Pressurized Rescue Module System Hull and Transfer Skirt Design and Experimental Validation

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Abstract- **The Pressurized Rescue Module System (PRMS) is a remotely operated vessel which is designed to recover crew members from a disabled submarine (DISSUB). It has a crew of two for supporting rescue operations, and can transport up to 16 people. The PRMS is transported to a rescue site at sea, first from its home base by truck, air transport and then by ship; such portability requirements placed significant constraints on the system size, shape and weight. The system is launched from the vessel of opportunity (VOO), where it is remotely piloted to a maximum depth of 2000 feet by topside personnel. Attached to the PRMS is a transfer skirt which mates with the DISSUB and allows for the transfer of personnel from the DISSUB to the PRMS hull. The transfer skirt has an articulated joint that allows the PRMS to mate with the sub at any vertical angle between +/- 45 degrees, and at any yaw angle, thereby eliminating the constraint that a disabled sub lie essentially horizontal on the sea floor. The PRMS hull structure consists of a stiffened cylindrical hull with hemispherical heads and two personnel transfer hatches. The Deck Transfer Hatch (DTL) is located in the forward head and facilitates transferring personnel out of the rescue vehicle while on deck. The Transfer Skirt Hatch (TS) is located in the bottom of the vessel and is where people are brought on board from the disabled sub through the transfer skirt. The total envelope of the PRM hull and exostructure is 2.4 m x 2.4 m x 7.3 m (8 ft x 8 ft x 24 ft), and the envelope for the transfer skirt (which is detached from the hull during shipping) is 2.4 m x 2.4 m x 2.4 m (8 ft x 8 ft x 8 ft).**

Finite Element Analysis (FEA) and closed-form solutions are utilized in the design stage to optimize the vessel strength, size and weight. Key to this analysis was providing safe predictions of the buckling strength, particularly in regions where large openings are present in the hull. The PRM hull has two such openings; the forward and bottom hatches with their attendant reinforcements. *Reduced* buckling strengths for both the ring stiffened cylindrical hull and the hemispherical end closures, are obtained by first calculating a nominal shell buckling pressure assuming no large opening exists. These nominal buckling pressures include out-of-circularity and out-of-fairness effects for the cylindrical hull, and out-of-sphericity effects for the hemispherical and spherical shells, plus fabrication reduction factors. The reduced buckling pressure for the structural component in question is then determined by multiplying the nominal buckling pressure obtained from classical equations by a reduction factor obtained from FEA. The reduction factor is the ratio of peak von Mises stress in a far-field region away from the opening to the peak von Mises stress at the edge of the opening reinforcement. Since the stiffened regions surrounding the large opening are subject to stress concentration, a gradual taper is provided between the thickened regions and nominal shell thickness so as to minimize the stress concentration.

Hydrostatic tests were performed for both the PRM hull and the transfer skirt (separately) to confirm the required minimum buckling strength of 1.5 times the maximum operating pressure. Strain gages were monitored at approximately 200 locations for the PRM hull, and approximately 100 locations for the transfer skirt. Satisfactory correlation was observed between strain gage data and FEA strain predictions, and in both cases, the minimum buckling strength requirements were satisfied using the procedures described above.

This paper provides descriptions of the buckling strength analysis procedures, the finite element analysis results, and correlations with the hydrostatic test data which were used to verify the buckling strength of the PRMS.

I. INTRODUCTION

The Pressurized Rescue Module System (PRMS) is a remotely operated vessel designed to aid rescue operations for a disabled submarine (DISSUB) by offloading crew members from the DISSUB and transporting to the surface vessel of opportunity (VOO). The PRMS is a portable system, capable of being transported by truck, air, and ultimately by sea to the rescue site. The primary structural systems include the PRM hull, which is a tee-stiffened cylindrical hull with hemispherical heads, and the transfer skirt (TS), a spherical shell structure with articulating joints. The adapter spool connects the PRM hull and transfer skirt, and allows rotation (yaw) of the transfer skirt, also through an articulating joint. Both the transfer skirt and adapter spool joints provide flexibility in performing mating operations with the DISSUB by accommodating mating surface orientations of up to 45 degrees relative to the horizontal plane, and any yaw angle. There are two hatches, both designed to open inward, to allow personnel transfer from the DISSUB to the PRM via the transfer skirt, and from the PRM to the deck transfer lock (DTL). The inward-opening hatches significantly reduced the total footprint of the PRMS for shipping and reduce system weight. Fig. 1 shows the PRMS structural components.

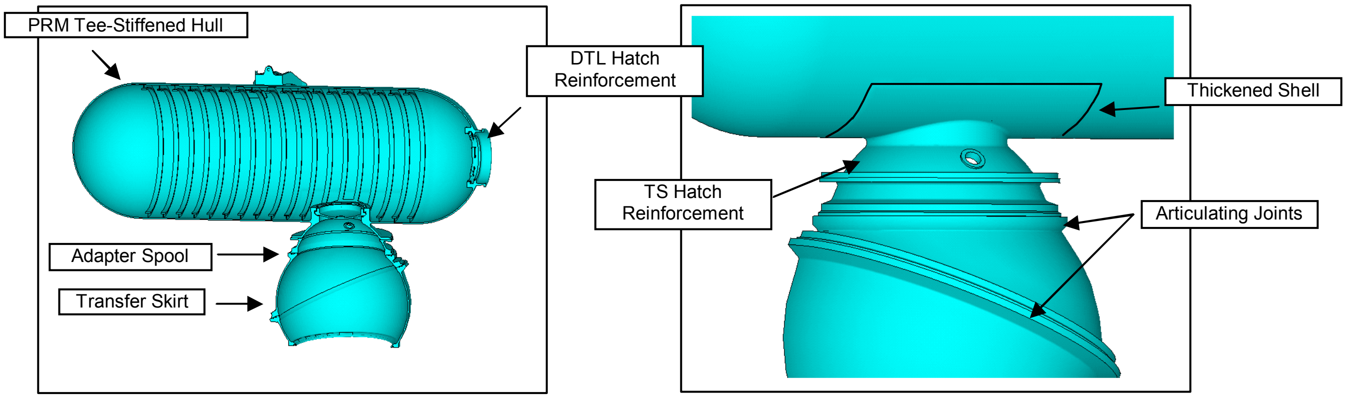

Figure 1. *(left)* PRMS showing tee-stiffened hull with hemispherical heads, transfer skirt, transfer skirt hatch and hatch reinforcement, adapter spool and deck transfer lock opening reinforcement. *(right)* Close-up of transfer skirt, adapter spool and transfer skirt reinforcement plus hull thickened shell region.

The design procedures for the PRM cylindrical hull and hemispherical end closures subjected to external pressure loading are those of the American Bureau of Shipping (ABS) "Rules for Building and Classing Underwater Vehicles, Systems, and Hyperbaric Facilities," 1990 Edition [1], with modifications specified in the contract statement of work, which are hereafter referred to as the "modified ABS rules." A key element of the modified ABS rules pertains to the evaluation of the buckling strength of a structure with large openings. Specifically, a simplified procedure for determining a *reduced* buckling strength was provided which was approved by the US Navy [2]. In this simplified procedure, the nominal buckling strength is evaluated assuming no large openings exist (e.g., using modified ABS Rules), and then reduced by a knock-down factor which is obtained as the ratio of the maximum von Mises stress in the structure without large opening, divided by the maximum von Mises stress at the edge of the opening. This formulation is shown in Eq. 1, where P_{red} is the reduced buckling pressure, P_{nom} is the nominal buckling pressure that would exist without large openings, σ_{trans} is the maximum equivalent stress along the edge of the transition region, and σ_{free} is the equivalent stress in the free field region not affected by the large opening. The knockdown factor K is the ratio of the free field stress to the peak transition region.

$$P_{reduced} = P_{nom}K = P_{nom}\frac{\sigma_{free}}{\sigma_{trans}} \qquad \text{(EQ.1)}$$

For the PRMS, the localized stresses at the edges of the large openings as well as *free-field* stresses in regions not affected by the large opening were obtained through linear/elastic Finite Element Analysis (FEA). This procedure significantly reduces the required computational effort by eliminating the need for a nonlinear buckling analysis while providing a conservative estimate of the buckling strength.

Hydrostatic proof tests were performed to confirm the design analysis and structural integrity. Strain gage data were monitored and compared with FEA strain predictions during proof testing to verify that the structure was performing as expected, and to provide criteria for limiting or terminating a test in the event that measured strains differed significantly from predicted strains. In all tests, it was found that the measured strains were in line with predicted strains; consequently all tests were completed at the maximum load levels. Von Mises stresses were calculated at each of the strain gage locations using measured strains, and comparisons with predicted von Mises stresses are provided below.

II. PRM AND TS HULL FEA MODELS

The PRMS has several locations where large openings affect the nominal buckling strength. These include *(a)* the transfer skirt hatch opening reinforcement in the bottom of the PRM cylindrical hull (see Fig. 1); *(b)* the deck transfer lock hatch opening reinforcement in the forward hemispherical head; *(c)* the flanged regions at the tops and bottoms of the transfer skirt spherical shells. The transfer skirt opening reinforcement is a thick cylindrical shell that intersects the bottom of the PRM hull, cutting through the shell and its stiffeners. It has 8 shear lugs for supporting the transfer skirt hatch in a bayonet-mount configuration. As noted in Fig. 1, there is a *rectangular* region in the PRM hull near the TS reinforcement where the hull thickness has been increased slightly, and it is at the edge of this thickened region where the buckling strength knock-down factor is evaluated.

The DTL hatch reinforcement is shown in Fig. 2. It has a tapered section that ends at the edge of the head spherical shell. The transfer skirt has four general locations (also shown in Fig. 2) where a constant-thickness spherical shell transitions through a tapered section into the four flanged regions.

The nominal buckling strength and knockdown factors were evaluated for each of the three large openings, considering all of the pertinent loading conditions. These included external pressure with the transfer skirt both watered and dewatered, DISSUB mating misalignments, current forces, external impact and drive-off forces, plus both rigid and flexible DISSUB mating surfaces covering the full range of mating orientations. In general, only the transfer skirt hull stresses were sensitive to the non-pressure related forces.

The design maximum depth was 610 m (2,000 ft), which corresponds to a maximum operating pressure of 6.2 MPa (893 psi). With a design safety factor of 1.75, the required buckling strength was 10.8 MPa (1,563 psi). Hydrostatic testing required a safety factor of 1.5, or a test pressure of 9.3 MPa (1,340 psi).

Finite element analysis (FEA) models of the PRM hull, the DTL head and the transfer skirt hulls were developed using the general purpose finite element code ANSYS. Quadratic 3D solid elements were used in a combination of brick, tetrahedral and pyramid shapes, with sufficient element density to provide a mesh-independent solution, and to capture the mild stress concentration at the edge of the respective transition regions. A linear/elastic material model was used to represent the HY-100 steel, with elastic modulus of 207 GPa (30x10^6 psi) and Poisson ratio of 0.3.

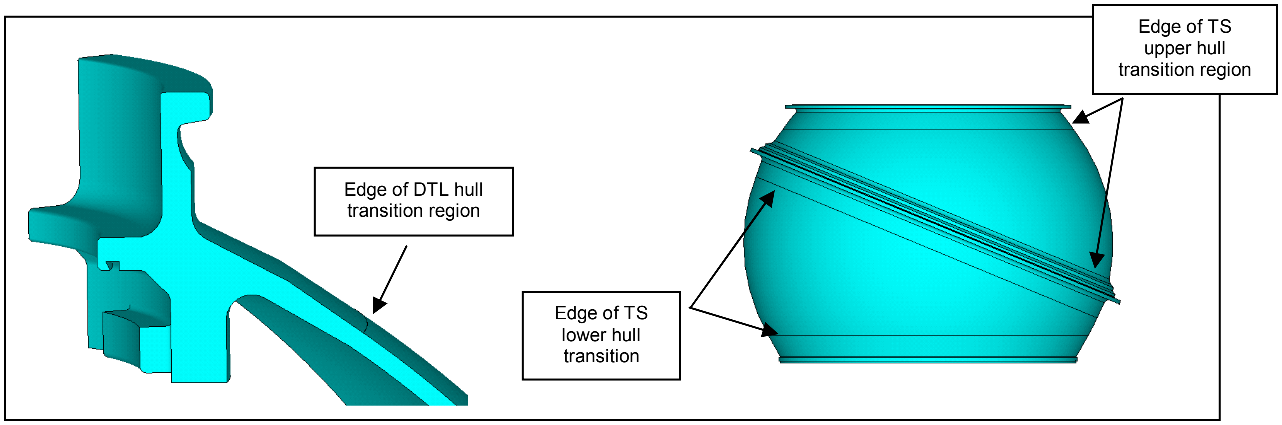

Figure 2. Location of transition regions for DTL *(left)* and TS *(right)* hulls.

A. Nominal Buckling Strengths

The nominal buckling strength of the PRM tee-stiffened hull was found to be 12.1 MPa (1,747 psi) per the modified ABS rules. This includes an allowance of 0.31% out-of-roundness for uncut stiffeners, and 0.42% for cut stiffeners. The corrosion allowance is zero. The DTL hatch opening reinforcement nominal buckling strength is 12.6 MPa (1,833 psi), based on a maximum out-of-sphericity of 0.3%, and usage factor of 0.67. This is also based on the modified ABS rules. The TS hull nominal buckling strength was evaluated using the empirical formulation presented by Krenzke and Kiernan [3], which utilizes a 03% out-of-sphericity plus a fabrication reduction factor of 0.98. For this case, the nominal buckling strength was found to be 22.7 MPa (3,298 psi) for the upper hull (20.3 mm thickness) and 23.5 MPa (3,405 psi) for the lower hull of 21mm thickness. These values are summarized in Tables I and II.

B. Transition Region and Free Field Stresses

The von Mises equivalent stresses along the OD and ID edges of the thickened region of the PRM hull near the TS opening reinforcement vary along the perimeter of this region, and the peak values are presented in Tables I and II. Similar results are shown for the DTL hatch opening reinforcement plus the upper and lower TS hulls. The free field stresses are also shown in Tables I and II, and it is noted that the for the stiffened hull model, the free field stress was taken as the maximum midbay stress.

The maximum transition region stress for the TS hull analyses was taken as the maximum equivalent stress for any location around the entire circumference of the transition region.

C. *Reduced Buckling Strengths*

The knockdown factors shown in Tables I and II were used to evaluate the reduced buckling strengths for the PRM hull (TS hatch opening reinforcement region), the DTL hatch opening reinforcement, and the upper and lower TS hulls. The required buckling strength of 10.8 MPa was met in all cases except for the PRM hull, where it was 10.7 MPa. Given the conservatism in the modified ABS rules for evaluating the nominal buckling strength, this value was deemed to be acceptable, and it was concluded that the PRMS had sufficient strength to resist the maximum design loads without buckling. This was confirmed via hydrostatic testing at pressures up to 1.5 times the maximum operating pressure. Further, it was found that the most conservative estimate of TS hull buckling strength was obtained with the transfer skirt mated to a base that assumed to be rigid base.

TABLE I
NOMINAL AND REDUCED BUCKLING STRENGTHS BASED ON KNOCKDOWN FACTORS – EXTERNAL SURFACES

Component	**Nominal Buckling Strength** *(MPa)*	**Equivalent Stress** *(MPa)*		**Knockdown Factor**	**Reduced Buckling Strength** *(MPa)*
		OD Edge of Transition Region	*OD Free Field*		
PRM Hull	12.1	*0.509*	*0.449*	0.88	10.7
DTL Hatch Opening Reinforcement	12.6	*0.214*	*0.293*	1.37 (*)	12.6 (*)
TS Upper Hull	22.7	*0.210*	*0.141*	0.67	15.2
TS Lower Hull	23.5	*0.190*	*0.136*	0.71	16.7

(*) The calculated knockdown factor for the DTL hatch opening reinforcement exceeded 1.0 for the external surface; therefore the reduced buckling strength was taken to be equal nominal buckling strength.

TABLE II
NOMINAL AND REDUCED BUCKLING STRENGTHS BASED ON KNOCKDOWN FACTORS – INTERNAL SURFACES

Component	**Nominal Buckling Strength** *(MPa)*	**Equivalent Stress** *(MPa)*		**Knockdown Factor**	**Reduced Buckling Strength** *(MPa)*
		ID Edge of Transition Region	*ID Free Field*		
PRM Hull	12.1	*66.1*	*65.1*	0.98	11.9
DTL Hatch Opening Reinforcement	12.6	*51.5*	*44.5*	0.86	10.8
TS Upper Hull	22.7	*29.3*	*21.7*	0.74	16.8
TS Lower Hull	23.5	*27.2*	*21.1*	0.78	18.3

III. EXPERIMENTAL VALIDATION

As noted above, instrumented hydrostatic tests were performed to confirm design adequacy by subjecting the PRM hull and TS hulls to 1.5 times the maximum operating pressure, or 9.3 MPa. Although strain gage data were recorded for both tests, the purpose for monitoring these data was to confirm that the PRM and TS hulls were performing as anticipated so that the testing could be interrupted if the strain/deformation exceeded what was considered safe levels. FEA strain predictions were used to specify the *safe* levels for each strain gage location. Since the full PRM with transfer skirt and DISSUB mating flange could not fit inside the hydrostatic test vessel, two sets of tests were performed; *(a)* the PRM hull with a *blanking dome* attached to the bottom of the transfer skirt opening reinforcement (see Fig. 3) and *(b)* the transfer skirt with a *blanking dome* attached to the top of the adapter spool and a 203-mm blanking plate representing a DISSUB mating flange (see Fig. 4), so it was necessary to prepare FEA models of both these configurations. FEA predictions were obtained that simulate the TS hatch open and closed. This section summarizes the FEA predictions and test measurement results.

D. PRM Hull Hydrostatic Test

Approximately 200 strain gages were attached to the PRM hull in biaxial and triaxial rosette configurations. The strain data were monitored during hydrostatic testing up to a maximum external pressure of 1.5 times the maximum operating pressure, which was 9.3 MPa. Equivalent stresses were obtained for each location in order to compare with predicted equivalent stresses. Table III shows stress comparisons for the test locations near the TS hatch opening reinforcement (thickened section), the corresponding free-field stress estimates, plus the DTL hatch opening reinforcement transition and free-field regions, and good agreement is observed. In nearly all cases, the predicted stress at 9.3 MPa external pressure was within 10% of the values obtained from the strain gage measurement; only three had relative errors exceeding 5%. For the instance where the prediction error exceeded 10%, the FEA equivalent stress was higher than the *measured* stress. The favorable agreement between analytical and experimental results provides validation of the FEA knockdown factors used to estimate the PRM hull buckling strength. Further, since the PRM hull in fact did not buckle under the full hydrostatic test loads, it is concluded that the reduced buckling strengths calculated above are conservative estimates of the true hull buckling strength.

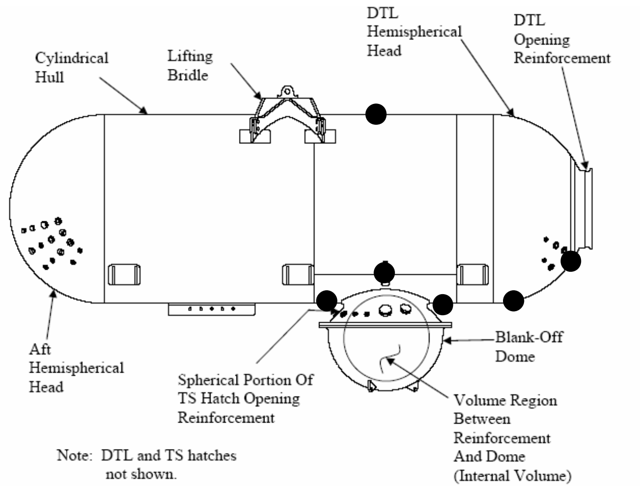

Fig. 3. PRM hull hydrostatic test configuration. The solid dots identify locations where comparisons are made in Table III.

E. TS Hull Hydrostatic Test

Approximately 100 strain gages were attached to the TS hull in triaxial rosette configurations. The strain data were monitored during hydrostatic testing up to a maximum external pressure of 1.5 times the maximum operating pressure, which was 9.3 MPa. Equivalent stresses were obtained for each location in order to compare with predicted equivalent stresses. Table IV shows stress comparisons for the test for the transfer skirt shell transition regions plus free-field, and for the most part, good agreement is observed. It is noted that while the relative error for locations 11 and 19 is high, the magnitude of the stresses and strains is low, which means the absolute error is low. Location 25 has a relative error of nearly 30%, where the predicted stress/strain is *less* than the measured values. While there are several plausible reasons for why this error occurs (such as stiction forces there were not modeled at the rotary joint between hulls), no definitive explanation has been set forth to date since the overall stress/strain levels during hydrostatic testing were low. The remaining analytical/experimental errors for the transfer skirt hydrostatic tests were less than 10%. And like the PRM hull, the fact that the transfer skirt did not buckle under the full hydrostatic test loads, it is concluded that the reduced buckling strength is a conservative estimate of the true transfer skirt buckling strength.

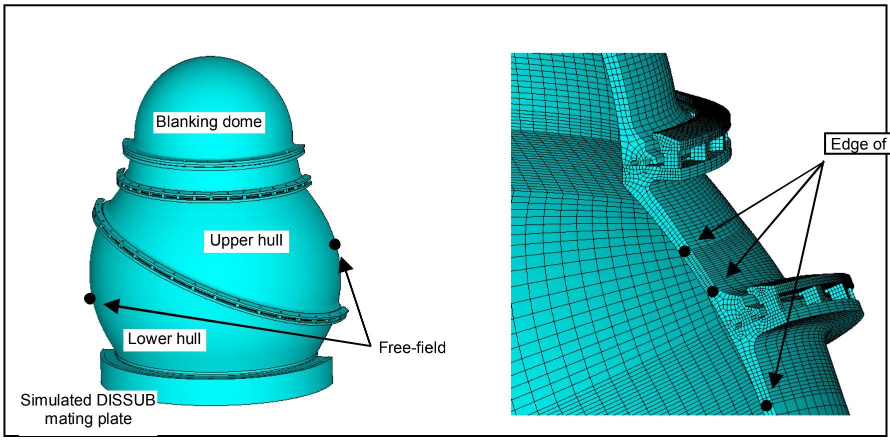

Fig.4. FEA model of hydrostatic test configuration for transfer skirt. (*left)* Transfer skirt with blanking dome and mating plate. (*right)* FEA mesh of upper/lower TS hull interface region. Dots correspond to edges of thickness transition regions and free-field regions where stress/strain comparisons are made.

Table III
Comparison of predicted and experimental strains and stresses for TS hatch opening reinforcement and DTL hatch opening reinforcement.

	Gage	**Strain** *(με)*		**Stress**	**Stress** *(MPa)*		
Location	**Direction**	*Experimental*	*Predicted*	**Type**	*Experimental*	*Predicted*	*Error*
3 Hull Free field OD	circ	-1635	-1687				
	long	-1342	-1185	σeqv	442	430	-2.7%
41 Hull Free field ID	circ	-1675	-1732				
	long	-229	-266	σeqv	345	358	3.8%
19 Hull Bottom (FWD,OD)	circ	-1260	-1387				
	long	-1285	-1289	σeqv	376	396	5.2%
20 Hull Bottom (AFT, OD)	circ	-1388	-1379				
	long	-1441	-1348	σeqv	418	403	-3.6%
70 Hull Bottom (FWD,ID)	circ	-1362	-1399				
	long	-514	-419	σeqv	301	301	-0.1%
71 Hull Bottom (AFT, ID)	circ	-1368	-1388				
	long	-375	-455	σeqv	292	301	3.3%
25 Hull Side Transition OD	circ	-900	-1714	σ1	-328	-349	
	45	-1328	-1367	σ3	-473	-459	
	long	-1809	-1020	σeqv	419	415	-1.0%
80 Hull Side Transition ID	circ	-1566	-1775	σ1	-160	-159	
	45	-884	-971	σ3	-372	-415	
	long	-233	-167	σeqv	323	362	12.2%
7 Head Free field OD	circ	-883	-864	σ1	-252	-255	
	45	-850	-864	σ3	-259	-255	
	long	-848	-864	σeqv	256	255	-0.2%
9 Head ID	circ	-1574	-1715				
	long	-1191	-1248	σeqv	412	442	7.4%
8 Head OD	circ	-553	-800	σ1	-184	-187	
	45	-661	-693	σ3	-236	-222	
	long	-868	-585	σeqv	215	207	-3.8%
29 Head Free field ID	circ	-951	-939	σ1	-270	-277	
	45	-921	-939	σ3	-278	-277	
	long	-903	-939	σeqv	274	277	1.3%

Table IV
Comparison of predicted and experimental strains and stresses for the TS upper and lower hulls.

	Gage	**Strain** *(με)*		**Stress**	**Stress** *(MPa)*		
Location	**Direction**	*Experimental*	*Predicted*	**Type**	*Experimental*	*Predicted*	*Error*
1 TS Upper Hull Top OD	circ	-1296	-1282	σ1	-199	-220	
	45	-813	-933	σ3	-329	-331	
	long	-491	-585	σeqv	287	292	1.7%
2 TS Upper Hull Mid OD	circ	-716	-719	σ1	-191	-214	
	45	-645	-734	σ3	-208	-219	
	long	-635	-749	σeqv	200	217	8.4%
3 TS Upper Hull Bottom OD	circ	-244	-450	σ1	-137	-174	
	45	-691	-749	σ3	-290	-269	
	long	-1204	-1047	σeqv	252	236	-6.2%
18 TS Upper Hull Top ID	circ	-1154	-1257	σ1	-218	-227	
	45	-794	-939	σ3	-308	-328	
	long	-627	-621	σeqv	274	291	6.1%
19 TS Upper Hull Bottom ID	circ	-403	-459	σ1	-116	-159	
	45	-381	-634	σ3	-123	-215	
	long	-403	-809	σeqv	119	193	62.2%
11 TS Mating Hull Top OD	circ	-529	-555	σ1	-135	-191	
	45	-416	-755	σ3	-166	-255	
	long	-492	-955	σeqv	153	230	50.0%
12 TS Mating Hull Mid OD	circ	-743	-672	σ1	-204	-204	
	45	-684	-712	σ3	-217	-217	
	long	-680	-752	σeqv	211	211	0.1%
13 TS Mating Hull Bottom OD	circ	-1183	-1134	σ1	-202	-214	
	45	-814	-868	σ3	-306	-299	
	long	-536	-601	σeqv	269	267	-1.0%
25 TS Mating Hull Top ID	circ	-567	-584	σ1	-203	-178	
	45	-823	-622	σ3	-286	-190	
	long	-1090	-659	σeqv	255	184	-27.9%
26 TS Mating Hull Bottom ID	circ	-1053	-1125	σ1	-265	-224	
	45	-919	-888	σ3	-298	-300	
	long	-852	-650	σeqv	283	270	-4.5%

IV. CONCLUSIONS

The reduced buckling strength evaluation is a simplified technique that provides conservative strength estimates using linear elastic FEA models. This significantly reduces the computational and sometimes experimental effort necessary to establish a conservative buckling strength. There is a tradeoff in using this technique, and that is the degree of conservatism is not known. Furthermore, it depends on the extent to which the estimate of the nominal buckling strength is conservative. For example, the ABS rules have an inherent, built-in safety factor, so that a design to 100% capacity will in fact have an unknown residual capacity. Additional studies that compare design margins using this approach with other standard methods of predicting buckling strength would be beneficial.

The design of the PRMS structural components for buckling using the *reduced* buckling procedures describe herein did in fact produce a safe design as evidenced by successful completion of the hydrostatic tests at full load levels. Strain gage data collected during these tests compared favorably with FEA predictions. This provides additional justification for using FEA results for evaluating the knockdown factors and overall buckling strength of thin shell structures with large penetrations.

References

[1] American Bureau of Shipping, *Rules for Building and Classing Underwater Vehicles, Systems and Hyperbaric Facilities,* 1990 Edition.
[2] Request For Information (RFI) SRI-007-2A1-0144, Pressurized Rescue Module System Contract N00024-00-C-4010, August 2002.
[3] M.A. Krenzke and T.J. Kiernan, “The effect of initial imperfections on the collapse strength of spherical shells,” David Taylor Model Basin, Bureau of Ships, 1965.