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STABILITY OF BELLOWS USED AS EXPANSION JOINTS BETWEEN SUPER-CONDUCTING MAGNETS IN ACCELERATORS

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ABSTRACT

For superconducting magnets, one needs many bellows for connection of various helium cooling transfer lines. There could be approximately 20,000 magnet interconnection bellows in the SSC exposed to an internal pressure. When axially compressed, internally pressurized, or insufficiently supported at their ends, bellows can become unstable, leading to gross distortion or complete failure. If several bellows are contained in a magnet assembly, failure modes might interact.

If designed properly large bellows can be used to connect the large tubular shells that support the magnet iron yokes and superconducting coils and contain supercritical helium for magnet cooling. We investigate here bellows design features and end supports to insure that instabilities will not occur in the bellows pressure operating region, including some margin. A model of three superconducting accelerator magnets connected by two large bellows is analyzed in order to ascertain that support requirements are satisfied and in order to study interaction effects between the two bellows. Specific details of large and small bellows design and reliability for our application will be addressed.

INTRODUCTION

Bellows can be compressed, extended, bent, internally or externally pressurized, supported more or less rigidly at the ends by magnets which act as lateral and torsional springs. (The torsion, or rotation, is in the magnet plane, not about the bellows axis). Instabilities are due to various reasons: axial compression, internal pressure, support system, manufacturing and installation inaccuracies. External pressure is not a cause for instability unless the bellows buckles, or is "crushed", due to excessive pressure.

Early magnetic field test set-ups at Fermilab and Brookhaven National Laboratory resulted in two apparent large bellows failures. These were caused by the design of the bellows end supports in the test facility. At one end the bellows was mounted on a magnet but on the other end it was supported by a rod with a much higher

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slenderness ratio (effective length/radius of gyration of the cross-section) than that of the magnet. Historically, it is the investigation of these incidents that led to the work reported in SSC Technical Note No.89 (SSCL-N-742) and in a shortened version (BNL-44548). These reports establish design guidelines for bellows, understanding of failure modes including those due to the support system, and interaction between bellows in a magnet-bellows assembly configuration. Some of the basic results derived in these reports are presented here again in a simpler, if somewhat less general, formulation which is sufficient to illustrate the theory's salient features. Specifically: force exerted by internal pressure on bellows, Euler buckling mode, bellows axial stiffness, critical pressure, critical precompression. Other aspects of the work which do not lend themselves to concise treatment are simply reiterated. These are: detailed bellows parameter design, stresses and deflection of bellows both unpressurized and internally pressurized, model of a bellows (equivalent to a bar) with offset and initial shape mounted on lateral and torsional springs, and assembly of three magnets and two bellows.

The current note draws on our previously developed theory to address the specific details of bellows design for the SSC. Two types of bellows are analyzed: the first type is a "large" bellows with diameter somewhat larger than the magnet shell, the second type is a "small bellows", specifically the small bellows used in the current Fermilab interconnect design for the SSC. This work shows that large bellows can be reliable if properly designed and supported. Small bellows require the use of "squirm-cans" which are tubes lining (internally or externally) the bellows to prevent them from distorting.

BASIC RELATIONS

Force Exerted by Internal Pressure on Bellows

It was shown in the above mentioned report that the walls of an internally pressurized (pressure p) bellows of average diameter D are subjected to the force increment

$$\Delta f = \frac{\pi}{4} D^2 p \frac{\Delta x}{\rho} = \frac{\pi}{4} D^2 p y'' \Delta x \quad (1)$$

where the curvature $\frac{1}{\rho}$ can increase with the internal pressure p (fig.A1). This can lead to instability at some critical pressure p_{cr} . Bending calculations show that the effect of internal pressure can be replaced by a force

$$F_x = pA \quad (2)$$

acting axially at the bellows ends ($A = \pi D^2/4$ is the average bellows cross-section).

Euler Buckling Mode

When pressurized, a bellows initially unstressed (fig.A2.a) of axial stiffness K with unsupported ends would extend by $\delta = F_x/K$ with F_x given above (fig.A2.b). However the bellows whose free length is l is mounted between magnets which act as fixed ends. Thus due to the lack of room to expand, the bellows is effectively compressed by the length d (fig.A2.c). The effect of internal pressure on the bellows is identical to a compressive force F_x on a bar. The lowest buckling mode for a bar simply supported at both ends ("hinged") is

$$F_{cr} = \frac{\pi^2 EI}{l^2} \quad (3)$$

(fig.A2.d), with E =elastic modulus (or equivalent elastic modulus for bellows) and I =moment of inertia of the bellows cross-section.

Bellows Axial Stiffness

The bellows axial stiffness is obtained from the following basic equations:

- stress-strain formula : $\sigma = E\epsilon$
- definition of strain : $\epsilon = \delta/l$
- relation between stress and force : $F = \sigma S$ (S =average bellows wall cross-section)
- relation between axial force and axial stiffness: $F = K\delta$

these relations give the "equivalent" Young's modulus of the bellows:

$$E = \frac{Kl}{S} \quad (4)$$

Critical Pressure

Using the Euler critical load of a bar F_{cr} , one finds that if $F_{cr} = p_{cr}A = \frac{\pi^2 EI}{l^2}$ then the critical pressure p_{cr} at which the bellows will fail under compression is related to the axial stiffness by

$$p_{cr} = \frac{\pi K}{2l} \quad (5)$$

where K is assumed to be given. It is noted that this expression is independent of the average bellows diameter D which is equal to the inner diameter D_i plus the convolution height d . The moment of inertia $I = \frac{\pi D^3 t}{8}$ and cross-section $S = \pi D t$ have been used for the bellows where t (much smaller than D) is the bellows wall thickness.

Precompression Length and Beam Tube Bellows

At installation, magnets are warm and bellows are precompressed by an amount λ_c . When cold, the magnets shrink, stretching the bellows past the free length and extending them by an amount λ_e . The total bellows travel usually must be split between extension and compression as follows: 3/4 for compression and 1/4 for extension in order to minimize bellows wall stresses, including considerations concerning material fatigue. This distribution of travel between compression and extension applies at room temperature. At lower temperatures it may be possible to increase the extension by the same fraction by which the material's strength is increased (about 20% to 40% at 4K).

From the relations listed above, one finds that the critical length by which one can precompress the magnets is

$$\lambda_{cr} = \frac{\pi^2 D^2}{8l} \quad (6)$$

The design pressure of the magnet containment is 300 psi (=20 atm=2 MPa). Using the usual margin of 50 %, the bellows should be designed for $p_{cr} = 450$ psi. (In the general theory axial deflection (λ) and pressure (p) effects are treated together.)

REVIEW OF PREVIOUS WORK

The previously mentioned report contains an approximate theory which is applicable if the bellows radial width of a convolution (d) is much smaller than the bellows diameter (D).

Bellows Design

The report gives the dependence of K on the wall thickness (t), convolution valley radius (r_i), other geometrical parameters, bending stresses (which are very close to yielding and thus require fatigue considerations), lateral spring constant, etc... The theory permits adjustment of bellows parameters to fit requirements and to compare with manufacturers' recommendations. Axial deflections and pressure effects are considered.

Spring - Supported Bellows Model

A model (fig.A3) of a bar with an initial offset and initial (prebent) shape mounted on lateral and torsional springs is used to gain insight into bellows behavior, in particular instabilities resulting from a weak support system. In the limit this analytical model which is a generalization of the conventional calculation for buckling of bars reduces to the classical cases (hinged, fixed, guided etc..) corresponding to various boundary conditions.

Magnet - Bellows Assembly

Finally a numerical model of any applicable combination of three magnets (dipoles or quadrupoles) and two bellows was also presented (fig.A4). It illustrates the effect of magnet prebend, magnet support and support location, and bellows to bellows interaction. In particular the question as to whether complete failure ("catastrophe") of one bellows causes failure of adjacent bellows is addressed. 18 bending equations, one for each span between supports or nodes, force and momentum equilibrium, boundary conditions resulted in a system of 51 equations. The solution of this system yields the deflection of the magnet and bellows. Of particular interest is the expression

$$\Delta l_{max} = |\lambda| + \frac{|y''|Dl}{2} \quad (7)$$

which is a measure of the behavior of the bellows. This expression combines the axial compression or extension λ of a bellows with the maximum local elongation or compression of the convolutions due to bending of the bellows. ($|y''| = \frac{\text{Bending Moment}}{EI}$). Δl_{max} , the maximum bellows local elongation, should not be confused with the strain in the bellows wall material. When plotted versus internal pressure, Δl_{max} shows peaks at instabilities and must be compared with the allowable elongations suggested by manufacturers. These in turn are based on maximum usable bellows wall bending stress and fatigue considerations.

NUMERICAL RESULTS

If the maximum operating pressure is $p_{op} = 300$ psi, a safety factor of 1.5 results in a required critical pressure of $p_{cr} = 450$ psi. Stability of large and small bellows at this critical pressure and effects of supports are presented next.

Large Bellows

A possible design for the interconnect region, originally considered but currently abandoned in favor of small bellows, made use of large bellows connecting the shells of adjacent magnets. Requesting that the bellows fail in compression at 450 psi for a free length of 9.3 in, inner diameter $D_i = 14.3$ in, radial convolution height $d = 0.5$ in, and precompression $\lambda = 1$ in, gives a computed axial stiffness $K = 2760$ lbs/in. (These numbers are tabulated in forthcoming tables). The first concern, whether the support system is adequate, is addressed by examining the spring-supported bellows model

with lateral stiffness $k_1=7000$ lb/in and torsional stiffness $k_{t1}=3e7$ lbs in/radian. (Left and right supports are assumed to have identical springs).

Fig.B1 shows instabilities due to bellows end supports only (the instability due to the Euler buckling mode at 450 psi here reduces to an indeterminacy and does not appear as a peak in the spring-supported model but is quite visible in the magnet-bellows assembly). Δl_{max} , in inches, is plotted versus the bellows internal pressure in psi. The bellows is assumed to be mounted with an initial offset of $\zeta = 0.03$ in and to be perfectly straight as $d_1 = d_2 = 0$ indicate on the legend. The value of Δl_{max} at the operating pressure of 300 psi is of interest since it must remain below manufacturer's recommendations. This value is quite small for the present case (no precompression was assumed, the effect of precompression would be to shift the curve upwards by a corresponding amount).

Since it is unlikely that the bellows is perfectly straight, one can assume that it has an initial sine shape $d_1 \sin(\pi x/l)$ of maximum amplitude $d_1 = 0.02$ in and an initial cosine shape $d_2(1 - \cos(2\pi x/l))$ with maximum amplitude $d_2 = 0.02$ in. Comparison between fig.B2 and fig.B1 reveals a large increase of Δl_{max} at p_{op} . (All curves are truncated at 5 in for clearer display of details.)

Should the supports be weak, instabilities due to support might well occur at lower values than $p_{cr} = 450$ psi for which the bellows axial stiffness was chosen. Such an occurrence is illustrated in fig.B3 with $k_{t1} = 1e4$ lbs in/radian where an unstability appears at 425 psi. Fortunately, this is far from being a realistic situation since torsional magnet stiffness is estimated to be $3e7$ lbs in/radian.

Proceeding now to the model described in fig.A4, two of the bellows are inserted between three dipoles in fig.C1. Both bellows are assumed to be perfectly straight (no sine or cosine shape) and ideally mounted (no lateral or angular offsets). The bellows are designed to fail at 450 psi where the mesh of points is denser on the curve. The graph shows two curves, one for each of the bellows. Since both bellows are identical, the two curves overlap. The first support peak which appears above 600 psi is well above 450 psi. No peak is visible at 450 psi because of the zero value of the mounting offset, but it is anticipated that the peak will appear in the next figure for non-zero offsets.

Fig.C2 shows the case where the first bellows only has a mounting offset $\zeta_6 = 0.03$ in and an initial sine and cosine shape of maximum amplitudes 0.02 in each. The Euler peak is now visible for the misaligned bellows (curve made of triangles). The support peak is much larger: in general, misalignments, offsets, eccentricities do not cause instabilities but they increase the rate of rise to peak. In the limiting case where these parameters are zero as in fig.C1 the peak is no longer visible on the curve but a potential instability is always there in the form of an indeterminacy. Companion tables appear in figs.C2a and C2b. Table 1 shows the input to the bellows design part of the computations, table 2 gives parameters which are the result of the input of table 1. Table 3 shows the corresponding wall stresses and deflections, the total maximum stress is the sum of σ_{max} and σ_{pw2} . Table 4 lists the misalignments of the two bellows and table 5 lists the maximum amplitude of the initial sine and cosine shapes. Tables 6, 7 give some of the geometrical details such as length, overhang, moment of inertia of the dipoles and the quadrupoles. Table 8 gives the maximum bellows elongations at the operating pressure of 300 psi. In table 9 are listed nodes' coordinates, support stiffnesses, and force at operating pressure. One can distinguish magnet nodes from bellows nodes by noting that external support stiffnesses for bellows ends are zero.

Fig.C3 investigates a "catastrophe" where the first bellows was made to fail by applying a couple of forces of plus/minus 12500 lbs at the bellows ends. Complete failure of one bellows does not seem to affect the behavior of the adjacent bellows. The interaction, if any, between bellows is negligible in the operating region.

In fig.C4 the bellows are connected by two dipoles and a quadrupole. There is an interaction effect which is moderate at operating pressure and is due to the fact that a quadrupole has only 3 supports instead of the five for a dipole. The interaction can be minimized by increasing magnet support stiffnesses or moving the supports closer to the bellows to reduce the overhang as much as possible.

Fig.C5 shows the deflected shape of the assembly of three dipoles and two bellows. The 0.03 in offset is the dominant feature of the first bellows deflected shape, the second bellows shows little distortion.

Small Bellows

The current Fermilab design for the SSC interconnect uses small bellows made of 2 ply material that have a very low axial stiffness of $K = 161$ lbs/in in order to accommodate the required travel of 2 in. These bellows are mounted on pipes and not directly on the magnets themselves as were the large bellows. Fig.D1 is an enlarged detail of fig.D2 and serves to illustrate the fact that these small bellows can become unstable already at 35 psi: they must thus be used with liners (called "squirm supports"). Bellows design parameters are: inner diameter $D_i = 3.875$, radial convolution height $d = 0.33$, free length $l = 5.75$, precompression $\lambda_c = 0.83$. The first bellows only is misaligned and has an initial bent shape. Fig.D2 shows the location of support peaks which occur well below 450 psi for bellows with such a low axial stiffness and mounted on pipes. The stresses of this two ply bellows are estimated by applying half of the internal pressure on one ply only. By demanding a critical pressure at $35/2=17.5$ psi, the stiffness of a bellows with same inner diameter and length is about half of the 161 lbs/in and the stresses are 113 ksi (sum σ_{max} with σ_{pw2} of table 3 fig.D2.a) which is high but seems to be in the range used by manufacturers.

One might ask the following question: why not use a single ply bellows without liner with axial stiffness such that it fail at 450 psi ? The answer appears in fig.D3 where the stiffness of the small bellows has been increased to $K=2100$ lbs/in. This bellows is mounted on pipes and support peaks occur above 450 psi. It is shown again in fig.D4 but mounted between magnets instead of pipes. The problem with such a bellows is that the bending stresses are greater than 370 ksi (sum σ_{max} with σ_{pw2} of table 3 fig.D4.a) which is unacceptable and explains why one needs a two ply bellows with low K and anti-squirm protection.

Beam Tube Bellows

There are currently two identical bellows in the beam tube region of the interconnection: one is the "beam tube" bellows which is not internally pressurized, the other is the "differential expansion" bellows which is pressurized. The differential expansion bellows is put in place to accommodate sudden contraction of the ends of the beam tube in case of too rapid cooling. Using Eq. (6) with $D_i = 2$, $d = 0.336$, $l = 5.25$, one finds $\lambda_{cr} \simeq 0.93$ in. The total motion that both bellows must be able to accommodate due to the thermal contraction of the magnets is 2 in.

The critical pressure (Eq. (5)) which was derived in our previous report is not affected by the presence of an internal tube inside the bellows. This expression was obtained by considering pressure acting on elements on the bellows walls and consequently did not depend on what was inside the bellows. In the case of the differential

expansion bellows which contains the beam tube inside, one finds that the critical pressure is $p_{cr} = 27.2$ psi.

CONCLUSIONS.

Large bellows which can travel the required two inches do not require liners since their axial stiffness is quite high and they can be designed to be stable at operating pressure with a safety factor of 1.5. Attention should be paid to proper design of the supports, in particular to their torsional stiffness.

Properties of small bellows and beam tube bellows used in the current interconnect design are summarized in Table 1 where eqs.(2.5) and (2.6) were used to compute the critical pressures and compression lengths.

Table 1: small bellows properties

bellows type	D_i in	d in	l in	K lbs/in	λ_{cr} in	p_{cr} psi
single phase He	3.875	0.33	5.75	161	3.8	35
beam tube	2	0.336	5.25	91	0.93	no int pres
diff. expansion	2	0.336	5.25	91	0.93	27.2

In order to accommodate the required travel without overstressing the bellows walls, the axial stiffness must be so low that instabilities occur well below the design pressure and liners must be used for these small bellows.

Table 2 lists the respective advantages and disadvantages of large versus small bellows.

Table 2: small vs large single phase bellows

bellows type	small	large
safety factor at 450 psi	0.12	1.5
squirm can	friction+guided	none
He interconnect volume	reduced	ok
He flow impedance	increased	ok
beam tube vacuum to vacuum	yes	need double bellows
access to beam tube liners	yes	more difficult

DISCLAIMER

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Fig.A1

fig.2a

fig.2b

fig.2c

fig.2d

Fig.A2

Fig.A3

Fig.A4

Spring-supported Bellows model: max elong vs pres

Fig.B1

Spring-supported Bellows model: max elong vs pres

Fig.B2

Spring-supported Bellows model-max elong vs pres

Fig.B3

MAX BELLOWS LOCAL ELONGATION

Fig.C1

MAX BELLOWS LOCAL ELONGATION

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Fig.C2

SSC ddd
 zet6=0.030 in
 zet7=0.000 in
 zet13=0.000 in
 zet14=0.000 in
 et6=0.000 in
 et7=0.000 in
 et13=0.000 in
 et14=0.000 in
 ds6=0.020 in
 ds13=0.000 in
 f(6)= 0.0 lbs
 f(13)= 0.0 lbs
 E= 2759.2 lbs/in

MAX BELLOWS LOCAL ELONGATION

Fig.C3

SSC ddd
 zet6=0.030 in
 zet7=0.000 in
 zet13=0.000 in
 zet14=0.000 in
 et6=0.000 in
 et7=0.000 in
 et13=0.000 in
 et14=0.000 in
 ds6=0.020 in
 ds13=0.000 in
 f(6)=-12500.0 lbs
 f(13)= 0.0 lbs
 E= 2759.2 lbs/in

Table 1: Input: bellows design parameters

r_i in	D_i in	d in	β	l in	per psi	λ in	E_b psi
0.040	14.300	0.500	0.094	0.300	450.000	1.000	3.307E+07

Table 2: Results: bellows properties

Nb	r in	D in	A in	t in	bc in	K in lb $g/\%$	KI in lb $g/\%$	I in 4
28	0.073	14.300	0.354	0.026	0.342	2759.212	1.046E+04	3.151E+01

Table 3: Results: maximum bellows stresses at p_{op}

σ_{max} psi	σ_a psi	σ_{pwt} psi	σ_{pwt} psi	σ_{ps}	σ_{max} in
1.523E+05	2.778E+04	6.640E+04	6.782E+04	2.907E+04	1.472E-03

Table 4: Input: bellows misalignments

$\zeta(6)$ in	$\zeta(7)$ in	$\zeta(13)$ in	$\zeta(14)$ in	$\eta(6)$	$\eta(7)$	$\eta(13)$	$\eta(14)$
0.030	0.000	0.000	0.000	0.000	0.000	0.000	0.000

Table 5: Input: bellows initial shape

$d_{i,6}$ in	$d_{i,13}$ in	$d_{i,6}$ in	$d_{i,13}$ in
0.020	0.000	0.020	0.000

Table 6: Input: dipole properties

s in	L_D in	ovh_D in	D_m in	t_m in	I_m in 4	E_m psi
0.200	670.	63.6	10.5	0.188	111.	3.000E+07

Table 7: Input: quadrupole properties

L_q in	ovh_q in	D_q in	t_q in	I_q in 4	E_q psi
210.	52.5	10.5	0.188	111.	3.000E+07

Table 8: Results: bellows maximum elongation at p_{op}

p_{op} psi	$dl6$ in	$dl13$ in
3.000E+02	1.262E+00	1.000E+00

C2.a

Table 9: Results: node location, stiffness, force at p_{op}

node	$x(i)$ in	$k(i)$ in lb $g/\%$	$F(i)$ lbs
1	0.00	25000.00	87.54
2	135.69	25000.00	35.69
3	271.38	25000.00	24.72
4	407.07	25000.00	17.59
5	542.76	25000.00	-12.71
6	678.45	0.00	0.00
7	814.14	0.00	0.00
8	949.83	25000.00	115.28
9	1085.52	25000.00	-13.68
10	1221.21	25000.00	18.37
11	1356.90	25000.00	26.95
12	1492.59	25000.00	37.80
13	1628.28	0.00	0.00
14	1763.97	0.00	0.00
15	1899.66	25000.00	27.45
16	2035.35	25000.00	24.13
17	2171.04	25000.00	22.63
18	2306.73	25000.00	35.16
19	2442.42	25000.00	87.70

Fig.C2.b

MAX BELLOWS LOCAL ELONGATION

SSC dgd :

zet6=0.030 in

zet7=0.000 in

zet13=0.000 in

zet14=0.000 in

et6=0.000 in

et7=0.000 in

et13=0.000 in

et14=0.000 in

ds6=0.020 in

ds13=0.000 in

$f(6)=-12500.0$ lbs

$f(13)= 0.0$ lbs

$K= 2759.2$ lbs/in

Fig.C4

ASSEMBLY DEFLECTIONS AT 303.5 psi

SSC ddd :

zet6=0.030 in

zet7=0.000 in

zet13=0.000 in

zet14=0.000 in

et6=0.000 in

et7=0.000 in

et13=0.000 in

et14=0.000 in

ds6=0.020 in

ds13=0.000 in

$f(6)= 0.0$ lbs

$f(13)= 0.0$ lbs

$K= 2759.2$ lbs/in

Fig.C5

Table 1: Input: bellows design parameters

r_1 in	D_1 in	d in	β	l in	per psi	λ in	E_b psi
0.060	3.875	0.330	0.094	5.750	17.500	0.830	3.297E+07

Table 2: Results: bellows properties

Nb	r in	D in	A in	t in	bc in	K lbs/in	KI lbs/in	I in ⁴
19	0.065	4.205	6.297	0.010	0.303	82.900	6.576E+01	2.767E-01

Table 3: Results: maximum bellows stresses at p_{op}

σ_{max} psi	σ_A psi	σ_{p01} psi	σ_{p02} psi	σ_{p0}	y_{max} in
1.041E+06	0.557E+02	0.690E+03	0.779E+03	3.625E+03	2.573E-04

Table 4: Input: bellows misalignments

$\zeta(6)$ in	$\zeta(7)$ in	$\zeta(13)$ in	$\zeta(14)$ in	$\eta(6)$	$\eta(7)$	$\eta(13)$	$\eta(14)$
0.030	0.000	0.000	0.000	0.000	0.000	0.000	0.000

Table 5: Input: bellows initial shape

$d_{,6}$ in	$d_{,13}$ in	$d_{,6}$ in	$d_{,13}$ in
0.020	0.000	0.020	0.000

Table 6: Input: dipole properties

e in	L_D in	eek_D in	D_m in	t_m in	I_m in ⁴	E_m psi
0.200	670.	63.6	10.5	0.188	111.	3.000E+07

Table 7: Input: quadrupole properties

L_Q in	eek_Q in	D_Q in	t_Q in	I_Q in ⁴	E_Q psi
210.	52.5	10.5	0.188	111.	3.000E+07

Table 8: Results: bellows maximum elongation at p_{op}

p_{op} psi	$d16$ in	$d113$ in
1.167E+01	1.384E+00	8.300E-01

Fig.D2.a

Table 9: Results: node location, stiffness, force at p_{op}

node	$x(i)$ in	$k(i)$ lbs/in	$F(i)$ lbs
1	0.000E+00	1.000E+06	1.597E+00
2	3.000E+00	1.000E+06	-1.330E+01
3	6.000E+00	1.000E+06	5.743E+01
4	9.000E+00	1.000E+06	-2.268E+02
5	1.200E+01	1.000E+06	2.658E+02
6	2.200E+01	0.000E+00	0.000E+00
7	2.845E+01	0.000E+00	0.000E+00
8	3.393E+01	1.000E+06	-7.317E+01
9	3.893E+01	1.000E+06	6.076E+01
10	3.993E+01	1.000E+06	-1.506E+01
11	4.293E+01	1.000E+06	2.705E+00
12	4.593E+01	1.000E+06	3.931E-01
13	5.693E+01	0.000E+00	0.000E+00
14	6.358E+01	0.000E+00	0.000E+00
15	6.706E+01	1.000E+06	2.306E-01
16	7.006E+01	1.000E+06	1.106E-01
17	7.306E+01	1.000E+06	-3.095E-01
18	7.606E+01	1.000E+06	4.103E-01
19	7.906E+01	1.000E+06	-6.577E-01

Fig.D2.b

pipe support, give K , closeup, max loc bel elong

Table 1: Input: bellows design parameters

r_i in	D_i in	d in	β	l in	per psi	λ in	E_b psi
0.060	3.875	0.330	0.894	5.750	450.000	0.830	3.297E+07

Table 2: Results: bellows properties

Nb	r in	D in	A in	t in	b_e in	K lbs/in	KI lbs/in	I_{eq}^4
17	0.072	4.305	0.321	0.824	0.176	2108.561	1.692E+03	6.996E-01

Table 3: Results: maximum bellows stresses at p_{op}

σ_{max} psi	σ_a psi	σ_{p01} psi	σ_{p02} psi	σ_{pe}	y_{max} in
3.310E+08	1.108E+04	3.504E+04	3.596E+04	1.400E+04	4.165E-04

Table 4: Input: bellows misalignments

$\zeta(6)$ in	$\zeta(7)$ in	$\zeta(13)$ in	$\zeta(14)$ in	$\eta(6)$	$\eta(7)$	$\eta(13)$	$\eta(14)$
0.030	0.000	0.000	0.000	0.000	0.000	0.000	0.000

Table 5: Input: bellows initial shape

d_{e6} in	d_{e13} in	d_{e8} in	d_{e13} in
0.020	0.000	0.020	0.000

Table 6: Input: dipole properties

e in	L_D in	ovh_D in	D_m in	t_m in	I_m in ⁴	E_m psi
0.200	870.	63.6	10.5	0.188	111.	3.000E+07

Table 7: Input: quadrupole properties

L_q in	ovh_Q in	D_q in	t_q in	I_q in ⁴	E_q psi
210.	52.5	10.5	0.188	111.	3.000E+07

Table 8: Results: bellows maximum elongation at p_{op}

p_{op} psi	$d16$ in	$d13$ in
3.000E+02	9.829E-01	8.301E-01

Fig.D4.a

Table 9: Results: node location, stiffness, force at p_{op}

node	$x(i)$ in	$k(i)$ lbs/in	$F(i)$ lbs
1	0.000E+00	1.000E+08	7.935E-01
2	3.900E+00	1.000E+08	-4.349E+00
3	0.000E+00	1.000E+08	1.908E+01
4	9.900E+00	1.000E+08	-7.496E+01
5	1.200E+01	1.000E+08	4.596E+01
6	2.200E+01	0.000E+00	0.000E+00
7	2.865E+01	0.000E+00	0.000E+00
8	3.395E+01	1.000E+08	2.032E+02
9	3.893E+01	1.000E+08	-2.371E+02
10	3.993E+01	1.000E+08	0.036E+01
11	4.293E+01	1.000E+08	-1.503E+01
12	4.593E+01	1.000E+08	3.367E+00
13	5.483E+01	0.000E+00	0.000E+00
14	6.268E+01	0.000E+00	0.000E+00
15	6.786E+01	1.000E+08	4.946E-01
16	7.086E+01	1.000E+08	-2.362E-01
17	7.386E+01	1.000E+08	0.159E-02
18	7.686E+01	1.000E+08	0.065E-02
19	7.986E+01	1.000E+08	7.356E-02

Fig.D4.b