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THE DOWN RATCHET

AND

THE DETERIORATION OF TANKER NEWBUILDING STANDARDS

by
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ABSTRACT

For the last 20 years, Hellespont Shipping has operated six ULCC's built in the mid 1970's. In 1999, Hellespont embarked on an eight ship newbuilding program in Korea: four 300,000 ton VLCC's and four 440,000 ton ULCC's. These are the first ships that Hellespont has built in over 25 years. These ships are required to meet both ABS and LR rules. Unfortunately, we found a number of areas where the Classification rules and yard standards have deteriorated drastically since the 70's, in some cases to imprudent levels. This paper outlines these areas and recommends corrective action. We argue that the primary cause for this deterioration is the "direct computation down ratchet". In the absence of a meaningful guarantee, the yards are constantly and cleverly searching for ways to shave the rules. Whenever any yard gets a relaxation in the rules, however risky, this new lower level becomes the standard. The class that approved the relaxation can't admit it was imprudent for both legal and commercial reasons. Other Classes feel they have to fall in line lest their ships become uncompetitively expensive. The rules ratchet down one level. The yards then compete away the saving and the process repeats itself. The individual changes generated by this process may not be large, but the cumulative effect can be massive. Over time, the yards are rewriting the rules, not Class Technical Committees. Examples supporting this position are offered.

1. GET RID OF UNREASONABLE AND INFLEXIBLE OPERATING RESTRICTIONS

It seems self-evident that a ship should be designed to handle all the operating conditions that it could reasonably be expected to face in normal commercial practice with a healthy margin for operator error. Current Class tanker rules use a different philosophy. The ship is designed to handle only the loading conditions in the Trim and Stability Booklet and then just barely. Just about everything else is implicitly illegal. The yard's job is to make sure these conditions are as narrow as possible to save a bit of steel. Class doesn't seem to care as long as these restrictions show up somewhere in the paper work.[1] We replace steel with restrictions. Don't want to put steel in the bow. No problem, just put a line in the Loading Manual that says the ship can't operate below x draft forward, can't operate with the forepeak tank empty, etc. Want to reduce the hogging moment allowable. No problem, tell the crew they can't use all the ballast tanks even when the cargo tanks are empty. Combine that with a restriction on draft forward, and you've got a ship which has absolutely no flexibility on the ballast leg. Need to inspect a ballast tank. Forget it. There's only one ballast configuration and it's the one that stresses the structure right up against the limit. Captain wants to get the ship a little deeper in the water in bad weather. Forget it, unless he wants to risk all sorts of problems for violating MARPOL.

When you talk to Class people about this problem, they invariably take the position that "It's not Class' fault. The owner accepted the conditions, so it's his fault."[2] But this is the same thing as

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[1] This philosophy was taken to particularly ridiculous heights in the case of the one-across tankers where the Mate was given a set of stability restrictions which couldn't in reasonable commercial practice be followed, resulting in a spate of lolling casualties.

[2] The most damning twist on this argument we know about is the ABS cargo tank density restriction. In 1997, ABS was falling behind in the race to see who could approve the flimsiest ship. The lightweight of a rule minimum ABS VLCC had become 500 to 800 tons more than that of a rule minimum DNV or LR VLCC. The owners were all going to LR and DNV in order to save a paltry $25,000 to $50,000 on an 80 million dollar ship. ABS felt it had to do something to "compete". The solution was to change the design liquid density for the purposes of sloshing force calculations in the cargo tanks from 1.025 (sea water) to 0.90. This change was not widely promulgated. Not only did this mean that the owner could not legally load many crudes and heavy petroleum products in his cargo tanks, it meant he couldn't put ballast in any of the cargo tanks including the tank that was labeled gale ballast. Many owners were not even aware of this new restriction.

But when Hellespont raised this with ABS personnel, we actually got the "if the owner want to put anything heavier in the cargo tanks, he has to tell us" argument. The idea that the owner has to tell class that he wants to be able to put ballast in the gale ballast tank was so patently ridiculous that ABS backed off from this position recently. And did something even worse. The 0.90 is now buried in the rules as a "calibration factor". The owner can now legally put sea water in his gale ballast tank, even though the tank is only designed for a 0.9 density liquid.

Another problem with the "owner accepted it" argument is that usually these restrictions don't show up explicitly in the Contract or Spec. The Spec does not say the "forward draft must always be more than x". The Spec says "Trim and Stability will be evaluated for the following conditions". The Owner has to guess what this implies in terms of restrictions before he's even seen any calculations.

The latest twist on this game is the yard's use of the IMO visibility requirement to build a still weaker ship. The yard argues that, since low draft forward would result in a longer than legal blind zone, the structure does not have to be designed for such illegal conditions. Talk about unintended consequences.

saying, Class have stopped being a regulatory body. Class originally was the creature of the insurers who recognized from hard experience that, left to their own devices, owners would take bad risks and the underwriters were the ones who were going to pay. (The crews paid too but they are riffraff or wogs, so who cares.) The whole idea of Class or any tanker regulation for that matter is to prevent owners from being imprudent. In the modern world, where more than just insurers and crews pay for owner imprudence, the owners must be regulated. If Class doesn't want to impose reasonable design conditions, somebody else will step in.

The Rules should allow:

(1) any transverse combination of cargo tanks across to be empty at or near design draft,

(2) a reasonable range of asymmetric cargo loads at full draft,

(3) all ballast tanks to be 100% full for a full range of bunkers,

(4) the ship to ballast down to a reasonably deep draft without resorting to ballasting cargo tanks.

(5) any single ballast tank to be empty with all the other ballast tanks full,

(6) normal ballast exchange (not flow through) sequence without restriction on weather.

If these reasonable requirements are imposed, you will find that you need a hogging moment allowable of about 1 million ton-m for a standard sized VLCC and a shear force allowable of around 30,000 tons. A rules minimum VLCC will have a hogging moment allowable of 620,000 ton-m and a shear force envelope which maxes out at under 20,000 tons. The extra hogging moment will cost about 600 tons ($300,000) and the extra shear force can be obtained largely by making the longitudinal bulkheads HTS (without reducing the thickness) at a cost of about $66,000.

2. ENFORCE ABILITY TO HANDLE REASONABLY LIKELY CASUALTIES

Not only are Rules ships designed to handle only a unrealistically restrictive range of operating conditions, they are not designed to handle even the most likely damage scenario. Double hull tankers have a glaring weakness: their exposure to flooding of the ballast tanks when loaded. The idea of the double hull is that it's a buffer between the sea and the cargo. Whatever the merits of this idea, if it has any real purpose, you expect to use this buffer from time to time. Once the tanker fleet is converted to double hulls, it is nearly guaranteed that there will be multiple such floodings per year. Yet the Rules ignore this risk.

It is nuts that IMO Reg 25 and IMO Reg 16 Raking go to great length to require that the ship be able to withstand a whole range of flooding scenarios from a stability point of view, many of which scenarios the ship cannot withstand structurally.

The Rules should require that any double hull tanker withstand the flooding of any single ballast tank and any single contiguous pair, trio and quartet of ballast tanks when loaded to scantling draft in the design wave without going plastic. For a standard sized VLCC, this will result in a sagging moment allowable of just over 1,000,000 ton-m. The IACS required sagging moment allowable for this ship is about 620,000 ton-m.

3. CHECK THE DESIGN LIGHTWEIGHT CURVE AGAINST THE BLOCK WEIGHTS

An important determinate of longitudinal forces is the lightweight distribution. Class takes the Sergeant Schultz approach toward the lightweight curve: it knows nothing. Not surprisingly, the yards have learned how to take advantage of this to build a weaker ship. On our ULCC, the yard modeled all the steel weight with just five trapezoids -- standard practice we were told. The inclining experiment revealed that the actual lightweight was 1,200 tons (about 2%) higher and a remarkable 0.7 m further forward than the design lightweight curve said it should be. The yard made an arbitrary adjustment to this clearly erroneous lightweight curve which added the weight and moved the LCG without affecting the all-important hogging moment. This was accepted by both ABS and LR without any check.

In fact, the yards know exactly where all the lightweight is. They carefully calculate the weight and centroid of each of the 250 or so blocks that make up a big tanker. They must do this in order to lift and handle the blocks safely. But these calculations have no impact on the lightweight curve that is used by Class. When we used the block weights and centers to generate the lightweight curve on our ULCC, we found that it increased the critical hogging moment by 4%. The yards take advantage of the lack of Class oversight to fraudulently move the lightweight toward midships since they have figured out that the decrease in hogging moment saves them more steel than the increase in sagging moment adds.

When I asked one ABS executive why Class allowed the yards to do this, he said "We don’t allow it, but we don’t check it." If you can understand this distinction, then you’ve gone a long way toward understanding the Classification Society approach to tanker newbuilding standards.

4. IMPLEMENT MUCH BETTER FINITE MODEL EXTENT AND MESH SIZE

Once we have a reasonably broad and conservative set of design conditions, we have to translate the resulting loads into actual structure. The main tool we have for this purpose is finite element (FE) analysis. One problem with this is that FE model extent and mesh size that is required by Class is hopelessly antiquated.

When Finite Element first became feasible 15 or so years ago, computational resources placed a severe restriction on the effort. Because of the computer constraints, Class elected to go with a model that was limited to two or three midships tanks -- usually one side only -- and a mesh size that was one web frame longitudinally (about 5 m) and three or four stiffeners girth-wise. This may have been a reasonable decision 15 years ago but it resulted in an extremely limited tool:

(1) A tool that could give only a very gross picture of the stresses in the mid body completely ignoring many critical "details" such as web and stringer corners.

(2) A tool that required a great deal of judgment in applying boundary conditions, especially at the fore and aft ends.

(3) A tool that could not model all sorts of interesting load conditions, including ballast exchange.

(4) A tool that said nothing at all about the forebody and the aftbody nor the connections between the middle of the ship and the ends.

(1) was addressed by a second layer of FE models which used the results of the first to obtain boundary conditions for much more detailed models which analyzed a single web ring or stringer. Given the limits of the computer in the early 80's, this was reasonable but the process of converting the Phase I model deflections into Phase II model boundary conditions was labor intensive, error-prone, and required a great deal of judgment. One problem with all these judgment calls is that they are subject to commercial pressures (see below). More basically, the process inevitably introduced errors and uncertainty into the analysis.

But a far more fundamental limitation of this approach is that many important problems simply cannot be analyzed at all. Ballast exchange which involves asymmetric loading of tanks extending the full length of the cargo area is an obvious example. At some point in the ballast exchange operation, we must empty the 1S, 3P, and 5S ballast tanks? This ballast exchange condition typically involves 5 degree heel and an overall level of torsional stress that the yards estimate is around 15% yield. Nobody to our knowledge knows how these torsional stresses are going to be distributed, nor how they combine with all the other stresses because we don't have a proper FEM. The standard model required by class hasn't a clue.

Still more importantly, the forebody and aftbody were ignored completely. The forebody and aftbody are at least as critical as the midbody. These are areas where we see far more problems than the midbody. The forebody is subject to the toughest external loads of any portion of the ship. Deflection in the aft body is critical to the all-important shaft reliability issue. The standard class FE model simply can't address these issues.

It is no longer necessary to accept the constraints of this approach. Nor has it been for some time. Computers are a 100,000 times more powerful than they were when FE was introduced into the Rules. The only proper model extent is the full ship. The proper mesh size girthwise is every stiffener (about 1 meter). The proper mesh size longitudinally is every frame, except where the frame spacing is more than three or four meters, it should be every half frame, but in way of the stringers it should be every quarter frame.[3] For our new ULCC, Hellespont belatedly developed a model that almost met this spec. It has about 300,000 nodes. It takes about two hours to solve a load case on a PC costing less than $2,500. Such a model is not only now computationally feasible; it's dirt cheap.

Class should require that all new tanker designs be modeled to the above level of detail. Paradoxically, by eliminating all the Phase II work, this will probably reduce rather than increase design cost.

5. INTERPRET FINITE ELEMENT RESULTS CONSERVATIVELY

Another problem with Class's use of finite element is the way the results are interpreted. Sometimes our philosophy seems to be: if the stresses come out low, reduce scantlings; if the stress comes out high, it's an artifact of the model. Before finite element analysis came along, naval architects were acutely aware of the fact that they couldn't predict stress with any degree of accuracy. Therefore, for the most part they adopted conservative practice, used upper bound estimates of stress, and were careful not to move very far away from established practice.

But even so we made some mistakes, and came up with some very marginal ships. Under severe economic pressure from the Japanese, the European yards pushed the envelope and made a number of big tankers in the mid-70's that had systemic structural problems. We've had some of these ships. Some of the failures were fatigue problems but others were more general including systemic cracking in the upper web corners, stringer buckling, etc.

When the first pictures of finite analysis of tanker structure became available in the early 80's, we were blown away. The FE models generate pictures which show stress level in colors: cool colors for low stress areas, and warm colors for the high stress areas. It was amazing; all the green and blue were in areas that had given us no problems; and all the problem areas were yellow and orange. In the usual color scheme, stresses above the legal level are red and stresses just below the legal level are orange. For the first time, we could see the stress flow and understand why we had failures in the corners of the upper webs and problems in the stringer toes. That's where almost all the yellow and orange were. It was obvious that this was a great tool. Now that we knew where all the yellow and orange areas are it would be an easy matter to make them green and blue. In fact, that was ABS's original idea: use FE only to increase scantlings, and not allow any decreases.

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[3] One can reasonably argue it should be every quarter frame everywhere in the cargo tank length to keep the element aspect ratio nearly square.

But that's not how finite element ended up being used. Instead of making all the orange, green and blue; the industry used finite element to make the whole structure orange. This is known as structural optimization. And the yards became very good at optimizing structure.[4] In 1999, when Hellespont went to the yards and told them that we wanted to reduce the maximum design stress by 10% (roughly make the ship yellow), they found that they had to increase the steel weight by about 7%. In other words, 70% of the structure was in the orange.

This is a prescription for disaster. The models simply aren't that good. Even the fine mesh recommended above leaves out all sorts of important details. And we don't know the loads or stresses that well. Anybody who thinks so should watch the yards during block fit-up. Often this requires a whole series of jacks and wedges and come-alongs. The induced deflection is far more than occurs in any design case. God knows what the residual stresses are. Two years ago an ABS VLCC suffered extensive stringer buckling during the stagger test. This was before even leaving the yard. The failure was blamed on moving an access ladder hole from one location to another without redesigning the stringer. But if the structure had been anywhere near robust enough to go to sea, a minor change like this -- locally compensated -- should have had no effect. When you go to sea you need margins, and we don't have those margins.

Steel is cheap. The marginal cost to the yard of increasing a scantling is a good deal less than $500 per ton. If we increase the steel weight of a VLCC by 10% in an efficient manner, we will get a far more robust ship at a cost of about two million dollars, a little over 2%. That's intelligent regulation. Class should adopt a more conservative design criteria. We recommend an average reduction in design stress of about 10%.

This will get us back to the good ships of the mid-70's, which by the way were not over-built. We see evidence of this on even the best of our mid-70's built ULCC's. Almost all these ships developed some cracks by age 15. In the "good" ships, these cracks are limited to a handful of localized areas. The owners eventually learn where these areas are and expect to have to repair a crack or two in these areas every docking. The mid-70's ships were much weaker than earlier generations. A 40,000 ton tanker built in the late 50's had a bottom plate thickness of about 35 mm. A very good mid-70's 400,000 ton ULCC -- ten times larger -- will have a bottom plate thickness of 28 to 30 mm.

And if we are going to depend on finite element, then we should use the numbers that the FE models generate. There should be no averaging of stress across elements nor any rounding down of scantlings. In absolutely no case, should the design stress be more than yield.

[4] Way too good. A structure can be meshed any number of different ways. The yards are experts at coming up with the model that minimizes calculated stress. And if they can't get the stress down to the number they want, then they go running to Class and ask that element stresses be averaged or in some cases be simply ignored as a model artifact. They never come running to Class pointing out the stresses look suspiciously low.

6. DON'T USE FINITE ELEMENT AS A SUBSTITUTE FOR GOOD STRUCTURAL DESIGN

There is an even more basic and pernicious problem with our current use of Finite Element Analysis. Before FEA came along, naval architects had to stick to structural concepts that were easy to analyze with the limited tools available. These structures tended to be simple and elegant. At their best, the structure flowed in a regular series of continuous rings -- transverse, horizontal and longitudinal. Discontinuities, sharp corners, small radii were studiously avoided for everybody knew they generated high stress concentrations but nobody pretended to know how high.

With the advent of finite element, a new philosophy emerged. At some yards, it's called "design for producability". The idea is to stick together flat plates of steel in whatever way best suits the production process, and then use FEA (aggressively interpreted, of course) to beef up the scantlings in the high stress areas that result. The result is complex and hard to follow stress patterns, small inserts of very thick plate, myriad discontinuities and sharp corners, plate thicknesses bouncing all over the place -- plug ugly structure which violates the fundamental rules of good design. As the Class rules become more and more FEA based, surveyors become more and more helpless to resist this development.

It's far from obvious that the yards gain much from this development. Elegant, simple structures tend to be the most efficient. But it's clear that owners, underwriters and eventually society as a whole lose big time. One possible solution is to separate structural approval into two stages. In the first stage the structural *concept* would be approved. This would require a series of simple drawings showing only the major structural members for the entire ship (not just the parallel mid-body) but no scantlings. Once the structural concept has been approved, then FEA can be applied (conservatively interpreted, we'd hope) to determine the scantlings. If we can't enforce decent structural design, then we must limit the size and frequency of scantling changes.

7. FATIGUE LIFE SHOULD BE AT LEAST DOUBLED AND APPLIED CONSERVATIVELY

Finite Element analysis even if carried out in the manner we suggest would still not examine the connection details. Prompted by an epidemic of fatigue cracking in 1980's built tankers, class introduced "fatigue analysis" to check these areas. Given the societies' inability to enforce qualitative standards, they probably had no choice. Fatigue analysis can in theory help to improve the robustness of connection details, such as bracket toes, stiffeners to web connections, etc. But the practice leaves much to be desired.

Prior to about 1985, there was no fatigue analysis. Designers would select connections and bracket details that were known to be robust and, if they didn't, Class would require improvement. The goal was to eliminate failure due to fatigue rather than to increase fatigue life. Thanks to advances in computer power, it became possible to consider simulated voyage trading patterns, the resultant cyclic loadings and their application to the transfer functions of the proposed structure connections. A number of heroic assumptions are required in this process, but the end result is a mean time to failure, called the fatigue life.

The rule standard fatigue life is 20 years. Most tankers operate for 25 years or more. It has always puzzled us as to why Class would set an average time to failure less than the expected ship life. When Hellespont specified a 40 year fatigue life, some class software had to be recoded to accept the larger number. But an equally pressing problem is that fatigue life depends critically upon the trading pattern that was implicitly included in the specification. According to both LR and ABS, a ship trading in the North Atlantic has a very different fatigue life than one trading in Indian Ocean. In this case, the class software is almost certainly correct, at least in a qualitative sense. The most graphic example of this is the American flag tankers trading on the Alaska-West Coast route. In this very severe environment, some of these ships turned out to have fatigue lives of one or two trips. Closer to home for Hellespont, if a new VLCC trades Mongstad to Philadelphia rather than Ras Tanura-Yokohama, it's fatigue life is halved. The nominal 20 year life become 10.

This whole approach assumes that our knowledge of ocean sea spectra throughout the world is so complete that we can fine tune our structures right up to the point of planning for an average time to fail that is well within the expected life of the ship. The transfer functions themselves are gouges that are based primarily on offshore oil industry studies of generic joints and brackets and then extrapolated to actual ship details in a semi-judgemental manner. The mechanism as to how sea state spectra is converted to wave forces and then to shear and bending moments on the hull and then transferred via primary and secondary structural members to the connections is at best an idealization. Loading and discharging loads are completely ignored despite the fact that this cycling is implicated in the early cracking seen in North Sea shuttle tankers.[5] There is no consideration of ballast exchange and its effect on the loading of the structure.

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[5] In fact, the rules add port time to the fatigue life. In a 10 voyage year with 5 days port time per voyage, the naively anticipated fatigue life of 20 years is actually 315 days x 20 or 17.3 years.

Our whole approach to fatigue analysis is overly aggressive. It seems as if every time we have a choice between a conservative assumption and an unconservative, we choose the unconservative. This is probably the result of inter-class competition. In any event, the ships are still cracking early in their lives. Given all the uncertainties a far more conservative approach is the only prudent alternative. At a minimum Class should require a mean time to failure, well in excess of the ship's expected life, at least 40 years. More importantly, we should run the voyage simulation over all reasonable trading patterns and design to the resulting fatigue envelope rather than some average. Given intelligent design, the increase in overall ship price will be barely noticeable.

8. NO TANKS AT RESONANCE EVER, FORCE MORE SUB-DIVISION

The single most important structural factor in reducing pollution in the event of damage is the one that's almost never talked about: tank size and arrangement. IMO has developed a method to evaluate a tanker's propensity to spill cargo in the event of hull damage. It is based on a hypothetical collision and a hypothetical grounding. The collision involves a wedge penetrating into the hull about B/5 where B is the ship's beam. The grounding involves damage from the bottom up to about B/15. These parameters were based on a study of past casualties. The collision/grounding is assumed to be equally likely to occur anywhere along the ship's side/bottom. The overall result of this analysis is the ship's Effective Oil Spill (EOS) number, which is the percentage of the ship's cargo which will be spilled on average given the IMO collision/grounding scenario. The IMO system is far from perfect but it is a reasonable starting point for evaluating a design's resistance to spillage.

Table 8.1 shows the IMO Effective Oil Spill numbers for four different pre-MARPOL VLCC's and ULCC's all built in the mid 70's, a typical single hull MARPOL VLCC built in 1986, and a brand new double hull VLCC. From age 25 on, the pre-MARPOL ships must operate under either IMO REG 13G4 (30% of the side or bottom tanks non-cargo) or IMO REG 13G7, usually known as Hydrostatically Balanced Loading (HBL). For these ships, Table 7.1 shows the EOS numbers for each of these regimes.

TABLE 8.1
IMO EFFECTIVE OIL SPILL NUMBERS, ARAB LIGHT, SUMMER MARKS

Design	Percent Cargo Spilled			Number of
	As-built	13G7	13G4	Cargo Tanks
Hellespont Embassy, 1976	2.1	1.8	2.2	34 five across
Empress des Mer, 1976	2.6	2.2	2.4	35 three across
Shell L-Class, 1974	3.6	3.1	3.7	22 three across
Ludwig VLCC, 1974	3.4	2.9	3.3	22 three across
Typical Marpol Single Hull, 1986	4.3			13 three across
SHI 1321, Double Hull VLCC, 2001	2.4	2.1*		17 three across

* Bottom outflow reduced by 50% for oil captured in double sides.

Table 8.1 makes a number of points including:

(1) In terms of expected spillage, 13G7 is clearly superior to 13G4. 13G4 involves keeping a number of tanks totally empty, which lifts the ship out of the water, while the remaining tanks are filled right up to the brim. From a spill minimization point of view, this is exactly the wrong way to go.[6] Spill minimization requires keeping the cargo as low in the

[6] 13G4 also involves higher loss in carrying capacity than 13G7, less flexibility in multi-parcel loads, very high stresses, and most importantly putting ballast in tanks that were not designed to handle sea water. Ballast in unprotected tanks as a result of conversion to segregated ballast was at a minimum a strong contributory cause to the Erika sinking and spill. If enough ships go 13G4, we will see more Erikas.

water as possible in order to reduce the outflow head. As Table 8.1 shows, often a ship will have a higher EOS number under 13G4 than if IMO Regulation 13G never existed.

(2) The MARPOL tankers have terrible EOS numbers. Like 13G4 ships, MARPOL tankers operate with about a third of their tanks completely empty and the rest filled to the brim. The difference is that the designers reacted to the MARPOL requirements by making the ships very tall and the tanks very big, both of which exacerbate spillage further. One can argue that this was an acceptable price to pay to obtain segregated ballast. We find such an argument unconvincing. The nearly new Exxon Valdez would have spilled a lot less oil if she had been an older ship.

But for present purposes, the interesting feature of Table 8.1 is the wide range in expected spillage under 13G7 for the pre-Marpol ships. *The Hellespont Embassy spills 65% less than the L-class in the same casualty scenario.* The other two pre-Marpol ships are in between with the Empress des Mer much closer to the Embassy and the Ludwig V's much closer to the L-class.

This is a product of small tank size. The Embassy and the Empress have a lot more tanks than the other two designs. The Empress, which was built to the last pre-Marpol restriction on tank size, has 13 pairs of wing tanks; the Shell L-class has 8. And in the case of the five across Embassy, the tanks are much more intelligently arranged.

One of the several tragedies of double hulls is that they finesse the IMO tank size limits. The IMO tank size limits apply only to cargo tanks that touch the side shell plate. Double hull wing tanks are thus exempted and the tanks in modern double hulls are enormous. The standard newbuild VLCC with a cargo cubic of 350,000 m3 has only 17 cargo tanks including two small slop tanks. Even the L-class, which we've just finished castigating, with a cargo cubic of 385,000 m3 has 22 cargo tanks.[7] *The EOS number for a new double hull VLCC fully loaded is about 2.4%.* For all the problems with the double hull we end up with a ship that spills more oil in the IMO damage scenario than a good pre-Marpol tanker.

Table 8.2 shows a little more detail of the Embassy (under 13G7) and new double hull comparison.

TABLE 8.2

	SHI 1321	Hellespont Embassy
CARGO	293,755	397,761
SIDE LOSS	15,330	12,195
BOTT LOSS	3,384	6,783

The double hull cuts bottom loss because the probability of the bottom damage extending into a tank is reduced but the new ship gets clobbered on side damage due to the massive wing tanks. The reduction in probability of damage penetrating into the tank is overwhelmed by the increase in outflow if the tank is penetrated.

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[7] The Marpol VLCC's were even worse. Most of these ships were built with 13 cargo tanks. If one were determined to increase spillage in a casualty via structural regulation, it's hard to imagine a more effective set of requirements than the Marpol single hull rules.

This is a little unfair to the double hull in that a portion of the bottom loss will be captured in the top of the double sides. IMO arbitrarily says cut the bottom loss number for double hull by 50%. Adopting this rule, the double hull's EOS number becomes 2.1%, the asterisked number in Table 8.1. But even if we do that, the double hull still has a 17% higher Effective Oil Spill number than the Embassy under 13G7. Paradoxically, double sides are more effective in containing grounding damage than they are in reducing side damage. The way to go after side damage is sub-division. And the double bottom doesn't do anything in grounding that could not be done far more efficiently by HBL.

Sadly Class is implicated in the growth of double hull tank size. The biggest tanks in a modern double hull VLCC are about 50 meters long. Tanks this long have a natural sloshing period which is close to the ship's natural pitch period, 13 to 15 seconds. When a tank's sloshing period matches the ship's pitch period, immense waves can build up in the tank, crashing from end to end. This is known as sloshing resonance.

In the mid-70's no one would design a tank to operate anywhere near sloshing resonance. Thus, even if an owner cared nothing about pollution -- and few did -- you either built smaller tanks or you had to use real (complete) swash bulkheads. The difference in cost between a complete swash bulkhead and a oil-tight bulkhead is not all that great, so the incentive to go with massive tanks was not strong. But over the years, Class has allowed the swash bulkhead to atrophy and then disappear. The argument is that we can operate these tanks at resonance because we can predict the forces and beef up the structure to handle them.

This argument is a sad joke on a number of levels. Nobody has any way of accurately estimating sloshing forces at resonance, certainly not Class. The best of the current lot of Class tools is probably LRFLUIDS. When Daewoo applied LRFLUIDS to the case of our new ULCC's center tank, the program indicated that at resonance the tie beams would impose an important dampening on the sloshing. ABS's empirical gouge said the same thing. The state of the art in sloshing analysis is the Hamburg Ship Research Institute's program which implements a full two phase Navier-Stokes but only 2-D. Despite being 2-D, this is an extremely computationally intensive program. One run simulating a little over one minute in real time took a cluster of 8 Dual-Pentium PC's over two days to compute. The results showed that at resonance the tie beams will have almost no effect on sloshing. The basic wave form is a kind of U that sneaks under the tie-beams as it moves from one end of the tank to the other, not a sort of semi-harmonic wave as Class claims. (Any housewife who has had water slosh out of a basin could have told us the same thing.) The Hamburg results are far closer to reality but the people at Hamburg will be the first to tell you that they cannot accurately predict the loads imposed on the structure. But when this wave crashes into a bulkhead, it climbs over 15 m into the air.

The only reasonable thing to do is to stay away from resonance. And that means real swash bulkheads, not overgrown webs.[8] And once you have real swash bulkheads, the move to oil-tight

[8] In the five years, before the swash bulkheads almost totally disappeared, the practice was to build swash bulkheads with progressively more massive openings in the upper half of the bulkhead on the grounds that the tank was only at resonance when it was less than half full. This gigantic hole saved at most two hundred tons of steel on a VLCC in return for a complicated, ugly structure with all kinds of opportunities for stress concentrations. Inevitably, these ships are now experiencing cracking in the corners of these senseless openings.

and far better sub-division is obvious. The result is a big reduction in spillage and far more flexibility in cargo parcels, ballast exchange, and tank inspection.

The tragedy is that this big improvement would cost very little. Shorter tanks mean:

(1) sloshing forces are markedly reduced which saves steel,

(2) much more importantly, a far more even distribution of transverse forces to the side shell and far better ability to withstand asymmetric loading and racking forces.

An egg carton is a very efficient structure. Cut out every other transverse corrugation and then see what you have to do to get the strength back. Because of the increased efficiency of the structure, limiting tank size doesn't require that much extra steel. The Empress' lightweight, 60,656 tons, is nearly the same as other U's of the same size with larger tanks built a year or two earlier. The Embassy's lightweight, 57,628 tons, is actually a bit smaller than other good pre-Marpol tankers of her size but it is easily the best tanker structure we have ever seen. After 25 years we have not found one crack in the Embassy. She is an instructive exception to the even-good-ships-have-a-few-cracks rule enunciated in Section 4. In other words, small tank size is not expensive. Our kind of regulation: substantial effect on spillage at nil economic cost.

In the case of the massive double hull tanks, we have just the opposite effect: immense forces in the critical lower hopper area near the transverse bulkheads, in way of the centerline buttress, and at the stringer corners and web toes, despite the Rules' staying away from any really difficult loading condition. The bottom bracket of the centerline buttress on our new ULCC has a 57 mm web and a 60 mm faceplate, both HTS. The center tank lower web toes have 50 mm faceplates. That's way too much stress in one place.

At a minimum, the IMO Regulation 24 tank size limits should be applied to all double hull wing tanks and the IMO Regulation 13G(7) prohibition against operating tanks near sloshing resonance should be applied to all tankers.

9. GET RID OF THE DOWN RATCHET

Everyone at ABS and LR we have talked to off the record about the swash bulkhead issue thinks it's absolutely nuts to build 50 m long tanks without a bulkhead. So how did we end up in such an imprudent situation? The answer is what we call "the direct computation down ratchet". In most parts of the Rules, there's a general out clause which says something like "other arrangements are possible if it can be shown by direct computation that they meet or exceed the above standard." Sounds innocuous enough but, when combined with competitive pressures in a litigious world, it results in "the direct computation down ratchet".

Here's the way the down ratchet works. A yard pressures a class to accept some cost saving relaxation of the rules by offering a "direct analysis" of the problem. Each yard has dozens of very sharp young naval architects who do nothing more than work on beating the rule. Once a contract is signed every kilogram of steel they save by so doing goes directly to the yard's bottom line. Inevitably some of these efforts are approved, however imprudently.[9] As soon as that happens, the new lower requirement becomes the standard. The class involved can't admit it was imprudent to approve the change. If it did, it would have legal problems on all the ships that had been approved with the change; not to mention some very angry owners asking why did you approve this mess on my ships and then not on his; and not to mention an extremely angry yard who bid the ship under the "new" rule and finds out it has to build the ship under the "old" rule.[10] The other classes have to fall in line because if they don't, their ships will be more expensive and they will lose owners. Having established a new lower standard, the yards then compete away the saving and must find new ways to save costs at that lower level, and the process repeats itself. Over time the ships get cheaper and lousier.

Figure 9.1 shows that in real terms, a VLCC today costs less than half as much as a VLCC built in the mid 70's despite the 12 to 15% increase in cost associated with double hulls. Yet there have been only incremental advances in ship building technology in the last 25 years. The big tanker yards look much the same now as they did in the mid-late 70's. By far the most important reason the ships are so cheap is the down ratchet.

There's only one way to eliminate the down ratchet: get rid of all the "other arrangements" clauses. If a yard wants to change the rules, make it go through the normal rule change process. As it is, the yards are slowly but surely re-writing the rules. Did any Class Technical Committee get together and say let's get rid of the swash bulkheads in 50 m long tanks? They did not. There was never any such meeting. And if anybody had come into a Technical Committee meeting with such a suggestion, he would have been thrown out of the room. Yet the swash bulkheads are gone.

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[9] Since the yard pays the newbuilding classification fees, in a commercial sense the yard is Class's customer. Occasionally, in this situation, the first rule of retailing applies.

[10] We know of at least one instance where a yard has threatened legal action against class for "changing the rules" in this manner.

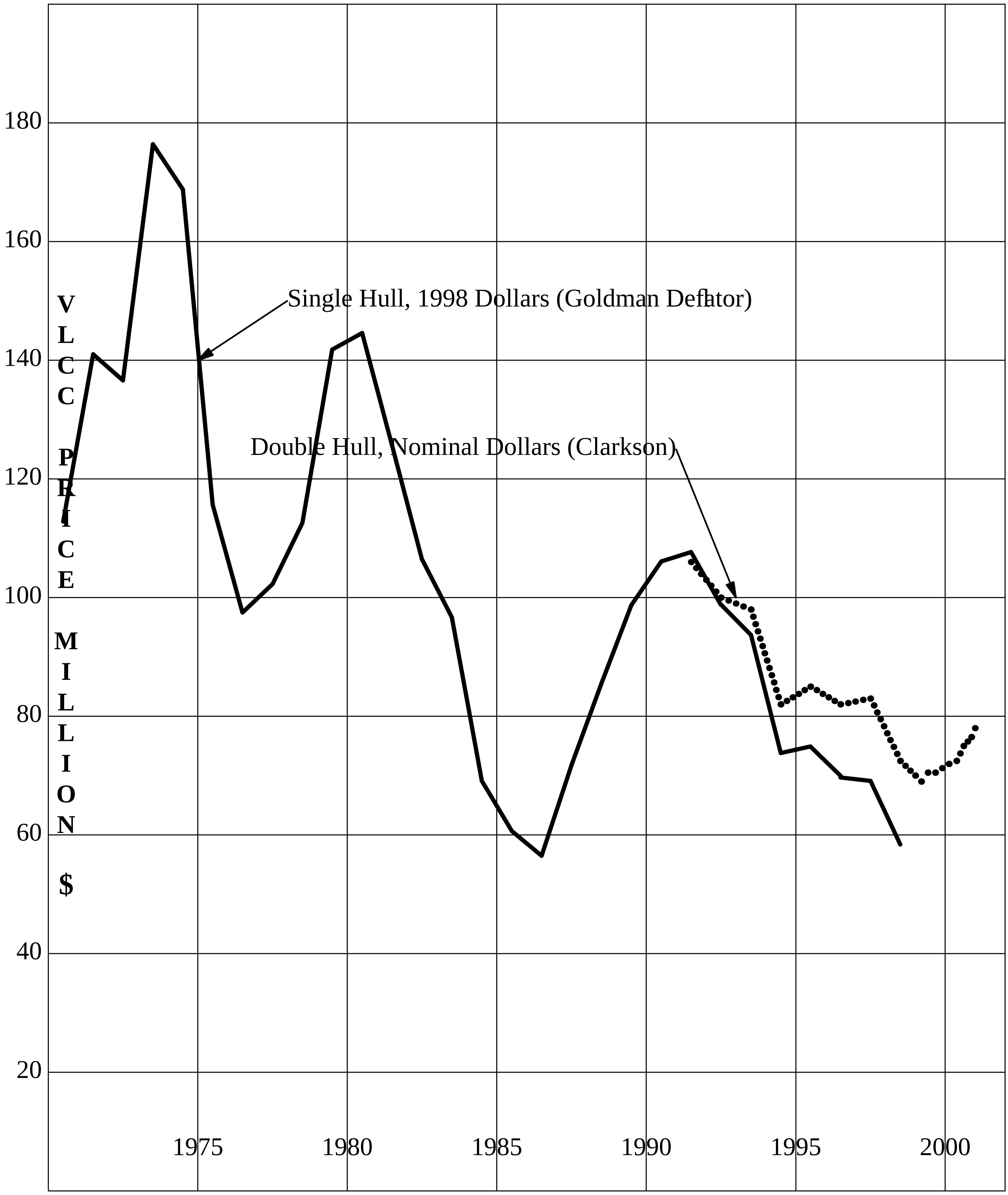

FIGURE 9.1 REAL PRICE OF A VLCC, 1970-2000

10. GET RID OF THE CALIBRATION FACTORS

Buried in the rules there are a number of numbers that don't make any physical sense. We've already seen one case in the 0.9 factor that ABS inserted in the sloshing force calculations. The idea here is that our method is too conservative so we need to "calibrate" it to actual experience. In practice, this means that some other class has used a less conservative method, and so far nothing terrible seems to have happened. This is a horrible design philosophy, and an open invitation to commercial pressures, precisely what happened in the design cargo density situation. The change wasn't based on anything other than the other classes were lighter.

Here's a more egregious case. Class claims that it designs the ship on a 20 year basis, that is to the situation which has a probability of 0.5 of being encountered in twenty years. This in itself is a revealingly strange philosophy. It seems to imply that a ship should last only twenty years. But even if you accept this invitation to shoddy construction, do you want to design so that there is 50% chance that the ship won't make it to age 20? Suppose you knew that every airplane was going to be scrapped at age 20. Would you accept a 50% chance that it would break in two in flight before then? A 1% chance maybe, 5% if you really want to be imprudent, but not 50%.

In fact, the ships are not designed to even this dubious criterion. For example, when you go in the rules, both ABS and LR, they say "compute the design sloshing forces on the basis of a pitch that is 0.6 of the 20 year pitch". We've asked a number of ABS and LR people where this 0.6 came from and got either blank stares or a semi-circular argument to the effect that this is the way the rules have been and we haven't seen any real problems yet. Forgetting about the fact that we only very recently got rid of the swash bulkheads on big tankers and most cargo tanks are not slack most of the time, so we have no real experience, the point is that the ship is not being designed on the basis of 20 year encounter, but something much less. The rules should make that clear.

Get rid of all calibration factors unless they are strongly supported by experimental evidence from carefully designed and publicly documented tests. And then make sure that these remaining calibration factors are very publicly documented, so owners who would prefer not to use them are alerted to their existence.

11. FOREBODY AND AFTBODY

Calibration factors are rampant in the forebody rules. In part this is because the finite element models haven't as a matter of course extended to this part of the hull. In part, it is because the forces, especially in the forebody are particularly difficult to predict. This means we should be particularly conservative.

Unfortunately, this is not the case. Class goes to some trouble to estimate peak slamming pressures and then immediately applies a bunch of calibration factors of unknown or dubious origin. In many cases, the resulting scantlings are lower than the less highly loaded structure in the midbody. The result is that forebody damage is still the most common form of structural failure.

Another big problem is structural continuity. Hellespont was amazed to find out that the yards felt no compulsion to extend the main horizontal members: the innerbottom and the double side stringers into the forebody or aft body in any reasonably continuous manner. Most were abruptly ended with minimal scarfing brackets and new flats and decks placed at whatever level suited the yard in the ends of the ships. At the aft end, longitudinal bulkheads deteriorate abruptly into a series of widely spaced pillars with no shear strength whatsoever. The yards argue that this is required by machinery arrangement issues but both the Hellespont V and U have real longitudinal bulkheads extending to the aft peak tank and the engine room works fine. Even the upper deck is abruptly ended, usually just aft of the accommodations with a big increase in vertical deflection.

Transversely, the engine room is inherently weak due both to the narrow sterns now being used and the large hole in the structure forced by the main engine. There is essentially no transverse structure between the forward and aft engine room bulkheads in the center third of the ship. Once again the yards claim that this is required by machinery arrangement. In fact, the web just aft of the main engine and just forward of the boilers and generators could be effectively filled in without affecting the engine room operation.

Class seems to have no rules that prevent this. They even allowed upper deck stiffeners to be cut at the engine room bulkhead in direct violation of the rules for primary longitudinal members. The argument seems to be these are not high stress areas so we don't have to worry about it. But the fact is that the stresses are not even analyzed. And the best way to turn an area that should not be a problem into a problem is with structure that violates basic principles of good design. The rules with respect to continuity between the cargo tank area and the ends of the ship must be massively strengthened. The side shell stringers and the inner bottom should extend from one end of the ship to the other in a continuous fashion. Ditto for the longitudinal bulkheads. And we need a real transverse bulkhead between the Main Engine and the boilers.

Another big problem in the aft body is hull deflection and its impact on shaft and engine bearing loads. Amazingly, Class doesn't even require this deflection to be calculated, despite the fact that there have been a rash of stern tube bearing failures and the repair yards are reporting very rapid weardown of VLCC shaft bearings. Needless to say shaft bearing reliability is absolutely crucial to these single screw ships. The rules should require detailed analysis of hull deflection -- both vertical and transverse -- in way of the main engine and shaft, and the results of those calculations fed to the shaft alignment analysis (see Section 13 below). The hull deflection numbers will be an automatic output of the full hull FEM discussed in Section 4 above, provided we pose a proper range of vertical and transverse loads. The transverse loads should be based on high speed turns, port and starboard.

12. WELDING

One of the authors inspected the Exxon Valdez in the dock in San Diego after the grounding. The real eye opener was not the damage, as extensive as it was, but the welding. Just about everywhere in the damaged areas the stiffener webs had pulled cleanly out of the welds leaving the welds on the plating. The welds were tiny. They looked like continuous tack welds. When the Exxon superintendent was asked if the welds were legal, he angrily nodded yes. This sort of failure absorbs almost no energy and may be a reason the Valdez rode so far up on Bligh Reef.[11]

In the bad old days, the standard filet welds in a large tanker cargo tank had a throat thickness of 6.4 mm (9 mm leg length). Now much of the filet welding in the tank area has a required throat thickness of 3.2 to 4.0 mm. And Class allows the yards a 10% negative margin. So often the actual weld is less than 3 mm throat. To expect any penetration at all with such welds is crazy talk. Those are the welds we saw in San Diego. Miniscule. And stupid. The yards can easily lay down 5 mm plus throat welds with a single pass. When Hellespont asked the yard to increase all welds in the ULCC cargo tank area to 5.3 mm throat -- after signing the contract -- the price was $44,000; and it cost the yard far less.

Whatever argument -- probably nothing more than the down ratchet -- led to the current weld sizes, it should be abandoned. Welds corrode much more rapidly than the other steel. And as the Valdez showed, these tiny welds fall apart on impact.

A particularly important weld is the lower hopper corner. First problem is that the weld is difficult to do because of the restricted space. The only reasonably sure way of laying down the root pass is manually (semi-automatic in yard parlance). This should be a requirement. Otherwise it is nearly certain that the root pass will not fill in the bottom of the notch. However, the biggest problem is that a great deal of stress is trying to turn a corner at this weld. We have no idea why we don't roll this corner as if it were the bilge radius. But if we are going to have a sharp corner and a natural stress concentrator, we must radius the weld very carefully.

Given a series of failures in hopper welds in double hull tankers operating in the North Sea, LR is finally starting to address this issue. LR performed a super detailed FEM of the hopper corner with a mesh size of T/12 where T is the thickness of the inner bottom. Unsurprisingly, the minimum radius of the weld is critical to the maximum stress in the corner. With a "Class standard" weld, the max stress was 660 N/mm2, *almost three times yield*. With a 15 mm radius weld, the max stress was 265 N/mm2. With a 30 mm minimum radius, the max stress was 210 N/mm2. It seems to us obvious that we should adopt a minimum weld radius of at least 30 mm for at least

[11] A complete digression. When the Valdez was brought to San Diego, Exxon was extremely worried about oil pollution in the yard as the tanks drained. The tanks were coated with crude after the grounding and with no sunlight and low water temperatures biological activity inside the tanks would be extremely slow. When the ship finally got off San Diego, seven months after the spill, Exxon sent a diver into the tanks to find out how much cleaning needed to be done. The oil had become a mass of algae and a population of zooplankton up to small shrimp was thriving. No tank cleaning was required.

300 mm on either side of each web. This is essentially a more careful version of the new LR weld profile 6.

Another area where the rules have fallen apart is heat treatment. The most egregious case we know of is the yard's refusing the LR surveyor's request to stress relieve the critical welds between the upper rudder casting and pintel casting to the 82 mm thick plate that connects them on our ULCC.[12] The yard was just following standard practice these days.[13] In defiance of good engineering practice *for non-critical areas*, stress relief is not explicitly required under either ABS or LR rules. In this case, the LR surveyor strenuously and courageously objected. But LR failed to support its own surveyor when the yard pointed out that stress relief was not explicitly in the rules. Not only are the days when the surveyor's word was law long since gone; the yard put Lloyds on notice that the surveyor's behavior was unacceptable and asked for his dismissal. In the yards' view, the surveyor has no right to require anything that is not explicitly in the rule. Sadly that's essentially the societies' position as well.

Hellespont had to pay extra for the heat treatment. The cost was trivial, $2,000. The yard was just making the point that a surveyor had no right to exercise his judgment. Obviously, if a yard doesn't have to heat treat this critical weld, there are almost no welds that it does. In the 1970's, Class required stress relief on all welds in excess of 40 mm thickness. The down ratchet has been working big time in this area. All welds involving castings and very thick plates should follow a carefully prescribed, conservative heat treatment.

[12] This request was refused despite a haphazard welding sequence which thermally cycled the steel several times.

[13] We know of one case where a new VLCC was found to have a badly cracked, pintle casting on its first docking.

13. SHAFTING SYSTEM DESIGN

If we had to pick the single most critical problem resulting from the down ratchet, it would be main engine shafting. There have been at least eight stern tube bearing failures *on brand new VLCC's* in the last three years. A stern tube bearing failure generally leaves a ship adrift and helpless. The only good thing about these particular failures is that they occurred so rapidly -- in two cases before the ship was delivered -- that the ships had not yet loaded oil. These failures are shown in Table 13.1.

TABLE 13.1
KNOWN STERNTUBE BEARING FAILURES ON BRAND-NEW VLCC"s

YARD	HULL NO	DATE
DHI	5109	1998-08-17
HHI	1089	1998-05-08
HHI	1090	1998-07-03
DHI	5120	1999-01-05
DHI	5121	1999-01-28
HHI	1164	1999-10-26
SHI	1241	1999-12-02
SHI	1241	1999-12-02

It is a rare month in which we do not hear reports or rumors of yet another shafting problem on young VLCC's. Thanks to the ability of this industry to hide its problems -- a practice in which the Classifi cation societies play an important role -- we can be sure that there have been many problems we haven't heard about. The repair yards tell us that the new VLCC's are showing up at their fi rst docking with very rapid bearing weardown. In July, 2001, HHI 1090 was again out of service with major stern tube bearing problems. Something is badly wrong.

Since all the Table 13.1 failures involved composite rather than white metal bearings, the immediate reaction was that there was something wrong with the composite material. Several owners replaced a newer composite material with a composite material, that had proven itself for over 25 years in hundreds of large tankers, and which in the opinion of most of the tanker industry including the authors was superior to white metal. Two of these proven bearings failed almost immediately. Are we to believe a material that almost always lasts 15 or more years in heavy duty at sea service is the cause of dual failures within a few hours of installation?

Attention then turned to alignment. This was a natural assumption. The Class approved alignment procedure used by the yards is very crude. Over the years, the yards had somehow received class permission to bore the stern tube at the block stage, then weld the stern tube block in place, align by piano wire with the ship still on blocks, and hope. In fact, until the new LR limit on misalignment comes into force, there are no truly concrete requirements with respect to alignment. It was suspected correctly that the yards were taking advantage of the rules and the self-aligning characteristics of the composite bearings to be very sloppy in alignment. However in late 1999, LR carefully aligned two shafts using modern strain gauging techniques with the ship afloat at maximum alongside draft. Both these bearings failed before the ship completed trials.

Alignment may be lousy but it is clearly not the root cause. At this point, Class seems to have no real idea of what is the root cause.

The current "solution" is to use white metal bearings and high volume, forced lubrication in place of the traditional oil bath system. This is a dangerous work-around, not a solution. The repair yards are reporting very rapid weardown in the white metal bearings, and in our opinion, it's only a matter of time before they begin failing. High pressure lubrication is an invitation to blown stern-tube seals, and more importantly forces the the crews to make an impossible choice. If a stern tube seal starts leaking on a ship -- and this happens all the time -- the crew's normal response is to reduce the pressure in the stern tube lube oil system to nearly the same as the external sea water pressure. In an old style oil bath lubricating system, this generally halts the leak with nil increase in the chance of a bearing failure. If the crews attempt this with the current forced lubrication system, they face a high risk of a disabling casualty. If they don't adjust the pressure down, they face the certainty of a large fine at the next port, and a very displeased employer. This is the down ratchet in action.

Before we discuss what we believe to be the cause of these failures, it is interesting to review the state of the art in shaft alignment system. In the early '70s the Japanese yards were modeling the aft stern tube bearing with a reaction point at each end of the bearing, and the forward stern tube bearing and intermediate bearing each with a middle single reaction. They would solve the problem with slide rules and calculators, plot the displacements and determine what if any stern tube bearing slope bore was required to minimize the slope mismatch between the shaft slope and the bearing slope. This would be done for a couple of static conditions and sometimes for quasi-static ballast and laden conditions involving a guestimate of propeller thrust eccentricity and hull deflection.

What do we do now in an era where PCs have more than a million fold increase in computational power over the mainframes of the early 70s? The two reaction point model of the stern tube bearing has been replaced by a single reaction centered at a point that arbitrarily ranges from L/4 to L/2 from the aft end of the bearing. The location of this point is not dependent on any variable that has anything to do with the problem, but on "yard practice". This is done for a single static condition. There is no modeling of aft body hull deflection and no modeling of load/pressure distribution within the stern tube bearing, and no modeling of the oil film at all. All this is per Class rule.

Given such a situation, it is hardly surprising that the real problem does not emerge from the Class required shaft alignment calculations. As Table 13.2 shows, over the last twenty five years, shaft diameters have decreased by at least fifteen percent for the same torque. This is a product of both higher strength material and the down ratchet. Since shaft bending goes as diameter to the fourth power, the net effect is that bending within the bearing has increased by more than 75% for the same propeller weight. At the same time, propellers have become bigger and heavier due to the decrease in Main Engine RPM. There are no class restrictions on bending within the bearing. However, there is one class that has the capability to study at least a part of this problem. Bureau Veritas has a program which has the ability to model bending within the bearing and determine the resulting pressure distribution and oil film thicknesses within the bearing.

TABLE 13.2
DETERIORATION IN SHAFT DIAMETER 1975-1999

SHIP	SHP	RPM	DIAM (mm)	
Hellespont Embassy, 1976	45,000	85	1,010	Smooth turbine torque
VLCC 1999	44,640	76	820	80% torque pulses

Figure 13.1 shows a typical result. Figure 13.1 is based on an alignment that was acceptable to class on the grounds that the misalignment at the aft end of the bearing was small (1.0e-4 radians) and the nominal pressure (bearing load versus overall bearing area) was a reasonably conservative 6.3 bar, well below the Class limit of 9.0 bar. But what counts is the distribution of the pressure within the bearing and the standard Class approved method simply cannot address that issue. In this case,the BV results show the pressure on the aft 10% of the bearing averages 140 bar, well over BV's recommended (and none too conservative) max of 100 bar. At this pressure the film thickness is a miniscule 31 microns.

And here finally we come to the reason why the composite bearings have failed immediately while the white metal bearings have taken longer. There is only one area where white metal is better than composite but in that area it is far better. That area is heat conductivity. The conductivity of white metal is over 30 times higher than that of the composite bearings. The composite bearing relies on the lubricating fluid to conduct away the 7.5 KW of heat generated in a VLCC stern tube bearing. But most of that heat is generated in the high pressure portion of the bearing where the film thickness is much too thin to do the job. The composite bearing burns out almost immediately. White metal has great deal more ability to conduct the heat away itself so there is no immediate burn out despite the thin film thickness. But that doesn't change the fact that the pressures are very high, in fact far above the yield point of the white metal, rapid weardown will occur, and premature failure is inevitable. We will see a lot of VLCC's dead in the water due to bearing failures. The only real question is: how many will drift ashore?

Hellespont decided to go with a 15% thicker shaft than Class requires. This brings the shaft diameter almost back to the standards of the 1970's and reduces bending in the bearing by over 70%. With this system we were able to get the max pressure according to BV down to 50 bar from 165 bar. It also allowed us to go to a two bearing system, to obtain the flexibility we needed with respect to bending moment and shear force at the Main Engine coupling. It is clear to us that the reason why the mid-70's V's and U's have had relatively good shaft performance despite the crude analytical procedures was the shaft diameters that were used. The industry has a choice:

(A) Go back to those diameters at a cost of less than $100,000 per ship.

(B) Implement a much stricter set of rules based on computing and limiting the distribution of pressure within the bearings.

Our view is that (B) will almost certainly lead to (A); in effect that we should do both.

However, the BV-style analysis is still lacking in at least three areas:

(1) We must do a much better job of estimating hull deflection in way of the shaft and main engine for a range of loading conditions, and the results should be fed into the shaft alignment calculation. For a typical modern VLCC, the difference in the shaft-line deflection in way of the intermediate bearing between loaded and ballast conditions is of the order of 5 mm. But under current rules, the actual deflection for a particular hull is not known. Unless we properly account for this deflection, it's nonsense to worry about aligning a shaft to +/- 0.1mm. Furthermore it may be possible to reduce this deflection materially by beefing up the engine room structure. We believe that proper shaft alignment design will require a stronger engine room structure. This is discussed in Section 3 which talks about the various reasons we need a full hull FE model. Proper shaft and bearing design is one of these reasons.

(2) We must do a much better job of incorporating the time varying nature of the loads on the stern tube bearing. Figure 13.2 shows the vertical moment on the shaft produced by the propeller as it revolves for our ULCC. This is a four bladed propeller so the pattern repeats every 90 degrees. Figure 13.2 takes advantage of this by showing only one-quarter of a revolution. Figure 13.2 shows that at full load on average the propeller imposes a moment on the bearing which lifts the shaft at the aft end of the bearing. This is due to the center of thrust being on average above the shaft line. This uplift is often offered as a reason why we don't have to worry about high pressures in the aft end of the bearing, although never in our experience with any real back-up.[14]

Figure 13.2 shows that for this ship at deep draft the average uplift moment is about 300 kN-m. But much more importantly, Figure 13.2 shows that this moment varies from an uplift on aft end of bearing of 700 kN-m at 70 degrees to a downward moment of 100 kN-m (0 to 35 degrees). In the ballast case, the moment is always pressing down on the aft end of the bearing. In short, the uplift is not always taking pressure off the aft end of the bearing.

The propeller also generates vertical and transverse forces which are almost always ignored. Figure 13.3 shows that these vertical forces are strongly downward and can be as high as 12 tons which corresponds to a downward moment on the aft end of the bearing of about 250 kN-m. These SSPA calculations are based on the measured wake without the propeller. The propeller will tend to smooth out the wake and should reduce these forces and moments.[15] But the point is that the time varying nature of the propeller forces imposes additional demands on the bearing which are currently ignored by the Rules as are transverse moments and forces which are roughly of the same size, even when the ship is going in a straight line. We have the capability of estimating these forces, and the results should be fed back into the shaft and bearing analysis. The simplistic assumption

[14] LR goes so far as recommending 1.0 to 2.0e-4 radians designed misalignment in the static case on the grounds that the uplift will then align the shaft better when operating.

[15] On the other hand, thanks to the nine cylinder engine, this ship has a finer aftbody than most tankers and a more even wake distribution.

of a steady uplift which will always reduce pressure in the aft end of the bearing is almost certainly wrong.

(3) We must incorporate the heat dissipation issue into the analysis. Once we have the film thickness and bearing pressure along the shaft for a particular operating condition, it is a fairly straightforward problem to use Reynolds equation to estimate the heat generated within the film and the resulting oil temperature rise. This in turn will change the viscosity which can then be fed back into the preceding analysis as necessary. In fact, Lloyds has a program (Program 1811 Stern Bearing Misalignment Conditions) which performs these calculations. But this program is rarely used in anger and never, to our knowledge, as part of a systematic shaft design process.

(4) Finally, we must do all this for a range of operating conditions: laden, ballast, aft peak tank full, aft peak tank empty, engine hot, engine cold, RPM at full speed, and RPM during maneuvering and lightering, and make sure that the shafting system can handle all these conditions. This implies running the analysis over and over, but that's what the computer is good at. A particularly difficult case is lightering in warm sea water. The fact that a VLCC has to operate for extended periods of time at 10 to 15 RPM during lightering, an extremely tough situation for shaft lubrication, is completely ignored by the Rules. But the best alignment will be a compromise. An alignment that looks very good in one operating condition can easily be horrible in other conditions. Unless you examine a full range of conditions, you will never know.

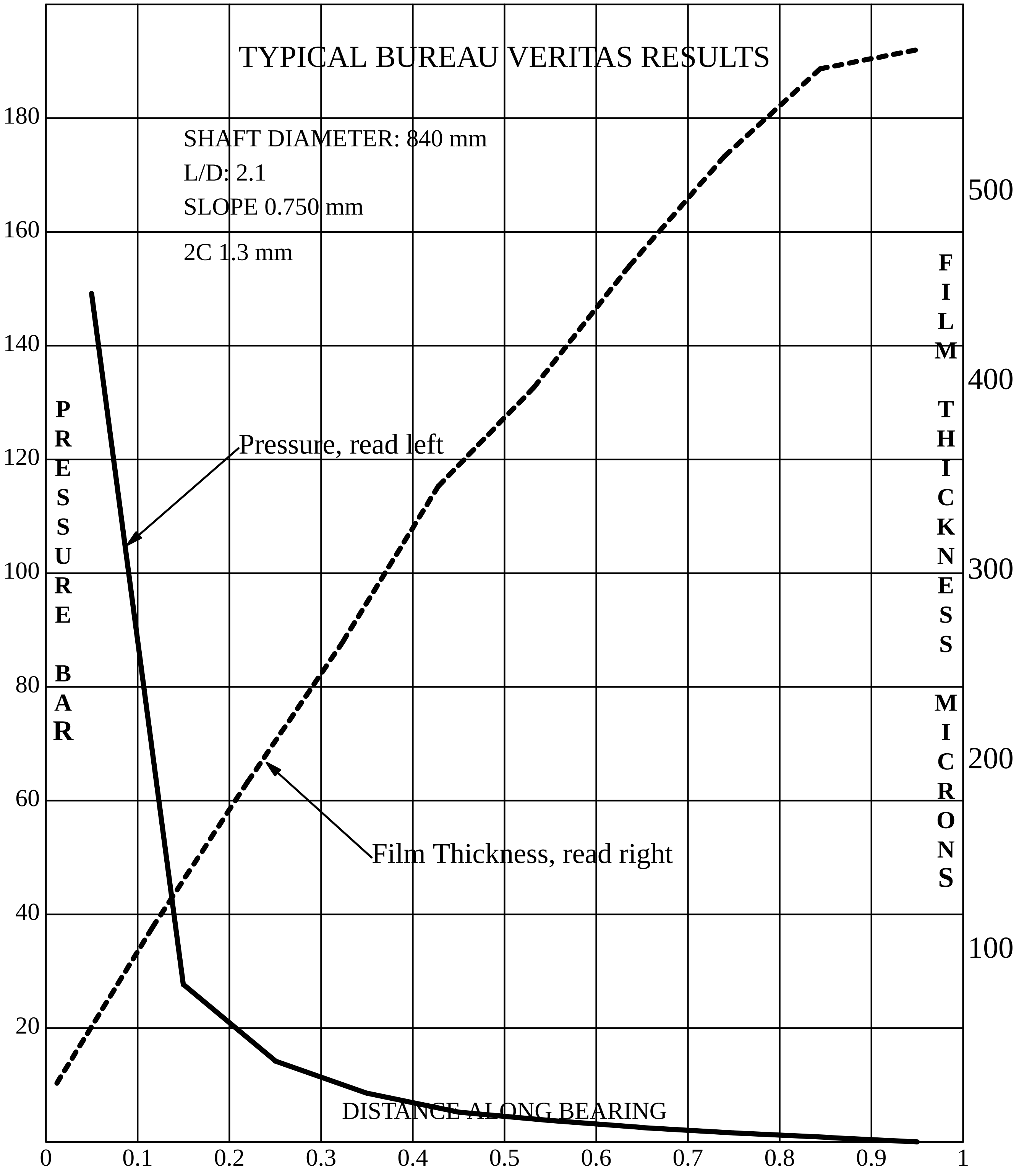

FIGURE 13.1 AFT STERN TUBE BEARING PRESSURE AND FILM THICKNESS

FIGURE 13.2 PROPELLER VERTICAL MOMENT

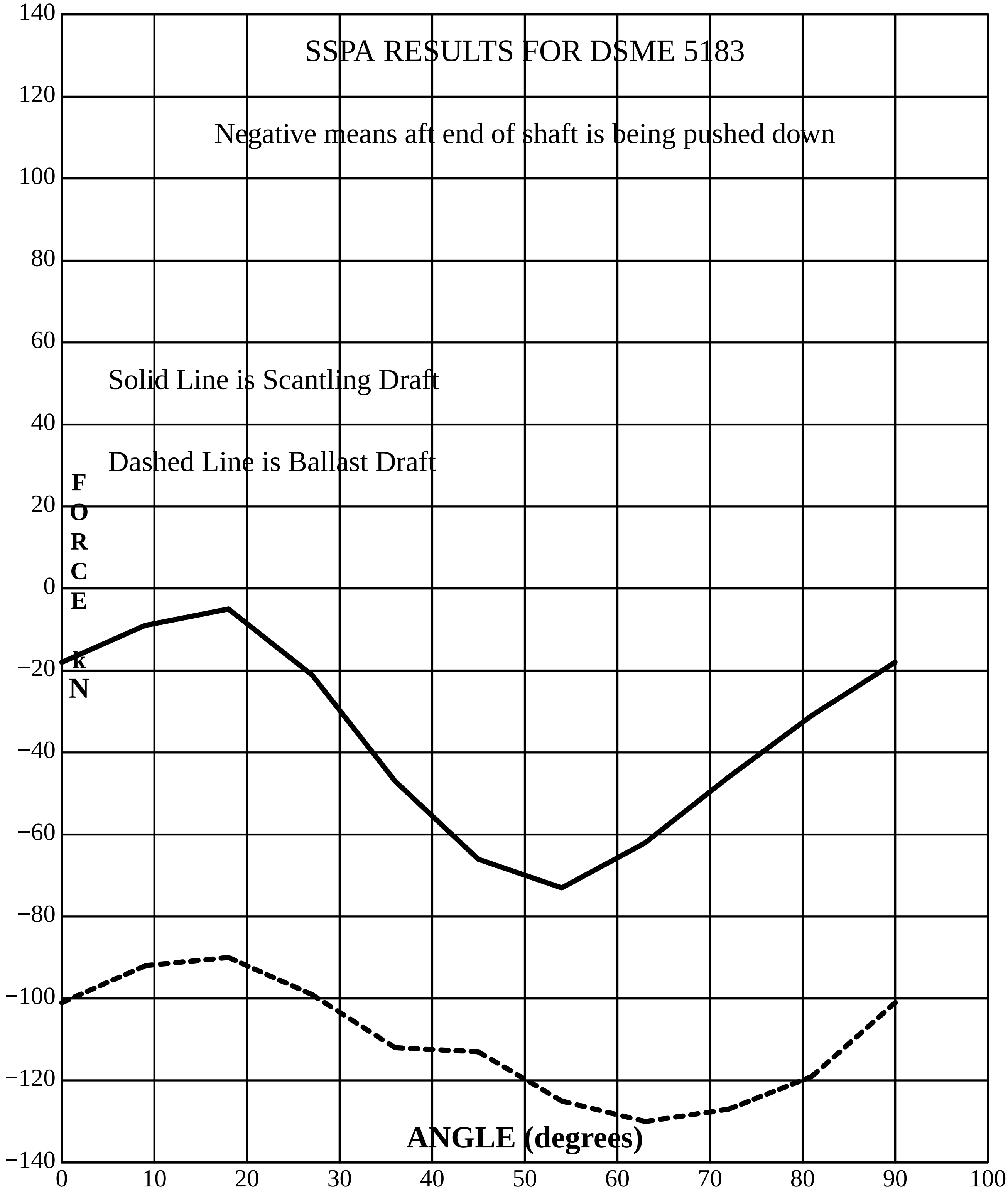

FIGURE 13.3 PROPELLER VERTICAL FORCE

14. SHAFTING SYSTEM ALIGNMENT

Once you have a robust shafting/bearing system design, you must align it properly. The level of accuracy required is +/- 0.1 mm outside the stern tube bearing and +/- 0.05 mm within the stern tube bearing. This is not being achieved.

In spite of the fact that laser alignment and optical procedures such as Taylor-Hobson have been around for more than a decade, the yards, with Class approval, still depend on piano wire for alignment of the stern tube. The shaft alignment proper is done by hydraulic jack-up tests which involve measuring the load on a jack's dial near the intermediate bearing and engine coupling flange. This procedure has at least four glaring limitations:

(1) Jack up tests can only measure what's happening at the intermediate bearing and the engine coupling. But what we are really interested in is what's happening in the stern tube. How do we calculate what's happening in the stern tube from the jack up tests? We use the yard's shaft alignment model -- the very model we just finished castigating in the previous section. This is a circular process, and what's being circulated is error. In particular, jack up tests tell us nearly nothing about the all important pressure distribution within the aft bearing.

(2) Jack up tests are completely static. No jack measurement can tell us what happens as the shaft revolves. But all the interesting shaft phenomena including the bearing oil film pressure only come into play with the shaft rotating.

(3) Jack up tests are limited to a couple of conditions. Currently, the Rules require only three conditions: in dock, cold along side at extremely low draft, and one condition with the engine hot. The first two conditions are totally artificial and the last is the RPM = 0 "operating" condition. We have absolutely no information on how the bearing reaction forces change with draft or major local load changes such as aft peak tank full, aft peak tank empty.

(4) Jack up tests are notoriously inaccurate and unrepeatable. Due to measurement errors and hysteresis effects, the allowed error range is +/- 20%. In many cases, the error range is larger than the design margins. The yard could use electronic load cells and reduce the hysteresis effect markedly, but the other limitations would still remain.

These limitations are totally unnecessary. For at least the last twenty years, naval vessels have had their shafts aligned by dynamic strain gauging. The results are far more accurate, can cover almost all the important system variables, and can be taken with the shaft operating for a large range of conditions including turning. *The down ratchet doesn't always operated by weakening the Rules. Sometimes it works by preventing an obvious improvement to the Rules from being implemented.* Any Class that attempts to impose a new procedure which might possibly force an improvement in standards -- no matter how obvious -- will run into exactly the same factors that make the down ratchet function in the first place.[16] It just doesn't happen.

Given the mini-epidemic of shaft bearing failures and the availability of dynamic strain gauging, we recommend that the Rules require that shaft alignment be checked by the following instrumentation:

a) A set of strain gauges attached to the shaft aft of the aft bearing. Properly placed these can directly measure the bending moment and shear in both horizontal and vertical directions that result from both the static and dynamic propeller forces.

b) Two sets of horizontal and vertical displacement sensors: one just aft of the aft bearing and one just forward of the aft bearing. These will determine the shaft centerline as it moves around under the influence of the oil film pressure and the loads from the propeller. The results during turns and at low RPM will be particularly interesting.

c) A set of strain gauges fixed to the shaft just forward of the aft bearing to determine the horizontal and transverse bending moments.

d) A set of gauges just forward of any forward stern tube bearing and a set forward of the intermediate bearing.

e) A final set of gauges at the main engine coupling to measure the shear force and bending moment that the shaft imposes on the crank shaft.

With such a system,

(1) The propeller forces and bending moments are measured directly. There is no need make any assumptions about the propeller forces.

(2) The displacement, shear force and bending moment of the shaft at both ends of the aft bearing are known. This should be enough to make a simple but reasonable model of the pressure distribution within the bearing.

(3) One can directly measure the shear force and bending moment at the main engine coupling through its complete firing order. This is critical to main engine journal bearing failures inter alia.

(4) These measurements can be taken continuously even while maneuvering and over a variety of drafts, loading conditions, and RPM.

(5) Finally one can leave the system in place and monitor the shaft in actual operation.[17]

Once this system become standard and routine, the additional cost relative to the current system will be negligible. Shaft alignment will go smoother and quicker. There will be fewer redos. The quality of the alignment will be drastically improved. And the data that will be generated will materially improve our understanding of this critical issue.

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[16] Ever wonder why the Finite Element models didn't get better as the cost of computation went down by a factor of 100,000?

[17] The cost of the gauges, the wireless transmitters, the signal conditioners is less than $20,000. If the situation starts to deteriorate, there a good chance this system will pick it up before the actual failure.

15. SUMMARY

We could go on and on. The down ratchet works everywhere. Rudder stock dimensions, steering gear torque, propeller blades, awkward and dangerous outfitting including *pump room ventilation ducts that don't go to the pump room bottom*, etc, etc. Table 15.1 shows one last example: main engine shaft couplings.

Table 15.1
Deterioration in Main Shaft Couplings, 1975 to 2001

	1975	2000
Power	45,000 SHP	44,640 BHP
RPM	85	76
Engine Torque Pulse	nil	80%
Flange Thickness	220 mm	140 mm
Coupling Bolts	14 x 150 mm conical	12 x 95 mm reamer

We are not talking about losing 15 to 20% here. We are talking about losing 150 to 200% despite the much harsher design conditions associated with the diesel and its massive torque fluctuations. We know of three shaft coupling failures in new VLCC's, which means we can be sure there have been more. The repair yards tell us there is no chance of unscrewing the shaft couplings when the new V's come in for their first docking. They expect to have to drill them out since they all been over-torqued and badly fretted. And what does the owner gain for taking these massive risks? A few thousand dollars per ship which he will give back to the repair yard on the first docking. This is the down ratchet gone berserk.

Table 15.1 is an extreme case, but it's very difficult to find any scantling or parameter that is not at least 15% weaker on the new ships than the old. As a result, large tankers built to current Class rules are far less safe and less reliable vessels than those built 25 years ago. And the mid-70's ships as a group were not over-built. They were at best just good enough. Yet in the same period the potential liabilities associated with large tanker casualties have increased one hundred fold. It simply makes no sense. If the classification society system is to continue to exist, if it should be allowed to continue to exist, the Rules must be rewritten returning at a minimum to the standards of the 1970's. And the direct computation down ratchet must be eliminated.