1	v2	v3			
+0010	0.0	0.0	0.0	0.0	0.0	0.0			
s#	ntAINEI pid	nsid	DES_SEI						
* "	99	99							
*SET_	NODE_LI	IST_TITLE							
rigid	l ring d	of nodes							
\$#	sid	dal	da2	da3	da4	solver			
¢#	99 nid1	0.000	0.000	0.000	0.000	AECH nide		nid7	nido
φ#	8041	8042	8043	8044	8045	8046		8047	8048
	8049	8050	8051	8052	8053	8054		8055	8056
	8057	8058	8059	8060	8061	8062		8063	8064
	8065	8066	8067	8068	8069	8070		8071	8072
	8073	8074	8075	8076	8077	8078		8079	8080
*INIT	'IAL_VEI	LOCITY_GEN	IERATION						تد ند ند
ŞĦ	1a 2	styp	omega 75 000		vy 0 0	vz 0 0	1	vatn	1010
\$#	xc	VC	75.000	0.0 nx	nv	0.0 nz	n	hase	irigid
25.0	00000 5	50.000000	6.725000	1.000000	0.0	0.0	P	0	0
*CONT	ACT_AUT	TOMATIC_SU	JRFACE_TO_S	SURFACE_TIT	LE				
\$#	cid								title
	1								
\$#	ssid	msid	sstyp	mstyp	sboxid	mboxid		spr	mpr
¢#	fa	∠ fd	د م5	3	U Trđa	Donahk		U ht	ט + ה
φ#	0 0	0.0	0.0	0 0	0.0	pencia		0 0	0 0
\$#	sfs	sfm	sst	mst	sfst	sfmt		fsf	vsf
	0.0	0.0	0.0	0.0	0.0	0.0		0.0	0.0
\$#	soft	sofscl	lcidab	maxpar	sbopt	depth	b	sort	frcfrq
	2	0.100000	0	1.025	0.0	2	_	10	1
\$# p	enmax	thkopt	shlthk	snlog	isym	i2d3d	sl	dthk	sldstf
¢#	0.0	ignoro	T	0	0	0		0.0	0.0
γπ	19ap 2	191101.6							
*ELEM	ient_she	CLL							
\$#	eid	pid	nl r	n2 n3	n4	n5	nб	n7	n8
	1	1	1	2 42	41	0	0	0	0
0		0	0.4.0 0.0/	0.041	0000	0	0	0	0
8 *NODE	,000	2 8	8040 800	01 8041	8080	0	0	0	0
S#	nid		x	v		7	tc	rc	
ΨII	1	0.	000	28.096500		0.000	0	0	
8	080	28.058	376	50.000000	7.2	209399	0	0	
*SET_	NODE_LI	IST	1.0	1.0		-			
\$#	sid	dal	da2	da3	da4	solver			
¢#	⊥ nid1	0.000 nid2	0.000 nid3	0.000 nid4	0.000	ALCH nide		nid7	nide
ŶΠ	1	2	3	4	5	6		7	8
	9	10	11	12	13	14		15	16
	17	18	19	20	21	22		23	24
	25	26	27	28	29	30		31	32
+	33	34	35	36	37	38		39	40
*SET_	NODE_LI	dol	dag	da2	dal	golyon			
ү #	2	0 000	0 000	0 000	0 0001	NECH			
\$#	nidl	nid2	nid3	nid4	nid5	nid6		nid7	nid8
	4001	4002	4003	4004	4005	4006		4007	4008
	4009	4010	4011	4012	4013	4014		4015	4016
	4017	4018	4019	4020	4021	4022		4023	4024
	4025	4026	4027	4028	4029	4030		4031	4032
*⊡∩™	4033 ייי עסגרו	4034 00 957	4035	4036	4037	4038		4039	4040
\$#nid	l/nsid	hin	dofv	dofv	dof-	dofrv	Ь	ofrv	dofra
	1	0	1	1	1	1	u	1	1
*BOUN	IDARY_SI	PC_SET							
\$#nid	l/nsid	cid	dofx	dofy	dofz	dofrx	d	ofry	dofrz
	2	0	1	1	1	1		1	1
^ END									

Notes:

- 1. The general contact *CONTACT_AUTOMATIC_SINGLE_SURFACE could also have been used. This type of contact has the following characteristics:
 - a single contact surface is created for all the parts included in the contact,
 - self contact is considered,
 - it is robust, reliable and accurate, making it the ideal choice for crashworthiness and impact applications.

By default, if ssid (slave segment id) is zero or blank, all part IDs are included in the contact. A *PART_SET entry can be used to reduce the size of the part list.

- 2. The most common contact-related output file, *rcforc*, is produced by including a *DATABASE_RCFORC keyword in the input deck. *rcforc* is an ASCII file containing resultant contact forces for the slave and master sides of each contact interface. The forces are provided in the global coordinate system. Note that *rcforc* data is not provided for single surface contacts as all the contact forces from this contact type come from the slave side (as there is no master side) and thus the net contact forces are zero. To obtain *rcforc* data when single surface contacts are used, one or more *force transducers* should be added via the *CONTACT_FORCE_TRANSDUCER_PENALTY keyword. A force transducer simply measures contact forces produced by other contact interfaces defined in the model.
- 3. By including a *DATABASE_SLEOUT keyword, individual contact interface energies are written to the ASCII output file *sleout*. The global contact energy is written to the ASCII output file *glstat*.

27. Copper Bar Impacting a Rigid Wall

Keywords:

```
*CONTROL_ALE
*CONTROL_CONTACT
*CONTACT_AUTOMATIC_SURFACE_TO_SURFACE
*RIGIDWALL_PLANAR
*INITIAL_VELOCITY_GENERATION
*SECTION_SOLID
*HOURGLASS
```

Description:

This problem is known in the literature as "Taylor Bar Impact Test" and is used to assess material properties (plastic flow) under dynamic conditions. A deformable copper bar impacts a rigid wall at high speed. The deformed length (shortening), spread (widening), and maximum effective plastic strain (ε^{p}) of the bar is determined.

The contact of the deformable body and the rigid wall can be modeled in one of the following ways:

- rigid wall (*RIGIDWALL_PLANAR), which provides an easy way to treat contact between a rigid-flat surface and the nodes of a deformable body,
- using geometric entities (*CONTACT_ENTITY),
- using a wall modeled with rigid shell elements and a *CONTACT_AUTOMATIC_ SURFACE_TO_SURFACE.

Two of these methods are demonstrated for rigid wall contact, (1) the rigid wall is modeled with rigid shell elements (penalty method) and (2) the wall is modeled as a planar rigid boundary. The latter uses a constraint method which represents a perfectly plastic impact since, once penetration into the rigid wall is detected, the acceleration and velocity of the nodes are set to zero. No friction is included.

The material is elastic-plastic with constant tangent stiffness and the material model *MAT_PLASTIC_KINEMATIC is used.

For comparison, different solid hexahedron (elform=1,2,-2,-1) and tetrahedron (elform=10,13) elements, are used to model the bar as shown in Figure 27.1. Hourglass control is used with the default value (ihq=2 - Flanagan-Belytschko viscous form with coefficient qm=0.10) for the one-point quadrature element formulations.

Traditional Lagrangian approaches, for large deformations, often result in highly distorted meshes for the elements close to the impacted region, leading to loss of accuracy and decreasing the critical time step for the simulation. Therefore, a simple Arbitrary Lagrangian-Eulerian (ALE) formulation (elform=5) is also presented. A mix of simple average smoothing and volume weighted smoothing is used for the interior nodal positioning.

The (elform=5) formulation is single material ALE with Lagrange outer boundary node treatment and mesh smoothing effective for only moderate deformation. Thus application of this formulation is limited to mostly academic problems, since there are not many practical applications for use of this feature. In fact, this "Taylor Bar Impact Test" may be the only known practical application.



Figure 27.1 – Finite element models of the impacting bar using hexahedron and tetrahedron elements (shell elements were used to model the impacted rigid plate). Each of these solid element meshes has 36 elements axially. There are 288 and 1440 elements per row for the hexahedron and tetrahedron models, respectively.

Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Velocity	Non- linear	Non- linear	3D	Explicit	Lagrangian and ALE

Units:

g, mm, ms, N, MPa, N-mm (gram, millimeter, millisecond, Newton, MegaPascal, Newtonmillimeter)

Dimensional Data:

 $L = 3.240 \times 10^{1} mm$, $d = 6.400 \times 10^{0} mm$

Material Data:

Mass Density	$\rho = 8.930 \times 10^{-3} g / mm^3$
Young's Modulus	$E = 1.170 \times 10^5 MPa$
Poisson's Ratio	v = 0.35
Yield Stress	$\sigma_y = 4.000 \times 10^2 MPa$
Tangent Modulus	$E_t = 1.000 \times 10^2 MPa$

Load:

Velocity	$V_{2} = 2.270 \times 10^{2} mm/$	ms
verocity	$V_{\pi} = 2.270 \times 10^{-10}$ mm/	110

Element Types:

Constant stress solid (elform=1) Fully integrated S/R solid (elform=2) Fully integrated S/R solid - for poor aspect ratio (acc) - (elform=-2) Fully integrated S/R solid - for poor aspect ratio (eff) - (elform=-1) 1 point tetrahedron (elform=10) 1 point nodal pressure tetrahedron (elform=13) 1 point ALE (elform=5)

Material Models:

*MAT_003 or *MAT_PLASTIC_KINEMATIC

*MAT_020 or *MAT_RIGID

Results Comparison:

The results for deformed shapes at 0.0, 5, 20, and 80 ms, taken from R.M. Ferencz studies on *Element-by-Element Preconditioning Techniques for Large-Scale, Vectorized Finite Element Analysis in Nonlinear Solid and Structural Mechanics*, March, 1989 (pg. 86), are reproduced here in Figure 27.2. Ferencz [1989] used NIKE3D and its implicit dynamics solver. The LS-DYNA results for deformed shapes at 80.0 ms (Figures 27.3a to 27.3g) using penalty method contact and rigid boundary contact (Figures 27.7a to 27.7g) are in good agreement. The maximum effective plastic strain ($\varepsilon^{p} = 2.248$) given by Ferencz [1989] differs significantly from the non-ALE results of LS-DYNA

($\varepsilon^{p} \cong 2.9 \text{ to } 3.9$), even though both use a Lagrangian approach; these values are taken from the highly distorted elements in the vicinity of the rigid wall.



Figure 27.2 – Deformed shapes (Ferencz [1989]) at 0.0, 5, 20, and 80 ms.

Penalty Method (Rigid Mesh Material)	Shortening (mm)	Widening (mm)	Max. plastic strain (ε^p)	Normalized CPU Time
Constant stress solid (elform=1)	10.928	8.125	3.523	1.40
Fully integrated S/R solid (elform=2)	10.976	8.395	3.671	5.20
Fully integrated S/R solid (elform=-2)	10.972	8.404	3.663	19.40
Fully integrated S/R solid (elform=-1)	10.976	8.418	3.700	6.40
1 point tetrahedron (elform=10)	11.088	7.537	2.944	3.00
1 point nodal pressure tetrahedron (elform=13)	11.017	8.533	3.924	10.50
1 point ALE (elform=5)	10.892	7.856	2.272	3.70

The above displacement and effective plastic strains results were obtained from the d3plot contour plots at 80.0 ms which were generated by the *DATABASE_BINARY_D3PLOT keyword.

Normalized CPU times shown in the above Penalty Method results table were normalized using the minimum value (the smallest value for all simulations - other contact type CPU times are to follow).

Large effective plastic strains develop at the impact end of the rod due to the severe local mesh distortion, also resulting in reduced accuracy.

For these simulations, a wide range of CPU times were associated with the different element formulations. The CPU time is controlled by the number of element operations required for that particular formulation, the complexity of the contact-impact approach and the element stable time step.

The one-point quadrature (low order) constant stress solid (elform=1) element formulation (the LS-DYNA default), the higher order, fully integrated selectively reduced solid (elform=2), and the higher order, fully integrated S/R solid (both so-called efficient and accurate formulation choices) intended to address poor aspect ratios (elform=-1 and -2, respectively), provide roughly the same dimensional changes and maximum effective plastic strain.

The one-point quadrature (low order) tetrahedron (elform=10) element formulation provides comparatively stiffer dimensional changes and maximum effective plastic strain than the constant stress solid and fully integrated element formulations. This is probably due to this element formulation being prone to volumetric locking (overly stiff behavior) in incompressible regimes, e.g., as in plasticity.

The one-point quadrature (low order) nodal pressure tetrahedron (elform=13) element formulation provides a less stiff, dimensional changes and maximum effective plastic strain comparison to that of the constant stress solid and fully integrated element formulations. This element formulation has no volumetric locking under plastic incompressible conditions.

The one point ALE (elform=5) element formulation provides similar dimensional changes to other element formulations. With its nodal smoothing capability controlling the aspect ratio of the elements, mesh distortion is reduced, yet a smaller maximum effective plastic strain ($\varepsilon^p = 2.272$) is achieved compared to the Lagrangian elements ($\varepsilon^p \cong 2.9 \text{ to } 3.9$). An explanation for these results is the moderate deformation limitation for the one point ALE formulation.

The LS-DYNA results for deformed shapes at 80.0 ms using penalty method contact with effective plastic strain contouring are given Figures 27.3a to 27.3g.



Figure 27.3a – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=1).



Figure 27.3b – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=2).



Figure 27.3c – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=-2).



Figure 27.3d – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=-1).



Figure 27.3e – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=10).



Figure 27.3f – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=13).



Figure 27.3g – Quarter-symmetry deformed shape (penalty method) with effective plastic strain contouring at 80 ms (elform=5).

The half-symmetry deformed shape (penalty method contact), which illustrates the different element deformation for elform=1 and elform=5 at 80 ms, is given in Figure 27.4.





The histories of the kinetic energy, internal energy, hourglass energy, sliding energy, and the total energy for (elform=1) of penalty method impact are given in Figure 27.5, while the histories of the stable time step increment for all elforms (1,2,-2,-1,10,13,5) associated with the penalty method impact are given in Figure 27.6.



Figure 27.5 – Histories of the kinetic energy, internal energy, hourglass energy, sliding energy, and the total energy for (elform=1) of penalty method impact.



Figure 27.6 – Histories of the stable time step increment for all elforms (1,2,-2,-1,10,13,5) associated with the penalty method impact.

Planar Rigid Boundary	Shortening (mm)	Widening (mm)	Max. plastic strain (\mathcal{E}^p)	Normalized CPU Time
Constant stress solid (elform=1)	10.897	7.889	3.243	1.00
Fully integrated S/R solid (elform=2)	10.936	8.139	3.366	3.40
Fully integrated S/R solid (elform=-2)	10.933	8.182	3.394	14.00
Fully integrated S/R solid (elform=-1)	10.936	8.183	3.405	4.10
1 point tetrahedron (elform=10)	11.044	7.201	3.057	2.70
1 point nodal pressure tetrahedron (elform=13)	10.987	8.214	4.288	9.00
1 point ALE (elform=5)	10.886	7.716	2.272	3.60

The above displacement and effective plastic strains results were obtained from the d3plot contour plots at 80.0 ms which were generated by the *DATABASE_BINARY_D3PLOT keyword.

The more efficient rigid boundary contact procedure requires less CPU time (20% to 60%) than the penalty method for contact-impact. The exception is the one point ALE multi-material formulation, where the CPU times were about the same, probably due to the smoothing operations control.

For all the element formulations (except the 1 point ALE) used, the contact-impact results provided using the penalty method and the planar rigid boundary differ due to the contact methods.

Comments provided for the penalty method results regarding element formulation CPU times, the (elform=1,2,-2,-1) similarities for dimensional changes and maximum effective plastic strain, the (elform=10) stiffer comparison, the (elform=13) less stiff comparison, and the (elform=5) similarity and difference are also appropriate for the rigid boundary contact results.

The LS-DYNA results for deformed shapes at 80.0 ms using rigid boundary contact with effective plastic strain contouring are given in Figures 27.7a to 27.3g.



Figure 25.7a – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=1).



Figure 27.7b – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=2).



Figure 27.7c – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=-2).



Figure 27.7d – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=-1).



Figure 27.7e – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=10).



Figure 27.7f – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=13).



Figure 27.7g – Quarter-symmetry deformed shape (rigid boundary) with effective plastic strain contouring at 80 ms (elform=5).

The half-symmetry deformed shape (planar rigid boundary contact), which illustrates the different element deformation for elform=1 and elform=5 at 80 ms, is given in Figure 27.8.





The histories of the kinetic energy, internal energy, hourglass energy, sliding energy, and the total energy for (elform=1) of rigid boundary impact are given in Figure 27.9, while the histories of the stable time step increment for all elforms (1,2,-2,-1,10,13,5) associated with the rigid boundary impact are given in Figure 27.10.



Figure 27.9 – Histories of the kinetic energy, internal energy, hourglass energy, stonewall energy, and the total energy for (elform=1) of rigid boundary impact.



Figure 27.10 – Histories of the stable time step increment for all elforms (1,2,-2,-1,10,13,5) associated with the rigid boundary impact.

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mate \$# *SEC \$# *MAI \$# *HOU \$# *HOU \$# *SH *SEC \$# 0. *MAI \$#	title pid pid 1 CTION_SC secid 1 T_PLASTI mid 1 Src 0.0 JRGLASS hgid 1 RT title pid 2 CTION_SF secid 2 tl .100000 T_RIGID mid	rpe # 3 (Ki secid 1 oLID c_KINEMATI c_KINEMATI c_KINEMATI ro 8.930e-03 srp 0.0 ihq 0 rpe # 20 (H secid 2 HELL elform 2 t2 0.100000 ro	nematic/Is mid 1 aet cc fs 0.0 qm 0.0 Rigid) mid 2 shrf 0.0 t3 0.100000 e	otropic E: eosid 0 0.35000 vp 0.0 ibq 0 eosid 0 nip 0 t4 0.100000	lastic-Plas hgid 1 sigy 4.000e+02 q1 0.0 hgid 2 propt 0 nloc 0 n	etan 1.000e+02 q2 0.0 grav qr/irid 0.0 marea 0.0 couple	adpopt beta 0.0 db 0.0 adpopt icomp	tmid qw 0.0 tmid setyp
mate \$# *SEC \$# *MAI \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *SEC \$# *MAI \$# *SEC \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *HOU \$# *SEC \$# *HOU \$# *HOU \$# *SEC \$# *HOU \$# *SEC \$# *HOU \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# \$# *SEC \$# \$# *SEC \$# \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$# *SEC \$#	citle pid pid 1 CTION_SC secid 1 T_PLASTI mid 1 Src 0.0 JRGLASS hgid 1 CTION_SF secid 2 CTION_SF secid 2 1 .100000 F_RIGID mid 2	rpe # 3 (Ki secid 1 oLID c_KINEMATI c_KINEMATI c_KINEMATI ro 8.930e-03 srp 0.0 ihq 0 rpe # 20 (H secid 2 HELL elform 2 t2 0.100000 ro 8.930e-03	nematic/Is mid 1 aet cc e 1.170e+05 fs 0.0 qm 0.0 agm 0.0 kigid) mid 2 shrf 0.0 t3 0.100000 e 1.170e+05	otropic E: eosid 0 0.35000 vp 0.0 ibq 0 eosid 0 nip 0 t4 0.100000 pr 0.35000	lastic-Plas hgid 1 sigy 4.000e+02 q1 0.0 hgid 2 propt 0 nloc 0 nloc 0	etan 1.000e+02 q2 0.0 grav qr/irid 0.0 marea 0.0 couple 0.0	adpopt beta 0.0 qb 0.0 adpopt icomp m 0.0	tmid qw 0.0 tmid setyp

1	.00000	0	7		7						
\$#1	co or	al	a2		a3	vl	v2		v3		
	Ο.	0	0.0		0.0	0.0	0.0		0.0		
*HO	URGLAS	S									
\$#	hgi	d	ihq		qm	ibq	ql		q2	qb	dw
		2	0		0.0	0	0.0		0.0	0.0	0.0
*C0]	NTACT_	AUTOMA	TIC_SU	JRFAC	E_TO_SU	JRFACE_TIT	FLE				
\$#	ci	d									title
		lcoppe	er bar-	-rigi	dwall i	nterface					
\$#	ssi	d	msid	-	sstyp	mstyp	sboxid	mbc	xid	spr	mpr
		1	2		3	3				-	-
\$#	f	s	fd		dc	vc	vdc	pen	lchk	bt	dt
	0.	0	0.0		0.0	0.0	0.0	L -	0	0.0	0.0
\$#	sf	s	sfm		sst	mst	sfst	S	fmt.	fsf	vsf
	0.	0	0.0		0.0	0.0	0.0		0.0	0.0	0.0
\$#	sof	t s	ofscl	1	cidab	maxpar	shopt	de	pth	bsort	frefra
Υ 11	501	2 0.1	00000	-	0	1.025	0.0	u.c	2	10	1
\$#	penma	- 0	hkont	g	hlthk	snlog	isvm	i 2	434	sldthk	sldstf
ΥII	0	0	0		1	0	10,10	12	0	0 0	0 0
\$#	iga:	n i	anore		-	0	0		Ū	0.0	0.0
ΥII	190	2 -	0								
* T N	τπτδτ.	VELOCT	TV GEN	лграт	TON						
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ŶΠ	1	1	2		0 0			-227	1 00	1 Vacii	1010
¢#	v	- C	VC		70	0.0 nv	0.0	221	.00 nz	nhase	iriaid
γĦ	0	0			0 0	0 0	0 0		0 0	pilase 0	111910
* एग	∪. ⊏мп:мп		0.0		0.0	0.0	0.0		0.0	0	0
сн сн	oid	01106. Di	d	n 1	2) 72	n /	ъБ	,	-6	, no
φ#	1	PI	1	111	112	113	114	206	2.	16 214	207
	T		T	т	ΞŪ) 11	2	300	5.	10 310	5 307
	10260		1 10	000	10693	10694	10749	11205	1000	20 1000	11052
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сн сн	oid	oneuu ri	d	n 1	2) 72	n /	ъБ	,	-6	, no
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+110	10512		2 11	1453	11454	11441	11440	0		0 (0
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	11454 5	-	-10.000	1000	1	.0000000	-0.	T00000		0 (J
^ EN	D										

Notes:

- 1. If a part is comprised entirely of tetrahedrons, there are several tetrahedral formulations to choose from, each with various pros and cons. Any of these formulations are preferable to using degenerate, elform=1 tetrahedrons. Two popular choices are (a) elform=10 which is 1 point tetrahedron with 4 nodes, but prone to volumetric locking (overly stiff behavior) in incompressible regimes, e.g., as in metal plasticity, and (b) elform=13 which is a 1 point nodal pressure tetrahedron developed for bulk metal forming; elform=13 is identical with elform=10 with addition of nodal pressure averaging that significantly decreases volumetric locking. There are also two relatively new 10-noded tetrahedron, elform=16 and 17 which have not been widely used.
- 2. To convert from a Lagrangian simulation to an ALE, the user needs to add the keyword *CONTROL_ALE which defines the ALE control parameters of advection
logic (dct), cycle between advection (nadv), advection method (meth), smoothing weight factors (afac thru efac), etc.:

*CON	TROL_AI	ΞE						
\$#	dct	nadv	meth	afac	bfac	cfac	dfac	efac
	-1	1	2	0.500000	0.500000	0.0	0.0	0.0
\$#	start	end	aafac	vfact	vlimit	ebc	pref	
	0.0	1.000e+20	1.000000	1.000e-06	0.0	0	0.0	

and modify the element formulation choice:

*SECTION_SOLID \$# secid elform aet 1 5

- 3. Hexahedral elements with reasonable aspect ratios should be used for the initial ALE mesh. Degenerate element shapes, such as tetrahedrons and pentahedrons, should be avoided as they lead to reduced accuracy and perhaps numerical instability during the advection.
- 4. The viscous contact damping parameter, vdc, on card 2 of the *CONTACT_AUTOMATIC_SURFACE_TO_SURFACE keyword is zero by default. Contact damping is often beneficial in reducing high-frequency oscillation of contact forces in crash or impact simulations. In contacts involving soft materials such as foams and honeycombs, instabilities exist due to contact oscillations. Using a value of vdc between 40-60 (corresponding to 40% to 60% of critical damping), improves stability; however, it may be necessary to reduce the time step scale factor. Generally, a smaller value of vdc, say equal to 20, is recommended when metals, which have similar material stiffnesses, interact.
- 5. Contact-impact results using penalty method and planar rigid boundaries can possible differ due to their approaches. The penalty method consists of placing normal interface springs between all penetrating nodes and the contact surface. The rigid boundary contact procedure for stopping nodes uses a constraint method which represents a perfectly plastic impact that results in an irreversible energy loss. The total energy dissipated is found by taking the difference between the total kinetic energy of all the nodal points slaved to the rigid wall before and after the impact with the wall. The advantage of the constraint method is that it guarantees the node to lie on the positive side of the rigidwall (no penetration).
- 6. To move from the contact-impact model used by the penalty method, i.e. that with a meshed rigid wall, to a planar rigid boundary model, the user needs to remove the following entries used to represent the contact-impact and the meshed rigid wall:

*C0	NTROL_CON	TACT						
\$#	slsfac	rwpnal	islchk	shlthk	penopt	thkchg	orien	enmass
0	.100000	0.0	2	0	0	0	1	
\$#	usrstr	usrfrc	nsbcs	interm	xpene	ssthk	ecdt	tiedprj
	0	0	0	0	4.00000			
\$#	sfric	dfric	edc	vfc	th	th_sf	pen_sf	
	0.0	0.0	0.0	0.0	0.0	0.0	0.0	
\$#	ignore	frceng	skiprwg					

0	0	0

*P#	ART								
2# mot	ullie omial tu	mo # 20 /T	i a i a i						
ша i	Lerial Ly	pe # 20 (r	(igiu)		الم من ما				5
ŞĦ	pia	secia	mia	eosia	ngia	grav	adr	popt	tmia
	2	2	2	0	2				
*SI	ECTION_SH	IELL							
\$#	secid	elform	shrf	nip	propt	qr/irid	ic	comp	setyp
	2	2	0.0	0	0	0.0			
\$#	t1	t2	t3	t4	nloc	marea			
(0.100000	0.100000	0.100000	0.100000	0	0.0			
*M7	AT RIGID				-				
¢#	mid	ro	0	nr	n	couple		m	
γĦ	2	0 0 2 0 2 0 2	1 1700-05	0 25000	0 0	COUPIE		0 0	
<i>ж</i> н	2	0.9300-03	1.1/00+05	0.35000	0.0	0.0		0.0	
\$#	cmo	conl	con2						
1	L.000000	7	7						
\$#]	lco or al	. a2	a3	v1	v2	v3			
	0.0	0.0	0.0	0.0	0.0	0.0			
*H0	DURGLASS								
\$#	hgid	ihq	qm	ibq	q1	q2		qb	qw
	2	0	0.0	0	0.0	0.0		0.0	0.0
*00	NTACT AU	TOMATIC SI	IRFACE TO S	URFACE TT	T.E.				
¢#	cid	10111110_00		John Hend					title
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ėщ		opper bar-	-iigiuwaii	Incertace	مام محمد أحا	الم المحمد الم		~~~~	
Ş₩	SSIG	liista	sstyp	mstyp	SDOXIO	IIDOXIA		spr	mpr
	1	2	3	3					
\$#	fs	fd	dc	VC	vdc	penchk		bt	dt
	0.0	0.0	0.0	0.0	20.0	0		0.0	0.0
\$#	sfs	sfm	sst	mst	sfst	sfmt		fsf	vsf
	0.0	0.0	0.0	0.0	0.0	0.0		0.0	0.0
\$#	soft	sofscl	lcidab	maxpar	sbopt	depth	bs	sort	frcfrq
	2	0.100000	0	1.025	0.0	- 2		10	1
\$#	penmax	t.hkopt.	shlthk	snlog	isvm	i2d3d	slo	lt.hk	sldstf
4.0	0 0	0	1	0	0	0		0 0	0 0
¢#	idan	ignore	1	0	0	0		0.0	0.0
φ#	Igap	Ignore							
	2	0							
*EI	LEMENT_SH	IELL							
\$#	eid	pid	nl n	ı2 n3	n4	n5	nб	n7	n8
	10369	2 11	L299 1130	0 11287	11286	0	0	0	0
	10512	2 11	L453 1145	4 11441	11440	0	0	0	0
*N(DDE								
\$#	nid		x	v		Z	tc	rc	
7 11	11286	10 000		10 00000	_0 1	00000	0	- 0	
	11200	10.000		10.000000	0.1		0	0	
	11/5/	10 000	000	10 000000	0 1	00000	0	0	
	11454	-10.000	0000	T0.000000	-0.1	.00000	U	U	

and replace them with the following planar rigid boundary entry (the user can also include the rigid wall force entry if desired):

*DAT \$# 5.00	ABASE_R dt 00e-04	WFORC binary 1						
*RIG	IDWALL_	PLANAR						
\$#	nsid	nsidex	boxid	offset	birth	death	rwksf	
	0	0	0					
\$#	xt	yt	zt	xh	yh	zh	fric	wvel
Ο.	000000	0.000000	-0.050000	0.000000	0.000000	1.000000	0.000000	0.000000

Implicit Studies:

NIKE3D (implicit dynamics solver) was used by Ferencz [1989] with the computation divided into 80 time steps of 1 microsecond and nodal boundary conditions constraining the impacting face to lie on the global X-Y plane. The half-symmetry deformed shape, at 80 ms (final state), is shown in Figure 27.11.

NIKE3D uses an element formulation, similar to the selected reduced integration of LS-DYNA (elform=2), defined as B-Bar method. The selective reduced integration splits the stress tensor into deviatoric and dilatation (mean) parts, whereas the B-Bar method splits the B matrix (a strain modification) into dilatational and deviatoric parts.

The contact of the deformable body and the rigid wall can be modeled in one of the following ways in this study:

- using nodal boundary conditions which constrain the impacting face to remain on the rigid wall,
- rigid wall (*RIGIDWALL_PLANAR), which provides an easy way to treat contact between a rigid-flat surface and the nodes of a deformable body.

In general, there are two different methods that are available in LS-DYNA to treat nodes impacting a rigid wall. The first method, which is the default method, is the constraint type that is used for all deformable nodes impacting a rigid wall. The second (optional) method is the penalty approach that is used for all rigid nodes or optional deformable nodes impacting the rigid wall. The primary difference between the two methods is in the conservation of energy and momentum. If using the implicit solver, only the penalty approach method is available.

The default constraint method does not conserve momentum and the energy. This is due to the fact that when a deformable node is found to penetrate a rigidwall, its velocity is immediately reset to zero and is moved back onto the surface of the rigidwall. The advantage of the constraint method is that it always guarantees the node to lie on the positive side of the rigidwall (no penetration).

The penalty method (optional for explicit solver/default for implicit solver) for rigid walls uses a scale factor that can be adjusted (default is 1.0) by modifying the rwskf parameter on *RIGIDWALL_PLANAR keyword. This works the same as the contact-impact interface treatment. When a deformable or a rigid node is found to penetrate a rigidwall, the penetrated distance normal to the rigid wall is computed and is resisted by applying a force that is proportional to the computed distance multiplied by a stiffness factor that is based on the material of the impacting node and the dimensions of the attached element. The penalty approach conserves both energy and momentum.



0.080 ms



Analysis Summary:

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Velocity	Non- linear	Non- linear	SPC's	Implicit	2-Nonlinear w/BFGS

and

Dim.	Туре	Load	Material	Geometry	Contact	Solver	Solution Method
3D	Dynamic	Velocity	Non- linear	Non- linear	R.Wall (penalty)	Explicit/ Implicit	2-Nonlinear w/BFGS

Element Type:

Fully integrated S/R solid (elform=2)

Different Considerations from Explicit Solver:

The contact of the deformable body and the rigid wall can be modeled in one of the following ways in this study:

- using nodal boundary conditions which constrain the impacting face to remain on the rigid wall,
- rigid wall (*RIGIDWALL_PLANAR), which provides an easy way to treat contact between a rigid-flat surface and the nodes of a deformable body.

Studies 1 and 2:

- NIKE3D and LS-DYNA (each using implicit dynamics solver) Comparison, and
- Implicit LS-DYNA Convergence,

with nodal boundary conditions constraining the impacting face for both studies.

Nodal Boundary (SPC's)	Shortening (mm)	Widening (mm)	Max. plastic strain (ε^p)	Normalized CPU Time
B-Bar solid (NIKE3D) - 80 time steps	11.446	7.68 est.	2.248	-
Fully integrated S/R solid 80 time steps	11.313	6.111	2.432	3.35
Fully integrated S/R solid 160 time steps	11.150	7.468	3.126	6.54
Fully integrated S/R solid 320 time steps	11.029	7.730	3.080	13.10
Fully integrated S/R solid 400 time steps	11.005	7.765	3.084	16.84
Fully integrated S/R solid 480 time steps	10.989	7.784	3.084	19.50
Fully integrated S/R solid 640 time steps	10.974	7.798	3.090	25.90
Fully integrated S/R solid 800 time steps	10.967	7.805	3.092	33.38

The above displacement and effective plastic strain results were obtained from the d3plot contour plots at 80.0 ms which were generated by the *DATABASE_BINARY_D3PLOT keyword.

Normalized CPU times shown in the above Nodal Boundary Condition (SPC's) results table were normalized using the explicit fully integrated S/R solid (elform=2) value.

In the implicit solver direct comparison (80 time steps) of NIKE3D which uses the B-Bar element formulation and the selected reduced integration element formulation of LS-DYNA (elform=2), similar maximum effective plastic strain results ($\varepsilon^{p} = 2.248$ vs. $\varepsilon^{p} = 2.432$) and length shortening (11.446 mm vs. 11.313 mm) are obtained. For some unexplained reason, the widening profiles (7.680 mm vs. 6.111 mm) differ significantly.

The maximum effective plastic strain obtained using the LS-DYNA implicit solver $(\varepsilon^p = 2.432)$ is significantly less than the explicit solver value $(\varepsilon^p = 3.366)$ - initial work with penalty contact condition). This is believed to be due to the relatively large time step increment used (only 80 steps) which fails to capture the correct dynamics of the simulation. It is shown in above table that increasing the number of time steps (reducing the time step increment) allows the implicit solver to more accurately capture the rate of material deformation (plastic flow) and appears to be converging to a unique solution ($\varepsilon^p = 3.100$ and 7.810 mm) with a consistent shape profile.

The half-symmetry deformed shape (nodal boundary constraint) with widening profiles and effective plastic strain contouring for selected implicit integration time step sizes at 80 ms, are given in Figures 27.12 (80 time steps), 27.13 (160 time steps), and 27.14 (640 time steps). Figure 27.12 provides a LS-DYNA deformed shape (80 time steps) comparison with the NIKE3D result (80 time steps) shown in Figure 27.11. Together, Figures 27.12 ($\varepsilon^p = 2.432$ and 6.111 mm), 27.13 ($\varepsilon^p = 3.126$ and 7.468 mm), and 27.14 ($\varepsilon^p = 3.090$ and 7.798 mm) illustrate the LS-DYNA converging results with increasing the number of time steps (reducing the time step increment).

Unfortunately, as is shown in above table, the CPU time becomes a deterrent when using implicit dynamics solvers. Thus, the explicit solver is often favored for these types of high deformation, impact simulations due to its ability to provide efficient and stable solutions.



Figure 27.12 – Half-symmetry widening and effective plastic strain contouring with nodal boundary conditions - 80 time steps.



Figure 27.13 – Half-symmetry widening and effective plastic strain contouring with nodal boundary conditions - 160 time steps.



Figure 27.14 – Half-symmetry widening and effective plastic strain contouring with nodal boundary conditions - 640 time steps.

Input deck:

*KE	YWORD							
*TI	TLE							
Cor	per Bar 1	Impacting a	. Rigidwall					
*CC	NTROL_IM	PLICIT_DYNA	MICS					
\$#	imass	gamma	beta					
	1	0.500000	0.250000					
*CC	NTROL_IM	PLICIT_GENE	RAL					
\$#	imflag	dt0	imform	nsbs	igs	cnstn	form	
	1	0.00100	0	0	0			
\$	1	0.00050	0	0	0			
\$	1	0.00025	0	0	0			
\$	1	0.00020	0	0	0			
\$	1	0.0001666	0	0	0			
\$	1	0.0001250	0	0	0			
\$	1	0.00010	0	0	0			
*CC	NTROL_IM	PLICIT_SOLV	/ER					
\$#	lsolvr	prntflg	negeig	order	drcm	drcprm	autospc	aspctl
	4	2	2	0	1	0	1	0
\$#	lcpack							
	2							
*CC	NTROL_IM	PLICIT_SOLU	JTION					
\$#	nsolvr	ilimit	maxref	dctol	ectol	rctol	stol	abstol
	2	11	15	0.0010	0.0100	1.00e+10	0.900000	1.00e-10
\$#	dnorm	diverg	istif	nlprint	nlnorm	d3itctl	cpchk	
	2	1	1	2				
\$#	arcctl	arcdir	arclen	rcmth	arcdmp			
	0	1	0.0	1	2			
*CC	NTROL_TE	RMINATION						
\$#	endtim	endcyc	dtmin	endeng	endmas			
8.	000e-02	0	0.0	0.0	0.0			

*CONSTRAINED_GLOBAL									
\$#	tc	rc	dir	x	У	Z			
	3	0	3	0.0	0.0	0.0			

*END

Notes:

Studies 3 and 4:

- LS-DYNA Explicit Solver (with rigid wall constraint method contact) and Implicit Dynamics Solver (with rigid wall penalty method contact) Comparison, and
- Implicit LS-DYNA Convergence.

Planar Rigid Boundary	Shortening (mm)	Widening (mm)	Max. plastic strain (\mathcal{E}^p)	Normalized CPU Time
Fully integrated S/R solid Explicit - 9883 time steps	10.936	8.139	3.366	1.00
Fully integrated S/R solid Implicit - 80 time steps	11.155	7.083	2.821	3.35
Fully integrated S/R solid Implicit - 160 time steps	11.285	7.326	3.109	6.54
Fully integrated S/R solid Implicit - 320 time steps	11.216	7.502	3.235	13.10
Fully integrated S/R solid Implicit - 400 time steps	11.179	7.534	3.270	16.84
Fully integrated S/R solid Implicit - 480 time steps	11.166	7.550	3.274	19.50
Fully integrated S/R solid Implicit - 640 time steps	11.045	8.503	3.586	25.90
Fully integrated S/R solid Implicit - 800 time steps	11.033	8.517	3.644	33.38
Fully integrated S/R solid Implicit - 1600 time steps	11.023	8.542	3.680	67.30
Fully integrated S/R solid Implicit - 3200 time steps	11.022	8.545	3.696	134.10

The above displacement and effective plastic strain results were obtained from the d3plot contour plots at 80.0 ms which were generated by the *DATABASE_BINARY_D3PLOT keyword.

Normalized CPU times shown in the above Rigid Wall Planar results table were normalized using the explicit fully integrated S/R solid (elform=2) value.

As for the previous nodal boundary condition method, the maximum effective plastic strain ($\varepsilon^p = 2.821$) and widening profile (7.083 *mm*) for the 80 time step solution are roughly 15% less than the explicit results ($\varepsilon^p = 3.366$ and 8.139 *mm*). As before, this is believed to be due to the relatively large time step increment used (only 80 steps) which fails to capture the correct dynamics of the simulation. It is shown that increasing the number of time steps (reducing the time step increment) allows the solver to better capture the rate of material deformation (plastic flow) which appears to be converging ($\varepsilon^p = 3.274$ and 7.550 *mm*) over a range of time steps studied.

For some unexplained reason, starting with the 640 time step solution, there is a further increase in maximum effective plastic strain and widening results and a distinct change in the widening profile with the outer row of nodes now turning more upward. The corresponding results ($\varepsilon^p = 3.696$ and 8.545 mm) appear to be converging, though greater than those provided by the explicit solver ($\varepsilon^p = 3.366$ and 8.139 mm) which also has the outer row of nodes turning slightly upward.

The half-symmetry deformed shape (planar rigid boundary) with widening profiles and effective plastic strain contouring for selected implicit integration time step sizes at 80 ms, are given in Figures 27.15 (explicit), 27.16 (80 time steps), 27.17 (160 time steps), and 27.18 (640 time steps). Figure 27.15 provides a LS-DYNA widening and effective plastic strain results ($\varepsilon^p = 3.366$ and 8.139 mm) for the explicit solver. Together, Figures 27.16 ($\varepsilon^p = 2.821$ and 7.083 mm), 27.17 ($\varepsilon^p = 3.109$ and 7.326 mm), and 27.18 ($\varepsilon^p = 3.586$ and 8.503 mm) illustrate the LS-DYNA converging results for the implicit solver with increasing the number of time steps (reducing the time step increment).

Unfortunately, as is shown in above table, the CPU time becomes a deterrent when using implicit dynamics solvers. Thus, the explicit solver is often favored for these types of high deformation, impact simulations due to its ability to provide efficient and stable solutions.



Figure 27.15 – Half-symmetry widening and effective plastic strain contouring with rigid boundary condition - explicit.



Figure 27.16 – Half-symmetry widening and effective plastic strain contouring with rigid boundary condition - 80 time steps.



Figure 27.17 – Half-symmetry widening and effective plastic strain contouring with rigid boundary condition - 160 time steps.



Figure 27.18 – Half-symmetry widening and effective plastic strain contouring with rigid boundary condition - 640 time steps.

Input deck:

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Notes: