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EVOLUTION OF A MODERN NAVAL STEERING SYSTEM

THE AUTHORS

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ABSTRACT

Ships' steering systems have progressed over the ages, driven by the need for more powerful and reliable systems. From simple mechanical devices thousands of years old to the latest electro-hydraulic systems, progress has been relatively slow but constant. Present systems tend to be the result of a series of modifications of older designs in response to a need for increased performance and reliability. This paper decribes the historical development leading to present designs and discusses a new approach which not only eliminates most problems inherent in present designs but also provides a more economical, more compact, and more cost effective solution as verified by detailed reliability and maintainability analysis.

INTRODUCTION

Several thousand years ago steering oars appeared on sailing vessels as a means of controlling the crafts' direction. Pushing the stern-mounted fore-and-aft oar to one side or the other developed a force which would turn the vessel in the required direction or bring it back on course when deflected by wind or sea conditions. The origin of the word starboard is often attributed to the steering oar or "steerboard."

As vessels increased in size, rudders were attached to the stern providing a stronger and much more effective means of inducing turning forces on the vessel. Initially these rudders were controlled by a single lever arm or tiller, but as vessel size and rudder area increased, some form of mechanical advantage was required in order to allow the helmsman to develop the required forces. A system of ropes and pulleys was devised to develop the required mechanical advantage and allow the steering wheel to be mounted at some location forward of the rudder shaft.

Eighteenth century "ships of the line," the largest warships of the time, carried triple or even quadruple steering wheels of over 10-foot diameter, each requiring two men under heavy sailing conditions. The limit of manual steering systems had then been reached, with the next century seeing the introduction of steam power-assisted steering. The steering system was eventually a steam-driven winch with cables from the drum driving a quadrant at the rudder shaft. The helmsman's bridge input command was transmitted by a rotating shaft to a steam differential valve which directed steam into reciprocating cylinders driving the drum. As a proportional amount of rudder was applied, the differential valve would close and stop rudder movement. In this way large rudder torques could be developed with minimal helmsman effort.

BASIS OF PRESENT DESIGNS

In the early twentieth century, battleships had such large rudders and such high hull speeds that the required torques could not be effectively developed and resisted by steam steering gears. At about the same time, the large rotating gun turrets had control problems and the Hele-Shaw hydraulic pump [1] was developed to provide a faster and more accurate means of positioning heavy loads. The power to drive the pump was generally supplied by steam, but the pump was constructed such that by moving its input lever, the output hydraulic flow could be reversed and modulated in rate. The Hele-Shaw pump was of a radial piston type such that the shifting of a cam ring by the control lever controlled the hydraulic power with little input effort. Having proved successful in positioning gun turrets, the Hele-Shaw pump was then applied to steering gears with equal success. Rotational steering inputs from bridge control shafting, as used in the previous steam system, were fed through a rotary differential. This differential mechanism provided an output arm which adjusted the Hele-Shaw pump control lever such that when the bridge-commanded rudder angle equalled the actual rudder angle the pump was off stroke and did not cause rudder movement. Any difference between rudder angle or helm angle would cause the output lever to move, thereby putting the pump on stroke to apply the appropriate flow direction and magnitude so that any error between the helm and rudder angle would be reduced to zero. Compared to previous systems, much faster, more accurate and more powerful rudder movements could be

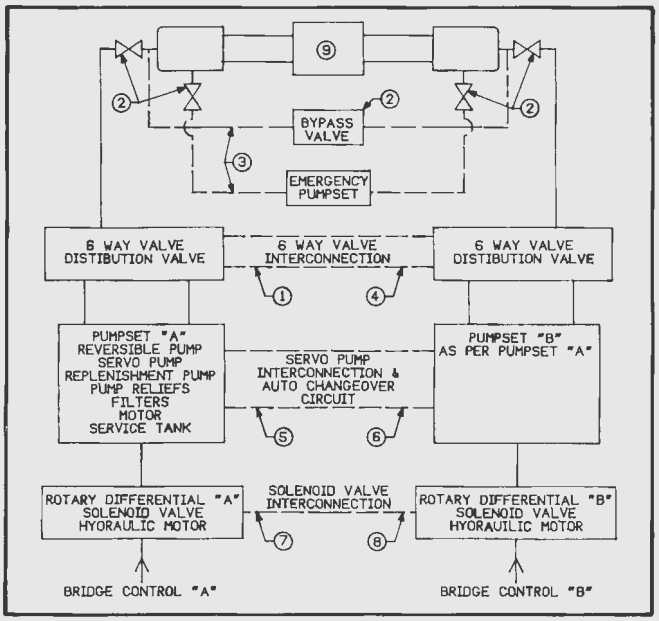

Figure 1. Block diagram of standard naval system.

made. The output flow of the pump was connected to a single or dual pair of rams which acted through a crosshead (Rapson slide) [1] and caused rotation of the tiller and consequently the rudder. This steering system was well conceived since it has remained largely unchanged for over 70 years.

The only significant changes have been the replacement of steam-driven pumps by electric motor-driven pumps, the substitution of hydraulic oil for a glycerine and water mixture, and the use of a rotary electro-hydraulic input in place of the early bridge-connected mechanical shafting.

STANDARD NAVAL SYSTEM

The present naval system [2] which will be installed on the majority of vessels currently under construction is similar to the 70-year-old design. Figure 1 shows a block diagram of the piping.

To activate the steering system, one of the pumpsets is started. For this description, assume pumpset "A" has been activated. The pumpset consists of a reversible-flow variable-displacement piston pump, its replenishment pump and a servo pump. The reversible-flow variable-displacement pump provides directional control of the ram actuator, its replenishment pump makes up for any lost oil in the system, and the servo pump provides pilot pressure for system changeover valves and for positioning of the rotary differential. When up to speed, the servo pump generates pressure which feeds the 6-way valve connecting the pump output to the ram, closing a bypass across the pump. Electrical signals from the bridge control feed the solenoid valve which causes the hydraulic motor to rotate, commanding a steering signal into the rotary differential and shifting the variable-volume pump stroke setting lever through an appropriate direction and magnitude. With the pump on stroke, oil flows through the distribution valves (6-way valve) to the ram and moves the rudder. As the desired rudder angle is approached, the mechanical feedback through the differential returns the pump to zero stroke and awaits the next steering command. This pumpset may be shut down manually or automatically in case of failure and the other pumpset operated. However, this system is vulnerable to failure, either of a temporary nature which can be compensated for by isolating the defective section and activating the remaining part, or of a permanent nature in which the failure cannot be isolated due to failure of a component common to both systems. This type of failure in which one defect could cause irrecoverable loss of steering is known as the single failure point concept [3]. Interconnections which comprise the redundancy of "independent" components and allow single failure are shown by dotted lines on Figure 1.

The Maritime Administration requires steering systems similar to standard naval systems on large commercial vessels. Therefore with these complicated and vulnerable systems in mind, records of steering gear failures were evaluated and recommendations were made to increase the reliability, safety, performance, and integrity of these steering systems. The resultant report [4], contributed to by author, P. Wagner, concludes the following:

1. The rotary differential is unduly complex and the source of numerous steering gear failures. This consists of a dual loop control with the bridge electronic loop controlling the steering gear mechanical loop to achieve the desired rudder angle. The report recommends that the rotary differential be eliminated and the single loop control be designed to enable the bridge electric control to directly control steering gear flow with feedback directly from the rudder shaft to the bridge. The single loop control would reduce steering delays and improve accuracy as well as reliability.
2. It was determined that steering systems designed to work at pressures under 2,000 psi had over eight times the reliability of those operating at over 2,000 psi. (Typical naval systems operate at 2500-3000 psi.) Design of components for operation below 2,000 psi was recommended.
3. Variable-volume reversible-flow pumps were found to cause many failures due to their mechanical complexity and low tolerance to small amounts of oil contamination. Their reliability was further compromised by requiring charge pumps or replenishment pumps, the failure of which could cause failure of the main pump. It was recommended that fixed-displacement pumps of the screw or gear type be used instead.
4. Rapson slide ram systems result in high side loading, which decreases theoretical steering gear efficiency and increases wear. Other types of actuators should be considered.

These recommendations, although specifically for commercial systems, are directly applicable to the stan-

Component Failure	MTBF	Average Failures Per Operating Year
1. 6-way valve shifting malfunction	11,281	.777
2. Ram shutoff and bypass valves (131,400 ea./5 valves)	26,280	.333
3. Ram piping	30,000	.292
4. 6-way valve interconnecting piping	60,000	.146
5. Auto changeover circuit and switches	50,000	.175
6. Servo pump interconnection piping	60,000	.146
7. Solenoid valve leakage failure	131,400	.067
8. Solenoid valve interconnection piping	60,000	.146
9. Yoke and tiller failure	131,400	.067
Total	4,077	2.15

Reliability $= e^{-T/MTBF} = e^{-8760/4977} = 0.117$

$Q = 1 - R = 0.883$

$\frac{1}{Q}$ = Probability that 1 ship out of a number of ships will have a steering failure in 1 year,

$\frac{1}{Q}$ = 1.131, i.e. 1 ship out of 1.13 will have a steering failure in one year of operation.

Figure 2. Reliability analysis of standard naval system.

dard Navy design. Aside from the documented failures of this design, the following additional failures have occurred: (See Figure 1)

1. Failure of the auto changeover cylinder or its piping causes permanent loss of steering since both servo pumps from each "independent" pumpset have a common connection at this point.
2. Failure of solenoid valves or their interconnected piping causes permanent loss of steering since both servo pumps have a common connection point.
3. Failure of ram shutoff valves or ram piping lines causes loss of steering.
4. Failure of any one of the 6-way valves or of servo pumps causes the pumpset to be isolated from the ram and cylinder group, resulting in steering loss.
5. Failure of pump relief valves result in bypass of the ram or loss of pump pressure causing steering failure. Location of these valves down a length of piping from the rams limits their ability to handle high rates of rudder overload with probable ram damage or burst piping in case of heavy rudder impact.
6. Failure of replenishment pumps causes steering failure.
7. Failure of servo pumps causes steering failure.
8. Reversible flow from the main pumps has no filtration on its suction or discharge lines, greatly increasing maintenance and decreasing reliability due to the potential passage of contamination through the pump.
9. Contamination of filters allows total bypass, permitting contamination to damage the pumps and valves.

Failures 3 and 4 are the most serious contribution to low reliability, and no correction has thus far been implemented. It should be clear from the analysis that many more serious reliability problems with present steering gear design would occur if it were not for regular maintenance and parts replacement by skilled personnel. This, of course, has a high operational cost. The reliability analysis of this system (Figure 2) indicates that due to the single point failure modes described above, an average of one ship out of 1.13 will have an unrecoverable steering failure in one year of operation. A maintainability analysis as shown in Appendix 1 indicates that an average of 266 hours per operating year is required to keep the system in operation.

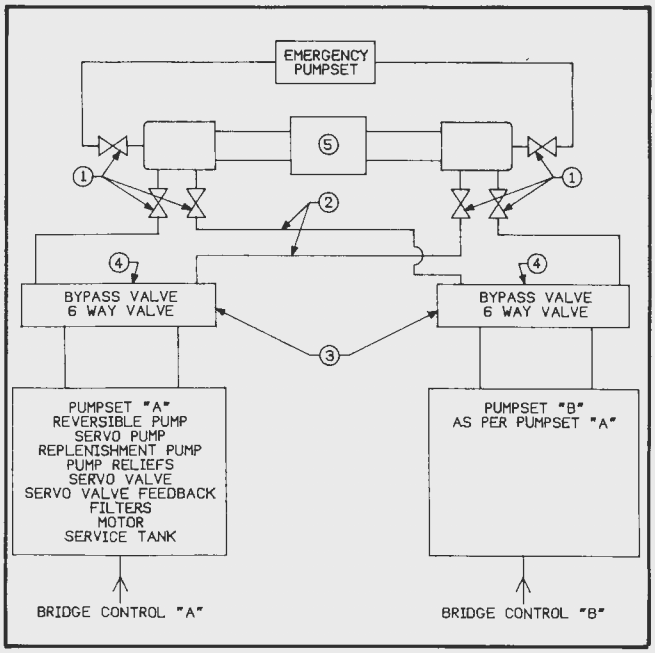

Figure 3. Block diagram of NAVSEA design concept.

LATEST NAVSEA DESIGN CONCEPT

The standard naval system is clearly a very old design which has been modified on an ad hoc basis over decades to improve reliability and minimize maintenance problems. As a result of increasing concern over the failures experienced and inherently poor reliability of the system, and to provide a more responsive system for use on rudder roll stabilization systems (RRS), a new design has been specified by NAVSEA. RRS is a relatively new technique for use on twin-rudder vessels in which a composite electronic command is given to the steering gear, so that the independently controlled rudders may be used in unison for conventional steering, and alternatively with a computer-generated signal which simultaneously gives both steering corrections and non-synchronous signals to dampen the roll of the ship. The roll period of the ship is of much shorter duration than the steering response period, therefore much faster rudder response times are required for effective RRS operation. Retention of the traditional rotary differential with its inherent slack and inertia problems is

Component Failure	MTBF	Average Failures Per Operating Year
1. Ram shutoff and bypass valves (131,400 ea. ÷ 6)	21,900	0.400
2. Ram piping	30,000	.292
3. 6-way valve oil leak	60,000	.146
4. Bypass valve oil leak	60,000	.146
5. Yoke and tiller failure	131,400	.067
Total	8,333	1.051

Reliability $= e^{-T/MTBF} = e^{-8760/8333} = 0.35$

$Q = 1 - R = 0.65$

$\frac{1}{Q} = 1.537$, i.e. Probability that 1 ship in every 1.54 will have a steering failure on one year.

Figure 4. Reliability analysis of NAVSEA design.

not possible. This requires changing to a servo valve-controlled variable volume pump, thus providing a single loop control as previously recommended for commercial ships [4]. Recognizing the importance of repurging the damaged steering system [6,7] in which an oil and air mixture prevents locking or accurate rudder positioning by the actuator, an additional pumpset was specified. While serving as a maintenance fill-and-drain system, it is also used to fill the steering lines, with the steering gear in operation and the vessel underway. This system is shown in Figure 3.

This new system is a definite improvement over the standard naval system, since most of the interconnections which could cause single failure point have been eliminated. However, a number of potential single failure points remain. Of most concern is penetration of the cylinder by piping at six points. Failure of these pipes or the shutoff valves connected to the pipes and valves beyond could allow oil to escape from the ram and cause a failure similar to that which resulted in the loss of the super tanker *Amoco Cadiz* [6]. While it might be argued that the shutoff valve could be closed (if it had not itself failed) by the time the failure could be detected and maintenance personnel directed to the steering gear room, sufficient oil would have been lost to leave the steering gear unrestrained in a seaway and difficult to refill. Restoration of steering could not be considered immediate, and the failure would be considered as unrecoverable for all intents and purposes.

The reliability analysis as shown in Figure 4 indicates that an average of one ship out of 1.54 would have an unrecoverable steering failure in a year of operation. The maintainability analysis as shown in Appendix 2 indicates that 268 hours per operating year is required to keep the system in operation.

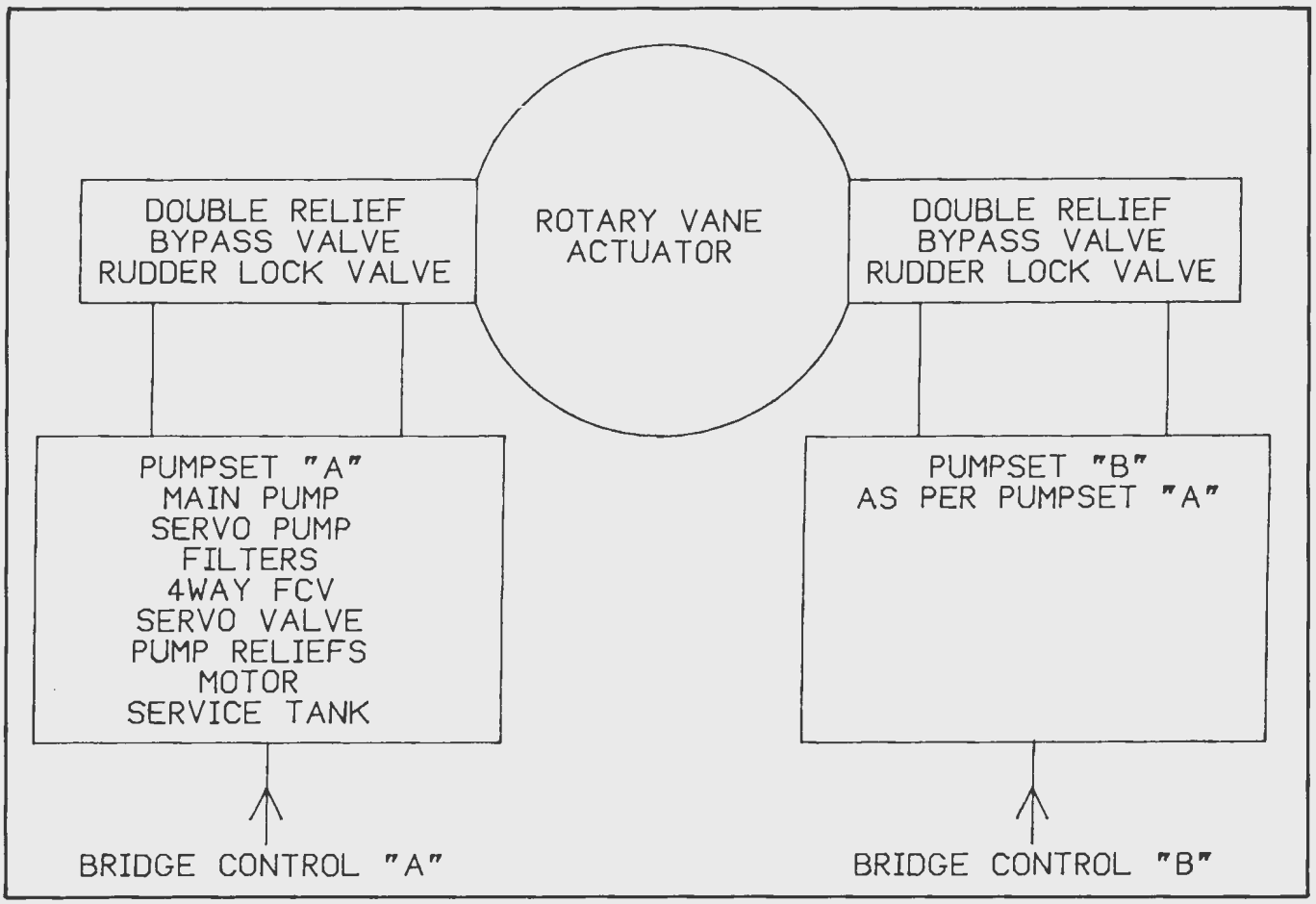

Figure 5. Block diagram of rotary vane design.

Component Failure	MTBF	Average Failures Per Operating Year
Rotary Vane Actuator Failure	131,400	.067
Manifolded Lock Valve Oil Leak	60,000	.146
Total	41,186	.213

Reliability $= e^{-T/MTBF} = e^{-8760/41186} = 0.808$

$Q = 1 - R = 0.192$

$\frac{1}{Q} = \frac{1}{.192} = 5.2$, i.e. Probability that 1 ship in every 5.2 ships will have a steering failure in one year.

Figure 6. Reliability analysis of rotary vane design.

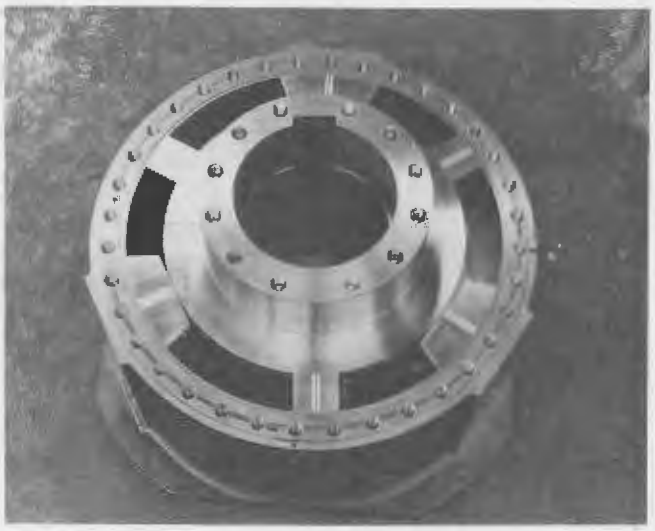

Figure 7. Rotary vane actuator construction.

ROTARY VANE STEERING SYSTEM

Clearly a better system is desired than even the latest NAVSEA design. Taking into consideration previous recommendations [4] for increasing steering system reliability and starting with a fresh design concept, a far simpler, more reliable, easier-to-maintain and less costly system has evolved. This system is presently being supplied to the Canadian patrol frigate program and has been proven in commercial use for seven years. The method of operation is illustrated in Figure 5.

The system operates as follows. Either pumpset "A" or "B" is selected by starting the electric motor. Assuming pumpset "A" is started, its motor then drives the main steering pump and servo pump. With no signal from the bridge, oil from the fixed volume pump flows to the pressure-compensated flow control valve and through its open center back to the reservoir at no pressure loss and with no heat generation. The pressure-compensated 4-way flow control valve is a hydraulic directional valve constructed such that unidirectional oil flow from a constant volume pump may be reversed in direction and modulated in amplitude by simply shifting the valve spool. Being pressure compensated, the flow is modulated regardless of pressure variations in the system such that output flow is directly proportional to spool displacement. This is similar to a conventional servo valve except that in the centered position (no output flow) oil is returned to the service tank at no pressure loss. This not only eliminates the power loss associated with high level servo valves, but provides a self-purging capability, as any air in the system returns to the service tank and is automatically purged.

When a proportional electronic steering command is sent from the bridge and fed directly to the servo valve, a differential pressure is applied to the spool ends of the pressure-compensated 4-way flow control valve causing it to shift a proportional amount, thereby modulating flow from the main pump into the steering lines. These lines are connected to the rotary vane steering actuator via a manifold. This flow automatically opens the lockvalve (double pilot-operated check valve) admitting oil into the actuator and causing it to rotate to the desired position. Due to the unique design of the pressure-compensated flow control valve, system pressure only rises to meet the actuator pressure and returns to zero pressure at null. As the desired rudder angle is approached, a rudder feedback transmitter reduces the servo valve drive, decelerating the actuator so that it arrives at the exact angle with no overshoot, shock, or noise. At this time, flow to the rotary vane actuator is stopped, and the lockvalve automatically closes, positively holding the rudder at all loads up to a maximum as determined by the double relief valve setting.

If at any time, due to battle damage or accidental hydraulic failure, oil should be lost from the system, the loss is confined to the defective section since the lockvalve prevents any oil from leaving the rotary vane actuator. In this case, steering is immediately restored by stopping the defective pumpset and starting the standby pumpset. No valve need be manually operated, and recovery is made in a matter of seconds; specifically the time required for the second pumpset to attain operating speed. And, no oil is lost from the actuator and repurging is not required. Alternatively, the standby pumpset may remain in operation with no signal fed to its servo valve. In this case, it can be activated immediately upon failure of the main pumpset. Figure 6 summarizes the reliability analysis for the rotary vane system.

Individual components of this system deserve further explanation. Figure 7 illustrates the rotary vane actuator's construction. Not only does it achieve direct rotation of the rudder without side loads or friction of any kind (other than seal friction), it includes a radial and carrier bearing automatically lubricated by the working oil. This coupled with the compact envelope of the rotary vane actuator greatly reduces the vertical and horizontal space requirements over separately mounted bearings and also eliminates the necessity of a separate greasing system as can be seen in Figure 8. Since the rotary vane actuator develops pure torsion when oil pressure is applied to the tiller vanes, a mechanical effi-

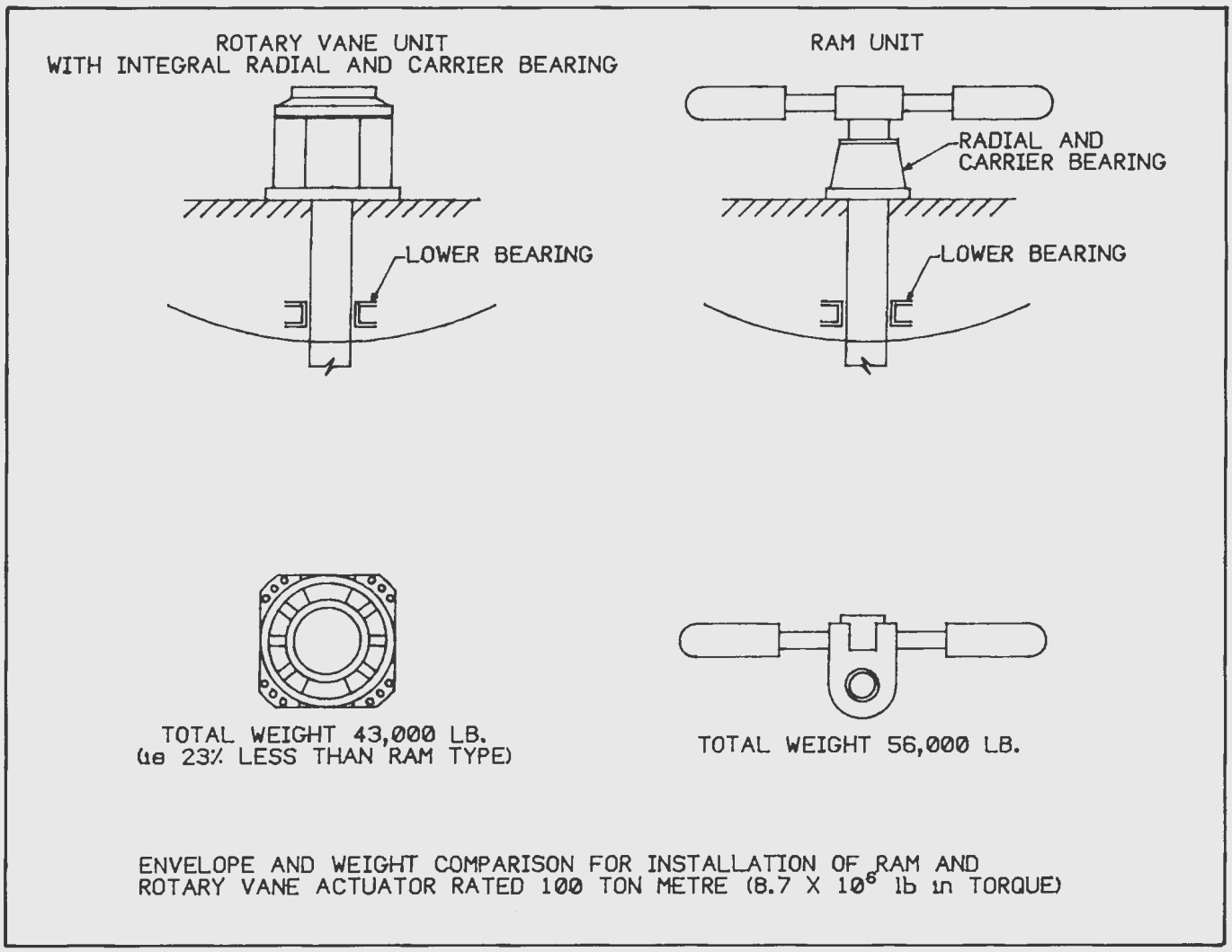

Figure 8. Rotary vane vs ram envelope and weight comparison.

ciency of 95 percent has been confirmed by actual sea trials. This is very high when compared to the 70 percent allowable with the Rapson slide system [2]. Because the only moving component is the tiller, wear, lubrication, and maintenance requirements are less than the more complex ram system. Due to its compactness and symmetrical distribution around the rudder stock, the rotary vane has a much higher inherent shock resistance. Since all working parts of the actuator are totally enclosed within the oil-filled housing, corrosion of working parts is eliminated. The housing further provides excellent protection from damage due to accidental impact with working tools, such as may happen when a wrench is dropped on an exposed ram, and provides much greater protection against battle damage from projectiles in the steering compartment. Due to direct rotation of the rotary actuator, rudder angles of 90 degrees port and starboard are possible, allowing extremely high maneuverability when using high-lift rudders such as Schilling rudders. This is especially useful on minesweepers or oceanographic research vessels.

A unique hold-down arrangement consisting of welded chocks and hold-down bolts eliminates costly and troublesome fitted bolts and machined and fitted beds. On installation, the rotary vane self-aligns itself to the rudder stock taper, and chocking compound is floated up to its mounting flange from a rough machined support table. Chocks are then butted up to the rotary vane mounting flanges and welded to the support table. The hold-down bolts are sized for loads well in excess of any shock requirements, but in the event of severe grounding with consequent excessive vertical rudder forces, these bolts are allowed to fail, permitting the actuator to rise vertically inside the chocks until the rudder stock jump collar bottoms without any internal damage occurring. When the excessive loads are removed, the rotary vane is then placed back in the chocks and spare hold-down bolts fitted in short order. The total enclosure of working components in the rotary vane, coupled with its high angle of rotation, suits it ideally to use outside the pressure hull of submarines for controlling rudders and diving planes.

The use of high-load replaceable bearing liners in the radial and carrier bearing allows higher than normal reaction loads to be accommodated such as may occur with spade rudders having a very short distance between the upper and lower bearings. Oil flow into and out of the rotary vane is ported through two manifolded rudder lockvalves (double pilot-operated check valves) which provide several unique features. When the rotary vane has been positioned to the required rudder angle, these valves automatically shut off, holding any rudder load up to that determined by the setting of the double relief valves. On the other hand, conventional ram systems have their variable delivery pumps constantly exposed to rudder loads and due to leakage inherent in the pumps must always be slightly on stroke to generate a load holding pressure and flow to make up for rudder creep. This causes unnecessary wear of the pumps, increases power consumption and degrades the rudder positioning accuracy. A further and much more important feature of the rotary vane system lockvalves is their ability to automatically isolate the pumpsets or the piping, tanks, and valves, etc. from the actuator. Any damage to the integrity of the pumpset and piping system allows only the damaged part of the system to lose oil. The lockvalves prevent any oil from leaving the actuator. With the typical ram system, similar damage will let oil drain out of the damaged pumpset or piping and the ram. With reference to the NAVSEA system in Figure 3, while shutoff valves and bypass valves are fitted to isolate damage, they are local manual valves which can not be shut off in time to avoid ram oil loss. To regain steering, the damage control party must go to the steering compartment, diagnose the problem, and close the appropriate valves. By this time, most oil will have leaked from the ram, and it will be oscillating wildly in any kind of seaway [6]. Under these conditions, and especially with the reversible flow pump which tends to reciprocate air back and forth in the system, purging and restoring steering is extremely difficult, if not impossible, under battle conditions. The use of the additional purging pumpset in Figure 1 has not been proven in practice to be effective when used simultaneously with the main pumpset as required by specifications [5]. Not only does the lockvalve automatically isolate the pumpsets from the actuator, it automatically isolates the pumpsets from each other without the necessity of complex sequence valve changeover as required by 6-way valves in the typical ram system (Figure 1). Clearly a much simpler, more effective and more reliable system is achieved with lockvalves directly manifolded to the rotary vane actuator.

Integral with the lockvalve at the rotary vane manifold is the double-acting relief valve. By coupling these valves directly to the actuator and adequately sizing them, extremely high rudder overloads can be handled without developing undue pressure at high impact rates. In case the valves should fail, either mechanically or due to contamination, steering would be lost in a typical ram system, but the rotary vane system is provided with manual 90 degree turn shutoff valves which allow rapid isolation and recovery of steering. Even though one of the double-acting relief valves is temporarily out of service, the other one is still operating at the second manifold and can handle overloads at one-half the maximum system rate.

Several unique features in the pumpset design provide further simplification in operation, with increased reliability, less heat and less noise. Paramount is the use of the pressure-compensated servo-valve controlled 4-way flow control valve. This valve accepts oil from a fixed-displacement pump and, regardless of load pressure variations, modulates its output flow proportional to spool displacement. As a result, much quieter fixed-displacement pumps, such as screw or gear types, can be used which offer noise reductions of between 15 and 20 db over conventional variable-volume reversible-flow piston pumps. These fixed-displacement pumps are much more tolerant of contamination and thereby provide far longer life with minimal maintenance. By designing the operating pressure to be less than 2,000 psi, additional improvements in reliability as previously documented [4] are achieved.

When no steering command is fed to the 4-way flow control valve, the flow from the main pump is returned via its open center back to the reservoir, providing not only a cooling path for the oil which eliminates the need of heat exchangers in many cases but also provides a means of settling out any air which may get in the system. Air in a reversible pump system tends to oscillate back and forth across its output lines to the ram without means of automatic escape. Exactly this problem caused the loss of the super tanker *Amoco Cadiz* [6]. The pressure-compensated 4-way valve is pilot operated by proportional differential pressure from the low-pressure servo valve, such that steering signals from the bridge control result in modulated flow to the rotary vane, accelerating and decelerating movement over the first and last 3 degrees of travel and providing smooth, accurate, and shock-free positioning in the same manner as a reversible-flow pump, but without its complicated parts and maintenance problems. Operating at a low pressure of 400 psi, the servo valve has a high tolerance to contamination, providing long life and giving better frequency response than a servo-controlled variable volume pump [10]. This makes it attractive for vessels requiring high rudder rates and accurate rudder positioning, such as RRS applications.

In case of total loss of the duplicated bridge control system, local electric control of the servo valve is possible as well as manual control of the valve. Local manual hydraulic helms may also be installed to provide rudder positioning and steering at reduced speeds and rudder rates even though all electro-hydraulic power may be lost.

A single-loop control is employed in the rotary vane system, greatly simplifying the steering gear control with consequent increase in reliability [4]. In comparison, the conventional two-loop ram system requires "on/off" electrical step signals to be fed to a solenoid valve which steps a hydraulic motor to drive the input of a rotary mechanical differential steering control which then compares the input command against the rudder

	Standard Naval System	Latest NAVSEA Design Concept	Rotary Vane Design
Number of single point failure modes	9	6	2
MTBF (hours)	4,077	8,333	41,186
Average failures per operating year	2.15	1.051	0.213
Reliability	0.117	0.35	0.808
Average maintenance manhours per year of operation	266.	268.	26.

Figure 9. Comparison of reliability and maintainability characteristics.

position through a helical rotary feedback. As may be appreciated, this is a mechanically complex system having potential reliability problems and requiring considerable maintenance. Furthermore, the solenoid valve step input is not capable of accurate positioning due to its tendency to limit-cycle or hunt at fast rudder rates. This, coupled with the inevitable slack buildup in the rotary differential, feedback, linkage, and pump swashplate linkages, causes a significant degradation in positional accuracy and speed of response. With the single-loop system, proportional signals from the bridge directly modulate flow to the steering actuator which has electronic rudder feedback transducers and limit switches. This simple direct control is a much more reliable, more accurate, quieter, simpler to maintain, less expensive, and lower weight system than the newer NAVSEA single-loop design. (Even the new NAVSEA design is not a true single loop since the pump swash plate has a second servo system to position it.)

COMPARISON OF RELIABILITY AND MAINTAINABILITY CHARACTERISTICS

The reliability and maintainability characteristics for the three steering systems are summarized in Figure 9.

The reliability analysis is based on the mean time between failures (MTBF) for the identified single point failure modes. Only components which can cause single point failure and unrecoverable steering are included in the calculations. The MTBF data is derived from Navy 3M repair data, service records maintained by a manufacturer and data used for reliability analysis in the Canadian patrol frigate program.

The results of the reliability analysis indicate that the rotary vane steering system is 2.3 times more reliable than the new NAVSEA design concept and 6.9 times more reliable than the standard naval system. This is primarily due to the use of fewer and more reliable components in a hydraulic circuitry which minimizes the possibility of a single fault failure disabling the entire steering system.

The maintainability calculations are documented in Appendices 1 through 3 and are based on the mean time between repair actions (MTBRA). The MTBRA is based on maintenance actions due to catastrophic failures. A catastrophic failure is defined as the inability of the part to operate thereby preventing direct steering gear operation or preventing changeover to a redundant (backup) component. The average maintenance hours is estimated for one year of operation to be 266 hours for the standard navy system, 268 hours for the new NAVSEA design and 26 hours for the rotary vane system. Maintenance costs for the rotary vane design are greatly reduced due to a reduction in components and use of components which inherently have a larger MTBRA. Furthermore, use of cartridge valves and manifolded components greatly reduce MTTR by replacement of parts instead of repair.

CONCLUSION

The new NAVSEA steering gear concept [5] is a step in the right direction for simplification of design and improvement of response but does not improve reliability or maintainability to the degree desirable for mission-essential equipment. Compared to the rotary vane system, it requires excessive space, is heavier, and is more vulnerable to battle damage. The basic ram design with variable-volume reversible-flow pumps has been modified for more than 70 years in an attempt to improve its operation and it would appear that little more can be done to optimize it.

A newer approach has been taken in design, unhampered by past endeavors and traditional concepts, resulting in a now proven system which has superior performance, greater compactness, lower weight, and higher reliability with less maintenance than existing naval systems.

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[6] Interim decision of the Commissioner of Maritime Affairs, R.L. and Interim Report of the formal investigation by the Marine Board of Investigation in the matter of loss by grounding of the VLCC *Amoco Cadiz*, O.N. 4773. Published by the Bureau of Maritime Affairs, Monrovia, Liberia.

[7] Hay, P.L. "Developing Technology for Improved Steering Gear Reliability," Great Lakes and Great River Section, the Society of Naval Architects and Marine Engineers.

[8] Reliability Analysis Center Publication NPRD-2 "Non-Electronic Parts Reliability Data," Rome Air Development Center.

[9] NAVSEA 3M Repair Data.

[10] *Fluid Power Handbook and Directory (1986-87)*, Cleveland, Ohio, Penton/IPC Publishers.

APPENDIX 1

Maintenance Requirements Analysis
Standard Naval System

Component	MTBRA (Hours)	Average Number of Repair Actions per Year of Operation A	MTTR (Manhours) B	Average Maintenance Hours per Year of Operation A × B
Variable volume pump and replenishment pump	7,854	1.11	4	4.44
Servo pump	15,000	.584	2	1.17
Pump motor	50,000	.175	5	.88
Replenishment pump accumulator	30,000	.292	2	.58
Servo relief valve	20,777	.422	1	.42
Replenishment relief valve	20,777	.422	1	.42
Main relief valves	20,777	.422	1	.42
6-way valve	11,281	.775	4	3.10
Servo filter	5,000	1.75	1	1.75
Replenishment filter	5,000	1.75	1	1.75
Solenoid valve	29,412	.298	1	.30
Rotary differential	24,200	.362	10	3.62
Rotary motor	20,000	.439	1	.44
Hydraulic lines and fitting	12,092	.725	2	1.45
Auto changeover cyl. and switches	20,000	.439	4	1.76
Shut off and bypass valves (5 of 100 K each)	20,000	.439	1	.44
Ram packing	9,438	.928	16	14.85
Yoke and tiller	35,477	.247	120	29.64
Emergency pumpset	10,000	.876	3	2.63
Radial and carrier brg.	100,000	.087	150	13.1
Greasing bearings 1/2 hr/day				183.
Total				266.

NOTE: MTBRA = Mean Time Between Repair Action
MTTR = Mean Time to Repair

APPENDIX 2
Maintenance Requirements Analysis
NAVSEA Design

Component	MTBRA (Hours)	Average Number of Repair Actions per Year of Operation A	MTTR (Manhours) B	Average Maintenance Hours per Year of Operation A × B
Variable volume pump and replenishment pump	7,854	1.11	4	4.44
Pump feedback linkage	30,000	.292	2	.58
Pump feedback pot and amplifier	6,000	1.46	2	2.92
Servo pump	15,000	.584	2	1.17
Pump motor	50,000	.175	5	.88
Replenishment pump accumulator	30,000	.292	2	.58
Servo relief valve	20,777	.422	1	.42
Replenishment relief valve	20,777	.422	1	.42
Main relief valves	20,777	.422	1	.42
6-way valve	11,281	.775	4	3.1
Servo filter	3,000	2.92	1	2.92
Replenishment filter	5,000	1.75	1	1.75
Servo valve	3,000	2.92	1	2.92
Hydraulic lines and fittings	12,092	.725	2	1.45
Shut off valves (6 of 100 K hrs each)	16,666	.525	1	.525
Ram packing	9,438	.928	16	14.85
Yoke and tiller	35,471	.247	120	29.64
Emergency pumpset	10,000	.876	3	2.63
Radial carrier brg.	100,000	.088	150	13.2
Changeover pressure switches	30,000	.292	1	.292
Greasing bearings 1/2 hr/day				183.
Total				268.

APPENDIX 3
Maintenance Requirements, Rotary Vane System

Component	MTBRA (Hours)	Average Number of Repair Actions per Year of Operation A	MTTR (Manhours) B	Average Maintenance Hours per Year of Operation A × B
Main pump	20,000	.438	2	.876
Servo pump	20,000	.438	2	.876
Pump motor	50,000	.175	5	.875
Main relief	66,667	.131	1	.131
Servo relief	66,667	.131	1	.131
Servo valve	3,000	2.92	1	2.92
4-way flow control valve	50,000	.175	2	.350
Double relief	87,600	.100	1	.100
Shut off and bypass valve	175,200	.05	2	.100
Hydraulic lines and fitting	20,000	.438	1	.438
Main filter	5,000	1.75	1	1.75
Servo filter	3,000	2.92	1	2.92
Rotary vane seals	17,000	.515	25	12.88
Rotary vane radial and carrier brg.	131,400	.066	20	1.32
Total				25.67