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SURFACE COLD ROLLING OF MARINE PROPELLER SHAFTING

By

L. L. Shook, Jr. and C. L. Long

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Introduction

Surface cold rolling (called either surface rolling or cold rolling) of marine propeller shafting to increase the fatigue resistance was first performed in January 1956 on the propeller shafting of the Aircraft Carrier U.S.S. RANGER (CVA61) by the Newport News Shipbuilding and Dry Dock Company. This event, which came after nearly thirty years of development of surface working procedures, was precipitated by the failure of the port tailshaft of the U.S.S. NORFOLK(DL-1) and the loss at sea of the tapered end of the propeller shaft and the propeller(9).[3] (A typical propeller and propeller shaft assembly is shown in Figure 1).

Prior to this failure, surface cold rolling of propeller shafting was being considered by the marine industry as a means to reduce shaft condemnations and failures. The shaft failure on the U.S.S. NORFOLK which occurred after the vessel had been in service less than two years and was the first failure at sea of a propeller shaft on a Naval vessel incident to normal operation excited considerable concern on the part of the Navy. As a result, the first equipment for surface cold rolling marine propeller shafting was developed by Newport News Shipbuilding and Dry Dock Company at the request of the Navy.

The design of the surface cold rolling equipment developed by Newport News was derived from the experience of the American Railroad industry in surface cold rolling railroad axles. The American railroads were among the first to take practical advantage of surface cold rolling, using this procedure to increase the fatigue resistance of railroad axles as early as the late 1930's.

1 - Senior Design Supervisor, Machinery Design Department, Newport News Shipbuilding and Dry Dock Company

2 - Engineering Technical Department, Newport News Shipbuilding and Dry Dock Company

3 - Numbers appearing in ( ) refer to the Bibliography

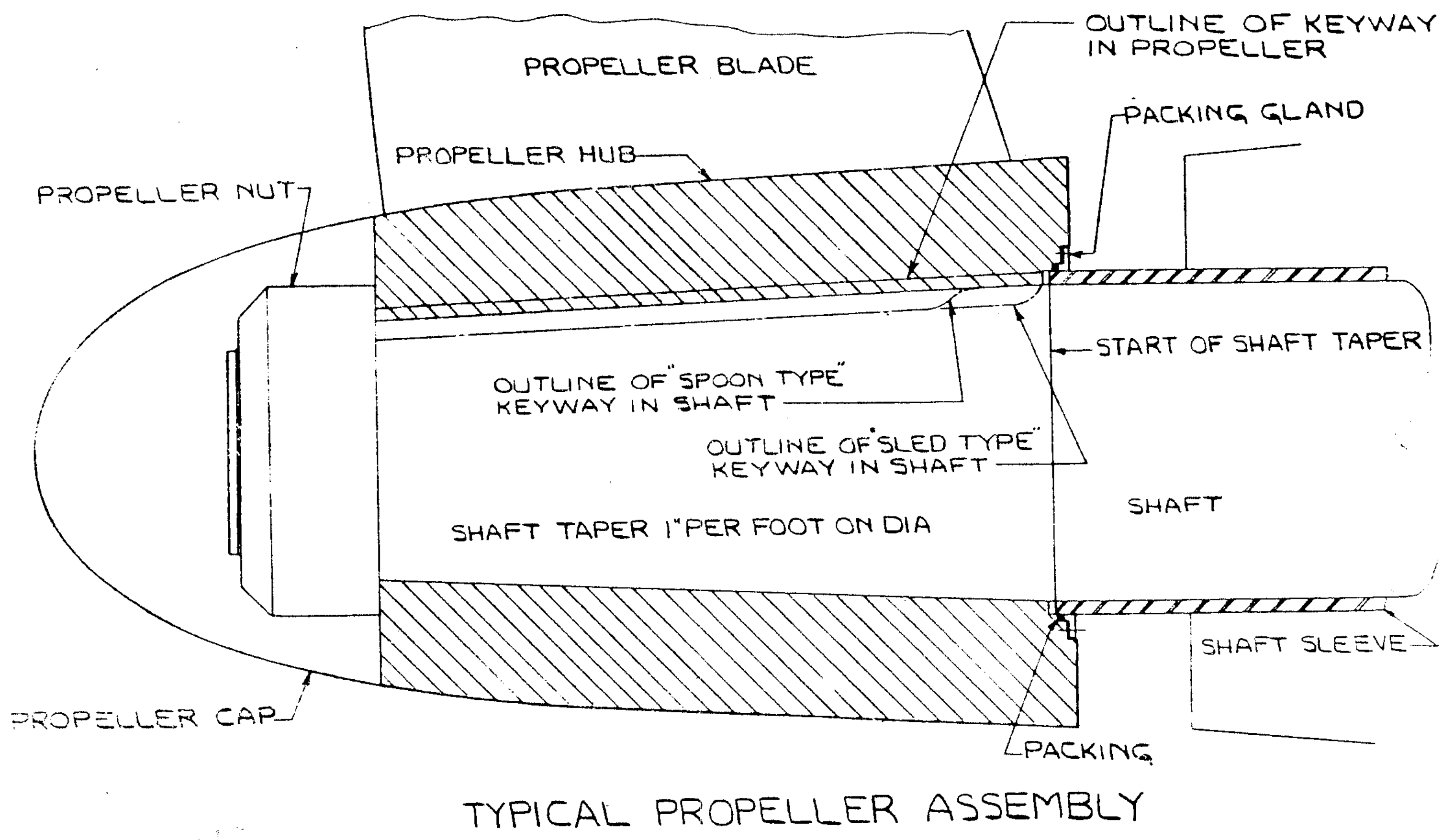


FIGURE 1 – TYPICAL ASSEMBLY OF PROPELLER AND PROPELLER SHAFT.

The work done by the American railroads was under the direction of Dr. Horger* and Mr. Neifert*, both of whom gave considerable assistance to the Newport News Shipyard in perfecting the equipment and procedure for rolling marine shafting.

A considerable portion of the available information concerning the effects of surface cold rolling has been derived by Dr. Horger and Mr. Neifert (1)(2). This information plus information available from other investigators, together with the experience gained in rolling service shafting is the basis for the conclusions and recommendations of this paper.

The paper presents a description of the equipment and procedure used for surface cold rolling marine propeller shafting and the results and experience gained thus far in this work. A review of test data is included to give an understanding of the effects of surface cold rolling. The major conclusions and recommendations made in this paper are:

1. There are no detrimental effects resulting from surface cold rolling of shafting (properly done) and there are definite advantages.

2. All new marine propeller shafting should be surface cold rolled and consideration should be given to surface cold rolling existing shafting.

3. The length of shafting subjected to surface cold rolling should be as follows:

   (a) For propeller shafting housed in a stern tube - one shaft diameter in length, or not less than 18 inches, forward of the forward end of the shaft taper to one third the length of the shaft taper aft of the forward end of the taper.

   (b) For propeller shafting housed in a strut - same as above plus 12 inches aft of the forward end of the after sleeve to 6 inches forward of the forward end of the after sleeve.

4. The use of surface cold rolling does not permit relaxing the requirement that the shaft not be returned to service with a crack.

5. The period between service inspection of propeller shafts should remain three years until more is known about surface cold rolling.

* Dr. Horger is Chief Engineer; Mr. Neifert is Supervisor of Railway Research: both of the Timken Roller Bearing Company, Canton, Ohio

Equipment

Marine propeller shafting to be surface cold rolled is mounted in a large lathe. The rolling equipment is mounted on the lathe bed, attached to the lathe carriage, so that it may be moved along the length of the shaft to perform the rolling operation. All rolling equipment developed to date for marine propeller shafting uses two rollers opposed to one another on opposite sides of the shaft to be rolled. (Rolling equipment with three rollers spaced about 120° apart around the shaft has been used by the railroad industry for rolling shafting with diameters up to about 15 inches. This three roller equipment has not been found suitable for marine shafting due to restricted lathe clearances involved with the larger diameter shafts).

The surface cold rolling equipment developed by the Newport News Shipbuilding and Dry Dock Company for rolling marine propeller shafting is shown in figures 2, 3 and 4. This equipment use two 9 inch diameter rollers, one a "hardening" roller with a 1-1/2 inch contour radius and the other a "smoothing" roller with a 4 inch contour radius. (The contour radius is the radius in the plane of the longitudinal axis of the shaft). The "hardening" roller bears on one side of the shaft with the load necessary to produce the desired depth of cold work and is opposed on the opposite side of the shaft by the "smoothing" roller. (The maximum roller load capacity of the Newport News equipment is about 44,000 pounds). The "hardening" roller is mounted on the end of a hydraulic piston which is prevented from rotating to keep the roller axis aligned parallel with the shaft centerline. The "smoothing" roller is fixed in its housing. The housing for each roller is mounted on an unrestrained cross slide to permit the rollers to follow a change in diameter of the shaft. The two housings are tied together by a large adjusting screw at the foot of the roller pedestals to permit adjusting the rolling rig for various diameter shafts. This rolling equipment will accommodate propeller shafting up to 30" in diameter. The stroke of the ram will permit a maximum change in diameter of a shaft of about 8 inches for any setting of the adjusting screw.

The hydraulic system for developing the roller loading is shown in figure 5. The system includes an accumulator charged with nitrogen for absorbing fluctuations in pressure. A hand pump is provided for maintaining pressure as the ram moves in to suit change in shaft diameter when rolling the shaft taper and to accommodate leakage.

Each roller is mounted on two Timken roller bearings #H414210 cup and #H414835 cone, as shown in figure 6. Both rollers have a face width of 1-1/2 inch. The rollers are made of Crucible Steel Company's "Air Kool" tool steel with the following chemical composition: C-.95

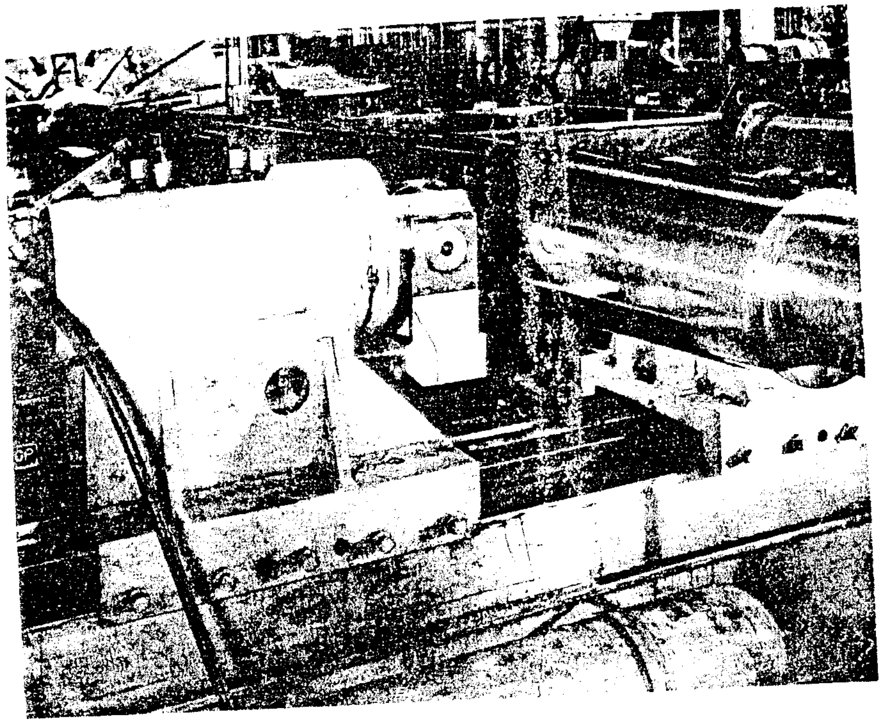


FIGURE 2 – NEWPORT NEWS ROLLING EQUIPMENT SHOWN SURFACE COLD ROLLING A PROPELLER SHAFT FOR THE AIRCRAFT CARRIER U.S.S. RANGER (CVA 61) – "HARDENING" ROLLER ON NEAR SIDE.

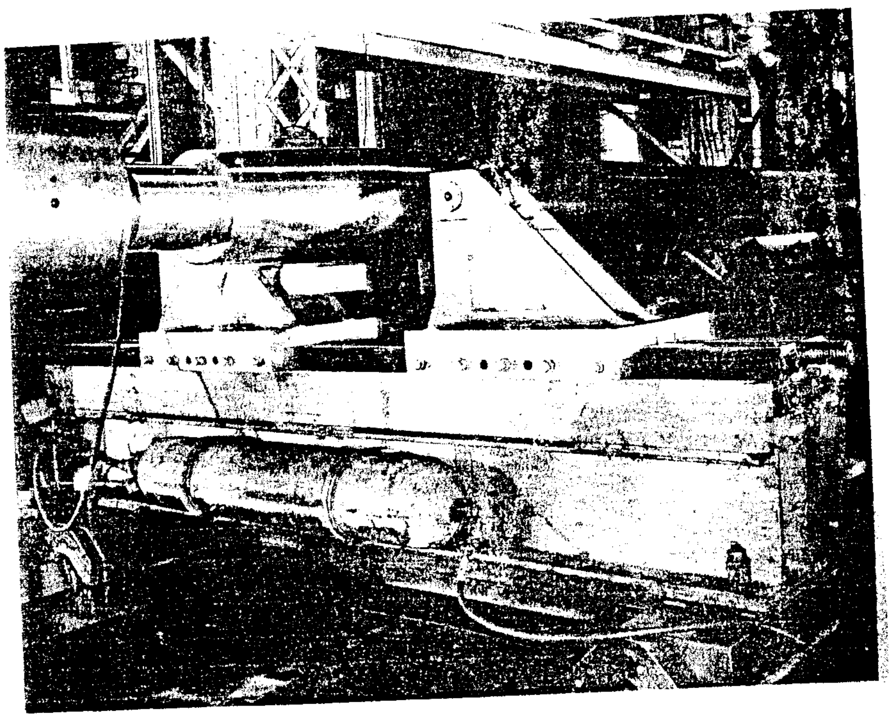


FIGURE 3 – NEWPORT NEWS ROLLING EQUIPMENT SHOWN SURFACE COLD ROLLING A PROPELLER SHAFT FOR THE AIRCRAFT CARRIER U.S.S. RANGER (CVA 61) – "SMOOTHING" ROLLER ON NEAR SIDE.

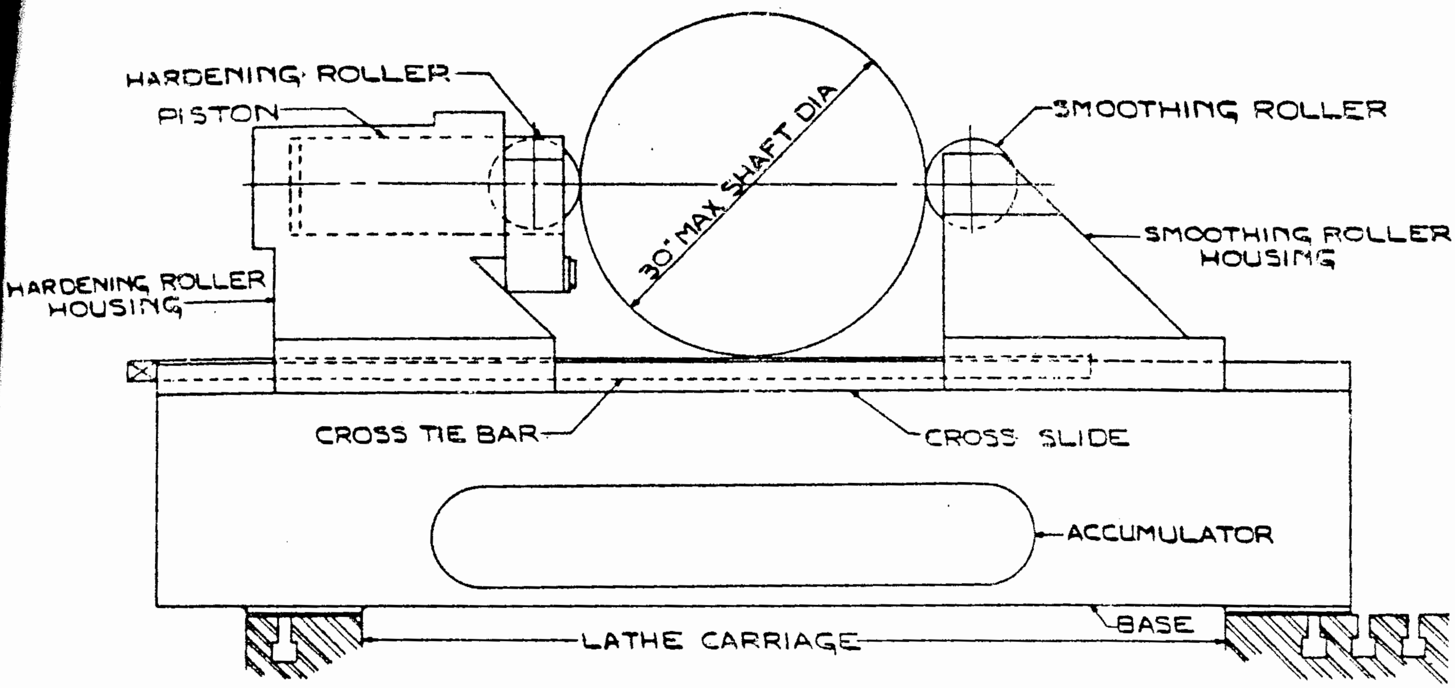


FIGURE 4 – GENERAL ARRANGEMENT OF NEWPORT NEWS SURFACE COLD ROLLING EQUIPMENT.

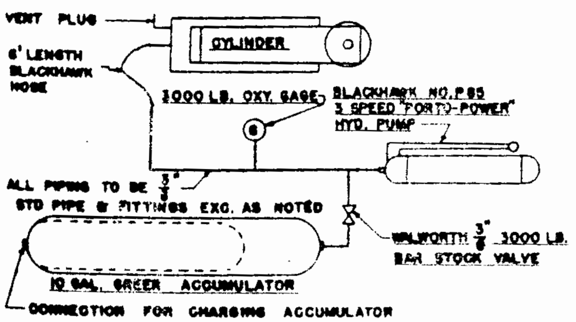


FIGURE 5 – ARRANGEMENT OF HYDRAULIC SYSTEM FOR NEWPORT NEWS SURFACE COLD ROLLING EQUIPMENT.

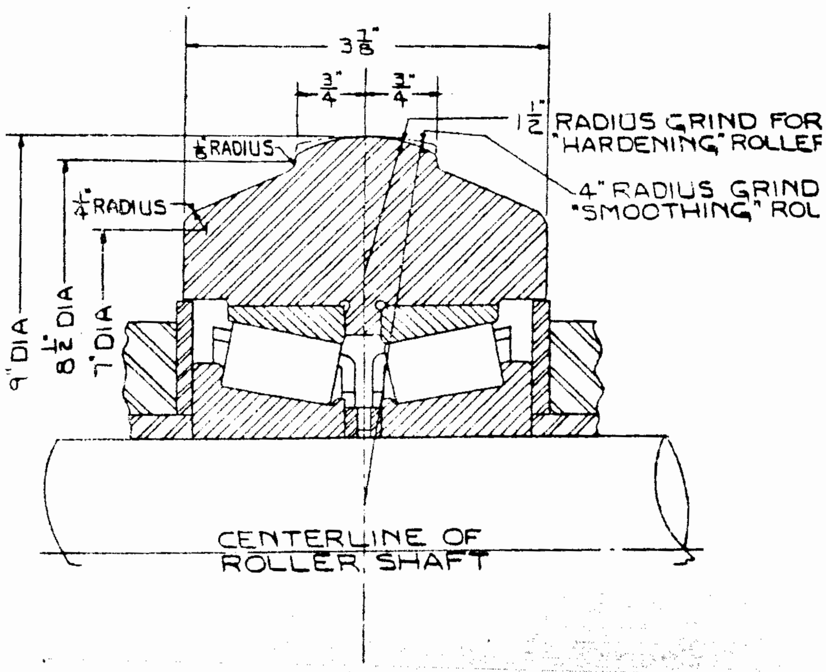


FIGURE 6 – DETAILS OF NEWPORT NEWS "HARDENING" AND "SMOOTHING" ROLLERS.

to 1.00, Cr - 5.25 min., V - .50 min., Mo - 1.15 min. The rollers were hardened to 57-58 on the Rockwell "C" scale by being heated to 1750 - 1850°F, cooled in air, tempered at 1000°F for two hours, cooled in air, tempered at 950°F for two hours and cooled in air.

Some difficulty was encountered with the design of these rollers, two sets having failed before the final set was made. The first set was 8 inches in diameter made of low grade carbon tool steel (C-0.75 to 0.85) case hardened about 3/16 inches deep to 58-62 on the Rockwell "C" scale, but identical in all other respects to the final set. The "hardening" roller of this set failed by flattening out under the roller load. The second set was also 8 inches in diameter made of regular high speed 18-4-1 tungsten tool steel oil hardened to 58-62 on the Rockwell "C" scale and identical in other respects to the final set. The "hardening" roller on this set broke under the roller load.

The rollers are installed in the rolling equipment with their centers (both horizontal and vertical) directly in line. When rolling the shaft taper, due to the different contour radii of the two rollers, the point of contact of the rollers on the shaft is not in line in the vertical plane. No difficulty has been experienced with this feature when rolling shaft tapers up to 1-1/2 inches per foot on the diameter. To suit rolling tapered shaft sections, the contour and face width of the "smoothing" roller must be selected so that the load on the roller will not be on the edge of the roller for the steepest taper to be rolled.

Similar equipment developed by the Bethlehem Shipbuilding Division at their Quincy yard is shown in figures 7, 8 and 9. This equipment is quite similar to that developed by Newport News, the principle differences being the tie rod above the shaft and the more complex hydraulic system. The material used by Bethlehem for the rollers for two sets of their rolling equipment is "Lehigh H" steel (C-1.55, Cr-11.50, Mo-0.80, V-0.40) and that for the rollers for the third set of their equipment is "66 High Speed" steel (C-0.83, Cr-4.15, W-6.35, V-1.90, Mo-5.00). All of these rollers were heat treated to obtain a Rockwell "C" hardness of 61-62. The Bethlehem equipment is rated at a maximum roller load of 44,000 pounds.

Erie Forge Company has developed rolling equipment, figure 10, that has both rollers mounted on hydraulic cylinders. This allows the rolling frame to be fixed with only one degree of movement, which is in the longitudinal direction. Rolling capacity of the equipment is 37,000 pounds for shaft diameters from 7 to 30 inches.

Bath Iron Works has developed a high capacity machine which can be used for rolling high strength alloy shafting, if ever necessary. This surface rolling equipment will apply a rolling load of 60,000 pounds on shaft diameters up to 24 inches. The general arrangement

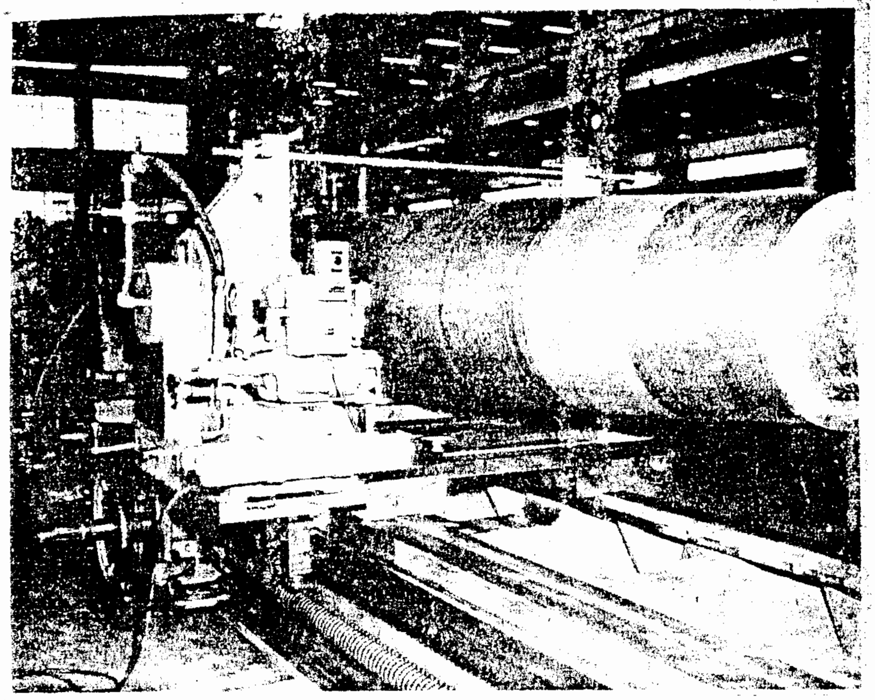


FIGURE 7 – BETHLEHEM ROLLING EQUIPMENT SHOWN ROLLING A PROPELLER SHAFT FOR THE AIRCRAFT CARRIER KITTYHAWK (CVA 64) – "HARDENING" ROLLER ON NEAR SIDE.

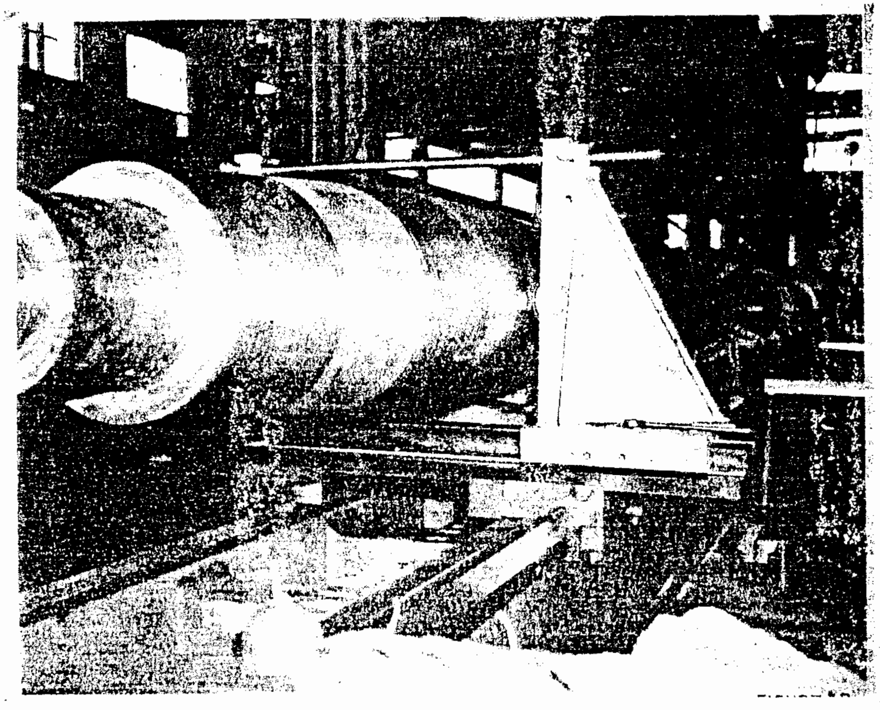


FIGURE 8 – BETHLEHEM ROLLING EQUIPMENT SHOWN ROLLING A PROPELLER SHAFT FOR THE AIRCRAFT CARRIER U.S.S. KITTYHAWK (CVA 64) – "SMOOTHING" ROLLER ON NEAR SIDE.

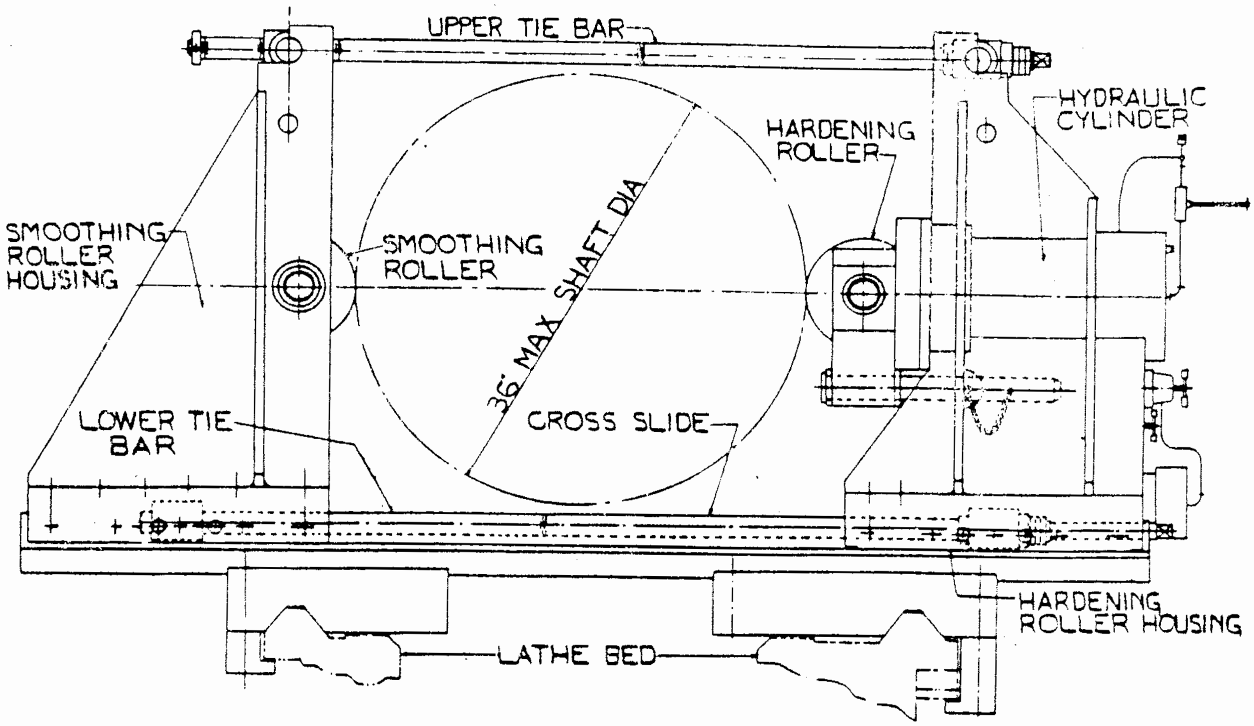


FIGURE 9 – GENERAL ARRANGEMENT OF BETHLEHEM SURFACE COLD ROLLING EQUIPMENT.

FIGURE 10 – SURFACE COLD ROLLING EQUIPMENT DEVELOPED BY ERIE FORGE AND STEEL COMPANY.

of this equipment is similar to the Bethlehem equipment.

All the other surface rolling equipment in the country to be used in rolling marine shafting is believed to be similar to the Newport News design.

Procedure

The rolling operation is performed in one pass with the rolling equipment moving along the length of the shaft at a rate of feed of about 1/16 inches per revolution of the shaft while the shaft is turned with a shaft surface speed of about 60 feet per minute. The maximum rate of feed of the rolling equipment along the shaft per shaft revolution should never exceed one half the width (minor semi axis) of the contact area of the "hardening" roller along the longitudinal axis of the shaft. The minor semi-axis of the contact area versus roller load and shaft diameter is given in the Appendix.

The surface speed of the shaft is limited by the rate of plastic flow of the shaft material and if too great, will cause the roller to jump. The shaft material builds up as a wave ahead of the "hardening" roller similar to a tire running in soft earth. If the surface speed is greater than the rate of plastic flow of the shaft material, the roller will jump the wave. This wave is easily observed while the shaft is being rolled. It is also suspected, but not known to be true, that surface speeds even less than required to cause jumping will leave fine microscopic cracks in the surface. The upper limit of the surface speed to avoid this condition, if it exists, is not known. The 60 feet per minute used by Newport News does not produce any of these effects.

When a shaft is to be rolled, the shaft diameter is left .005 inches to .010 inches greater than the required finish diameter and with a finish of about 125 micro-inches. The rolling operation decreases the diameter about .001 inches to .002 inches. After rolling, a finish cut is taken to reduce the shaft to the required finish diameter. (It is important that this cut be not more than necessary to avoid loss of the beneficial effects of surface cold rolling. Newport News practice is to limit this cut to not more than .005 inches on the radius). The roller load is applied gradually for a distance of 2 inches along the shaft ahead of the area to be rolled and decreased gradually for a distance of 2 inches along the shaft after the area to be rolled. The area to be rolled is kept smeared with an extreme pressure lubricant during rolling, although a good machine oil might serve as well.

The roller load used is a function of shaft diameter and material, roller diameter and contour radius and desired depth of cold working.

Calculations and curves for determining the roller load are given in the Appendix.

Rolling should start on the shaft and progress towards the taper so that the taper will be rolled "down hill". Down hill rolling of the taper is considered better procedure for the following reasons:

1. In rolling down the taper, the edge of the smoothing roller will not dig into the shaft even if contact is near the roller edge.

2. The force required to move the rollers along the shaft and the resulting load on the rolling equipment is less when rolling down the taper than when rolling up the taper.

3. Rolling down the taper is safer, since if the hydraulic cylinder or carriage slide should jam, the rollers move away from the work and not into it.

It has not been found necessary to round off the knuckle at the start of the shaft taper to suit the rolling operation.

In new construction the keyway (s) is (are) cut in the taper after the surface rolling is completed. No surface cold working of the keyway has been done by Newport News after cutting the keyway. The location of the keyway on new shafting ("spoon type" keyway Figure 1) places it well aft of the start of the shaft taper and out of the critical area in the shaft subject to cracks and failure. It is felt that with this condition, little gain would be realized by surface cold working the keyway(by peening or other means) and that cutting the keyway does not reduce the advantages gained by surface cold rolling.

For surface cold rolling existing shafting, it is necessary to fill the keyways with dummy keys flush with the shaft surface, filling in with weld metal as necessary and recutting the keyways after the rolling operation is completed. For existing "sled type" keyways (Figure 1) extending to the forward end of the shaft taper(and into the critical area on the shaft), it is suggested that this operation incorporate a change to the "spoon type" keyway. For "sled type" keyways which are not changed to "spoon type" keyways, surface cold working the forward end of the keyway after rolling the shaft may offer some advantage. It is interesting to note that tests (3) on Model tailshafts both with the "sled type" keyway and with the "spoon type" keyway showed no gain when subjected to reversed bending fatigue tests. These tests shafts were not subject to torsion and the result might have been different had torsion been present.

Any welding done to a shaft requires heat treatment after welding to remove the undesirable residual stresses caused by welding. This heat treatment should be done before rolling for two reasons:

1. To reduce any tendency of the shaft to go eccentric when rolled as described below.

2. To prevent the removal by heat treatment of the desirable effect of cold rolling.

When ordering forgings for tailshafts that are to be rolled, the specifications should require that special precautions be taken by the forging manufacturer to minimize residual stresses in the shaft forgings. This precaution is based on experience with shaft eccentricity caused by rolling, believed due to residual stresses in the shaft before rolling.

Two shaft forgings which developed eccentricity at Newport News when rolled were found to have been straightened while the forgings were still warm from the tempering treatment without being returned to the tempering furnace. It is not definitely known that this was the cause of this eccentricity and it can only be surmised that the residual stresses in the forgings contributed to it. (One thing that seems to refute this is that cutting one keyway in a rolled shaft, which contains residual stresses on the surface due to rolling, caused no eccentricity). Consultation with the steel supplier pointed out the desirability of specifying minimum residual stress forgings.

The two shafts mentioned above had been cold rolled for 12 inches on each side of the start of the taper and developed an eccentricity of .0135 inches in one shaft and .009 inches in the other shaft.(If a greater rolling length had been used more eccentricity of the shaft would have been expected). The eccentricity of the two shafts was corrected by shifting the aft turning center and taking a finish cut as shown in figure 11. From figure 11 it can be seen, had the two shafts in question been rolled for a length much greater than 12 inches on each side of the start of the shaft taper, this correction could not have been made. No other simple reliable method of correcting shaft eccentricity is known. Extreme care should be taken concerning this tendency towards eccentricity until more is known about it. It is suggested for shafts which are to be rolled for long distances (say the entire length of the after strut), that the concentricity of the shaft be checked several times during the rolling operation. To make this check, the roller pressure should be removed by removing the load gradually for a distance of two inches. When re-establishing the roller load, the load should be applied gradually for two inches of the length already rolled. To avoid the possibility of eccentricity when rolling shafting that has been in service, consideration should be given to heat treating the shafts to remove any residual stresses that might remain from initial forging operations, straightening, etc. If any eccentricity does develop, remachining of the shaft

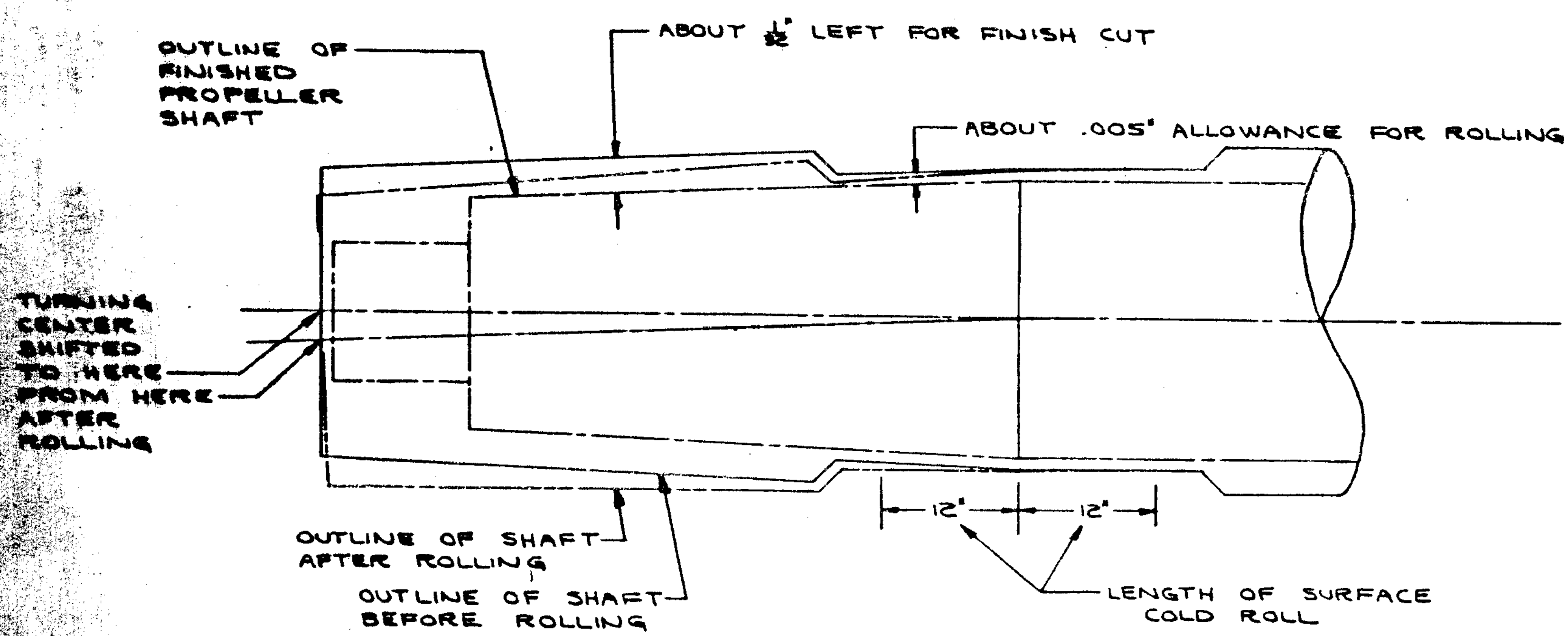


FIGURE 11 – SKETCH SHOWING METHOD OF CORRECTING ECCENTRICITY ENCOUNTERED DUE TO SURFACE COLD ROLLING THE PROPELLER SHAFT OF AN ESSO TANKER.

to remove this eccentricity could reduce the diameter to the point where the shaft would have to be discarded.

Information is needed concerning the tendency of a shaft to go eccentric when rolled and the relationship of this tendency to any residual stresses in the shaft before rolling. (Eccentricity caused by surface cold rolling should not be confused with that resulting when shrinking on a shaft sleeve. Eccentricity resulting when shrinking on a shaft sleeve is not uncommon in the marine industry and is corrected by peening the sleeve).

The length over which a tailshaft is to be rolled has not been officially recommended for merchant shafting. This length should be sufficient to encompass the length of shafting subjected to fretting. Serious fretting occurs adjacent to and on either side of the start of the shaft taper, and all failures start within this area. To suit this condition, it is recommended that merchant shafting be rolled for a length forward of the start of the shaft taper equal to 1 shaft diameter or a minimum of 18 inches and for a length aft of the start of the shaft taper equal to 1/3 of the length of the shaft taper.

On ships with outboard struts with sleeves fitted in way of these struts, fretting can occur at both ends of the sleeve. Benefit from surface rolling can be obtained by surface rolling at both ends of the sleeve. The Bureau of Ships has specified that Navy shafts be rolled continuously from 2 inches forward of the after end of the taper to 12 inches forward of the forward end of the after shaft sleeve. This length of cold rolling reduces the possibility of correcting any eccentricity by machining in the manner described in figure 11. Furthermore, the greater rolling length would tend to increase the amount of any eccentricity encountered. To minimize these conditions it is suggested that the tailshaft be rolled at the after end as described above and that the shaft in way of the forward end of the sleeve be rolled for 12 inches under the sleeve and to a point 6 inches forward of the sleeve.

When designing new shafting which is to be surface cold rolled, provision should be made to allow for future surface cold rolling made necessary by stress relief, etc. This would require that allowance be made for reduction in size of the shaft due to the compressing action of the roller and skim cut following rolling. Service data in years to come may show that re-rolling shafting is never necessary and that this provision can be dropped.

Results

The first surface cold rolling operation on a marine propeller shaft was performed on one of the propeller shafts of the Aircraft

Carrier RANGER (CVA-61) in January 1956. The first surface cold rolled marine propeller shaft placed in service was that of the Esso Tanker GETTYSBURG which started its service life in March 1957. (The RANGER did not go in service until August 1957).Since the normal inspection interval of shafts is once every three years, we do not expect to have any service data much before 1960.

Preceding the surface cold rolling of the shafting on the RANGER, a short section of shafting identical to the RANGER shafting in diameter, material and shaft taper was test rolled using the same roller load designated for the RANGER shafting. It was with this section of test shaft that the Newport News equipment was proven and on which the two sets of rollers failed. Following the final rolling test on this section of shafting, specimens were cut from the test shaft to determine the radial hardness traverse and the residual stress. The radial hardness traverse taken on the test specimens is illustrated in figure 12 and shows a depth of cold work of about 0.45 inches. This compares well with the calculated depth of cold work of 0.50 inches. The residual stresses determined from the test shaft appeared to be erroneous when compared with reliable residual stress results published by other investigators and are not included in this paper. The residual stresses were determined using strain gages mounted on the circumference of the shaft. A ring 1-3/4 inches wide by 1-1/8 inches thick containing these strain gages was cut out of the shaft surface and then split into two pieces to permit removal from the shaft. The ring halves were cut into small segments containing the strain gages and these segments were reduced in thickness by a series of slicing cuts made on the surface opposite the strain gages. Strain gage readings were taken after each operation. The only benefit of this residual stress analysis was an indication of the residual compressive stresses present.

A summary of shafts rolled by Newport News and others giving roller load used, degree of eccentricity encountered, etc. is shown in table 1.

A report received from the Bethlehem Shipbuilding Division of Quincy, Massachusetts, advised that their first surface cold rolling operation was conducted in October 1957, and that they experienced no difficulty with eccentricity of the shaft due to rolling. This operation by Bethlehem was performed on one of the propeller shafts of the Aircraft Carrier KITTY HAWK (CVA-63) for the full length of the shaft sleeve and shaft taper. The New York Naval Shipyard has surface cold rolled sixteen shafts for the FORRESTAL (CVA-59)class Aircraft Carriers. Their experience in rolling this shafting was similar to that of Newport News and is shown in Table 1.

No report of the surface cold rolling operation by others is available for inclusion in this paper.

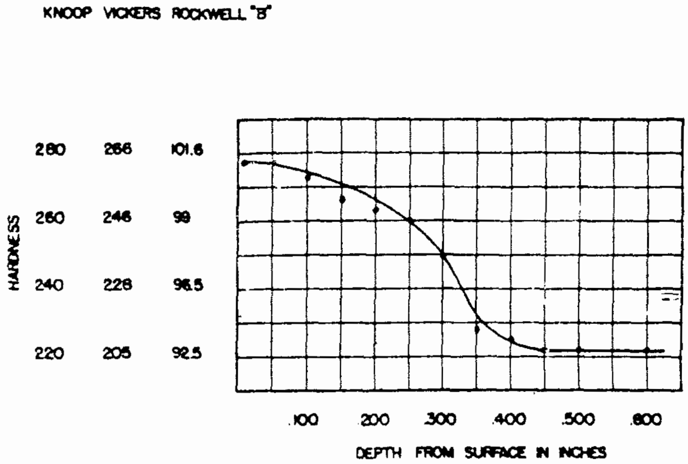


FIGURE 12 – RADIAL HARDNESS TRAVERSE OBTAINED ON A 28 ¼" DIAMETER TEST SHAFT SURFACE COLD ROLLED AT NEWPORT NEWS – INDICATING DEPTH OF PENETRATION OF COLD ROLLING EFFECTS.

PLACE SURFACE COLD ROLLING EXECUTED	SHAFTS ROLLED	SHAFT SIZE	SHAFT MATERIAL	LENGTH ROLLED	CALCULATED DEPTH OF COLD WORK	MAXIMUM REDUCTION IN DIAMETER DUE TO ROLLING	MAXIMUM ECCENTRICITY ENCOUNTERED DUE TO ROLLING	MAXIMUM ECCENTRICITY DUE TO CUTTING KEYWAY AFTER ROLLING	NO. OF KEYS	SHAFT SURFACE HARDNESS (BRINNELL)		DATE FIRST SHIP OF CLASS PLACED IN SERVICE
										BEFORE ROLLING	AFTER ROLLING	
NEWPORT NEWS SHIPBUILDING & DRY DOCK CO	CVA 61 (4 SHAFTS)	2'-4¼" DIA	STEEL (MIL-S-890) CLASS "AN")	24"	½"		.0015"	.001"	2	180	247	AUGUST 1957
	ESSO TANKERS (8 SHAFTS	2'-3⅛" DIA	STEEL (A.B.S. GR. 2)	24"	½"	.005"	.0135"	.002"	1			MARCH 1957
	LST 1173 CLASS VESSELS (6 SHAFTS)	12⅜"	STEEL (CL'B-S' (SPECIAL) MIL-S-890)	9'-2⅛"	½"	.003"	.004"	.001"	2			NOVEMBER 1957
	GRACE LINES PASSENGER VESSELS (6 SHAFTS)	19 9/16" DIA	FORGED STEEL (A.B.S. GR. 2)	24"	½"	.003"	.0015"	.002"	1			NOT YET IN SERVICE
BETHLEHEM SHIPBUILDING DIVISION QUINCY MASS	CVA 63 (ONE SHAFT ROLLED AS OF DATE TABLE PREPARED)	2'-4¼"	STEEL (MIL-S-890 CLASS "AN")	ABOUT 16'-5"	½"	INFORMATION NOT AVAILABLE	FIGURES NOT AVAILABLE BUT LESS THAN AMOUNT MEASURED PRIOR TO ROLLING	INFORMATION NOT AVAILABLE	2			NOT YET IN SERVICE
NEW YORK NAVAL SHIPYARD	CVA 59 CLASS VESSELS (16 SHAFTS)	2'-4¼"	STEEL (MIL-S-890 CLASS "AN")	INFORMATION NOT AVAILABLE	½"	.0015"-.002"	.0025"	INFORMATION NOT AVAILABLE	2	160-170	230-240	INFORMATION NOT AVAILABLE

TABLE #1 SUMMARY OF SURFACE COLD ROLLED MARINE PROPELLER SHAFTS

Review of Test Data

The effect of surface cold rolling shafting is to increase the fatigue strength of the shaft. The marine propeller shaft with its shrunk on sleeve and driven up propeller presents a very low level of fatigue strength. This is illustrated in figure 13. The pressure due to the fit of the sleeve on the shaft and of the propeller on the shaft reaches a peak value at the ends of the sleeve and forward end of the propeller hub due to the end restraint of the protruding shaft, as shown in figure 13a. Under bending conditions, this pressure is aggravated by the ends of the sleeve and forward end of the propeller hub impinging under heavy pressure against the compression side of the shaft as shown in figure 13b. In addition, a minute sliding action of the ends of the sleeve and forward end of the propeller hub on the shaft surface due to the alternate elongation of the shaft surface fibers causes loosening of finely divided virgin material at the surface of the rubbing parts. This action is known as fretting and the resulting corrosion is known as fretting corrosion. In addition the marine propeller shaft and propeller assembly is subject to the admission of sea water inside the propeller hub which causes corrosion of the shaft, resulting in pits in the shaft. These pits act as stress risers from which cracks initate, lowering the fatigue strength of the shaft.

Tests have been carried out by many investigators to determine the increase in fatigue strength gained by surface cold rolling. The results of some of these tests are shown in figures 14 and 15. A summary of test data available is shown in Table 2.

As shown in figures 14 and 15 and in Table 2, the fatigue strength of shafts not subject to fretting or corrosion and those subject to fretting only is considerably greater for a rolled shaft than for a non-rolled shaft. Figure 14 also shows that for shafts subject to corrosion, the number of cycles before fracture (at the same stress) is considerably greater for a rolled shaft than for a non-rolled shaft.

It was found in tests 5 and 6 shown in figure 15 and Table 2 that fretting fatigue cracks were present in both rolled and non-rolled shafts stressed below the endurance strength and that these cracks initiated at extremely low values of stress. The cracks were found in the shaft just inside the wheel fit, after the wheel was pressed off. The relationship of the depth of these cracks to one another, plotted in figure 16, shows that the fatigue crack depth and its propagation in rolled shafts is only a fraction of what was found in non-rolled shafts, basing the comparison on equal stress levels.

Figures 17 and 18 show photomicrographs of cracks which occurred in corrosive fatigue tests 14 and 16 of figure 14 and table 2. These

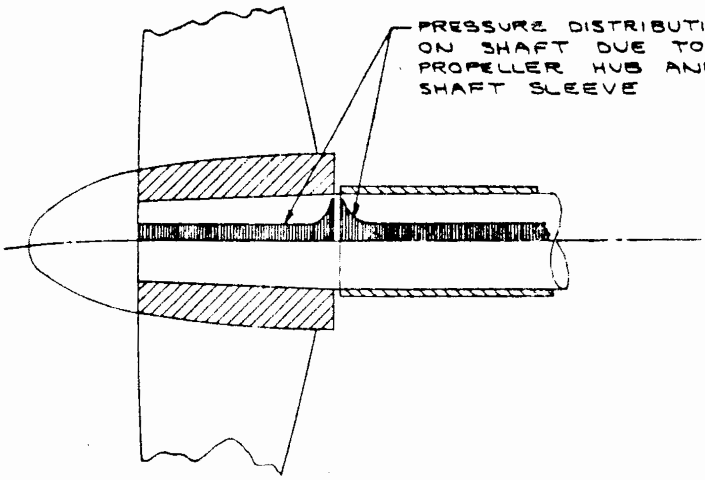


FIGURE 13 a

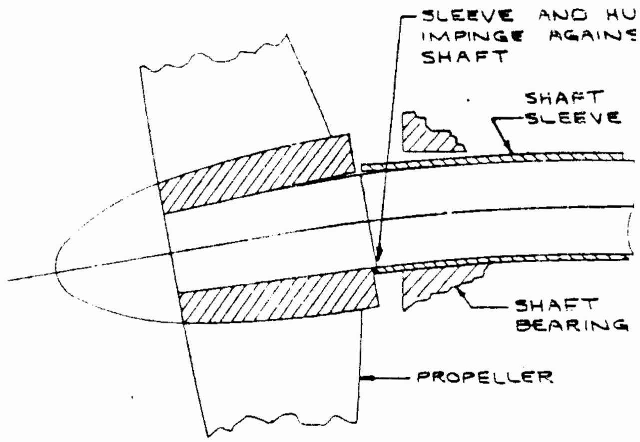


FIGURE 13 b

FIGURE 13 – SKETCH ILLUSTRATING THE FRETTING PROBLEM ENCOUNTERED WITH MARINE SHAFTING.

TEST NO	REF	DIA-TAPER	MATERIAL	TENSILE STRENGTH	SHEAR YIELD (ASSUMED)	CALC DEPTH COLD WORK	DEPTH COLD WORK DETERMINED FROM HARDNESS TRAVERSE	ENDURANCE LIMIT	REMARKS
		IN - IN/FT		PSI	PSI	IN	IN	PSI	
			ROTATING FATIGUE IN AIR WITHOUT FRETTING						
1	8	0.50 - ½	ALLOY 4	141000	59250	—	—	71,000	
2	8	0.50 - ½	ALLOY 4	141000	59250	0.016	0	83000	
3	4	0.468-0	0.487. C STEEL	93000	28000	—	—	39,200	
4	4	0.468-0	0.487. C STEEL	93000	28000	0.044	—	47,200	
			ROTATING FATIGUE IN AIR WITH FRETTING						
5	1	9.50	0.50% C-0.80% MN	96000	25000	—	—	11,000	
6	1	9.50	0.50% C-0.80% MN	96000	25000	0.450	0.550	722000	
7	3	5.75-1	0.30% C-0.41% MN	59 000	12 400	—	—	10 000	SLED TYPE KEYWAY (SEE (3) pg 490)
8	3	5.75-1	0.30% C-0.41% MN	59000	12 400	—	—	10000	SPOON TYPE KEYWAY (SEE (3) pg 498)
9	3	5.75-1	0.30% C-0.41% MN	59,000	12400	.222	0.10 TO 0.18	17,000	SPOON TYPE KEYWAY
10	8	0.750-⅛	ALLOY 4	141 000	59 250	—	—	20,000	
11	8	0.750-⅛	ALLOY 4	141,000	59 250	.015	—	60,000	
12	4	0.468	0.48% C STEEL	93 000	28 000	—	—	17,800	
13	4	0.468	0.487. C STEEL	93000	28000	.055	—	36,100	
			ROTATING FATIGUE SUBJECTED TO CORROSION WITHOUT FRETTING						
14	8	0.50-½	ALLOY 4	141000	59250	—	—	—	NO ENDURANCE LIMIT 4.2 X 10^6 CYCLES @ 20,000 PSI - ONE TEST SEVERN RIVER WATER AND ONE TEST SEA WATER
15	8	0.50-½	ALLOY 4	141000	59.250	0.016	0	—	NO ENDURANCE LIMIT 3.8 X 10^7 CYCLES @ 20,000 PSI - SEVERN RIVER WATER
16	8	0.50-½	ALLOY 4	141000	59250	0.026	0.007	—	NO ENDURANCE LIMIT - 1.21 X 10^8 CYCLES @ 20,000 PSI - SEVERN RIVER WATER

TABLE #2 SUMMARY OF TEST RESULTS OBTAINED WITH SURFACE COLD ROLLED SHAFT SPECIMENS

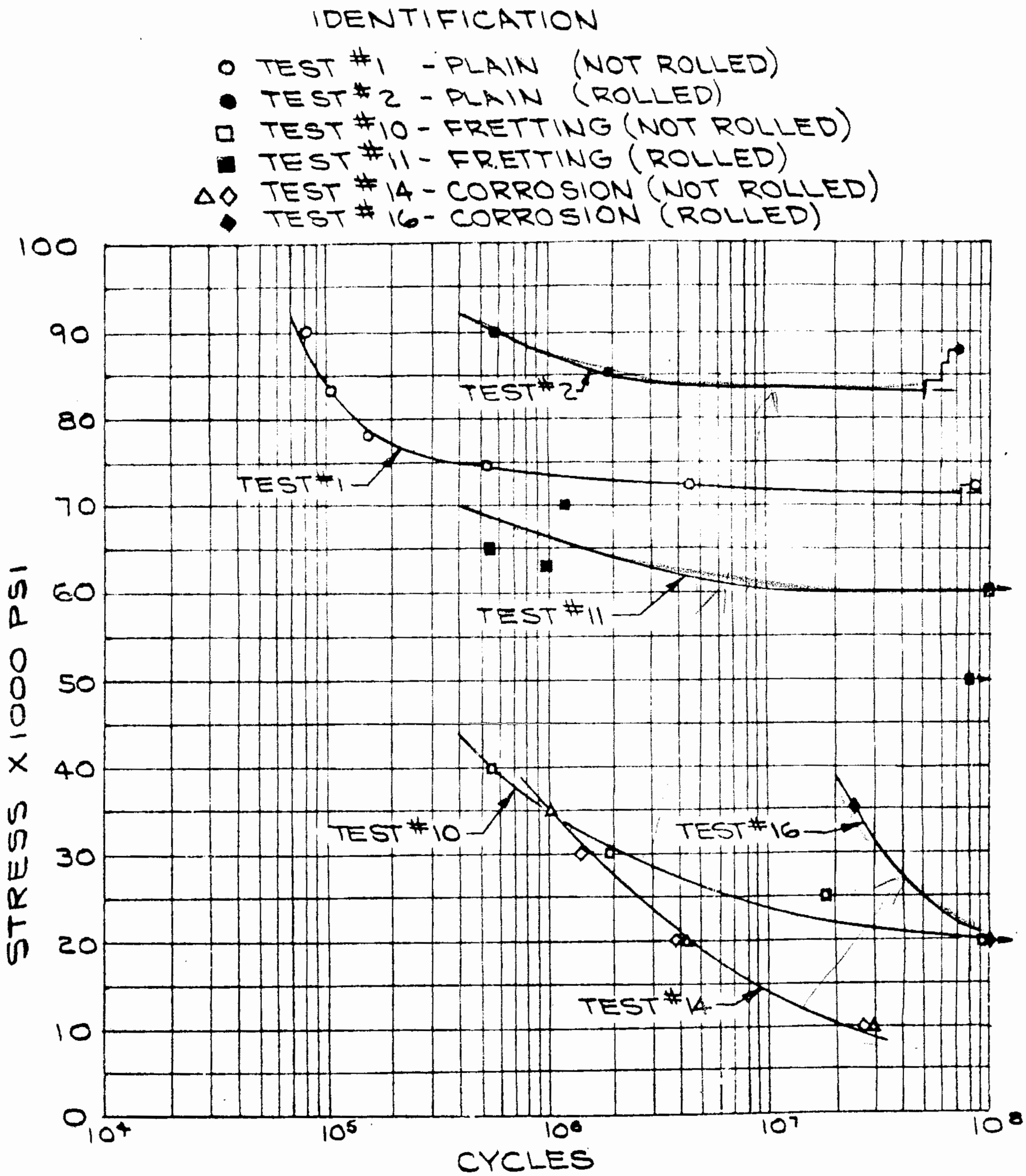


FIGURE 14 – FATIGUE TEST RESULTS OBTAINED BY THE UNITED STATES NAVAL ENGINEERING EXPERIMENT STATION (8) – FOR IDENTIFICATION OF TEST NUMBERS, SEE TABLE 2.

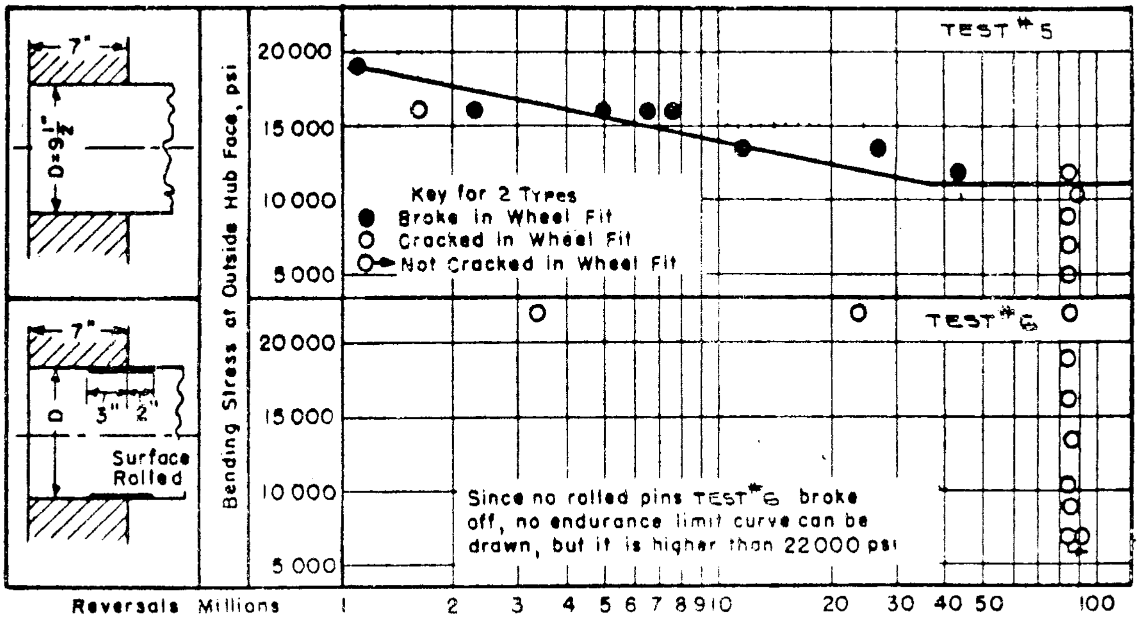


FIGURE 15 – FATIGUE TEST RESULTS OBTAINED BY THE TIMKEN ROLLER BEARING COMPANY (1) – FOR IDENTIFICATION OF TEST NUMBERS, SEE TABLE 2.

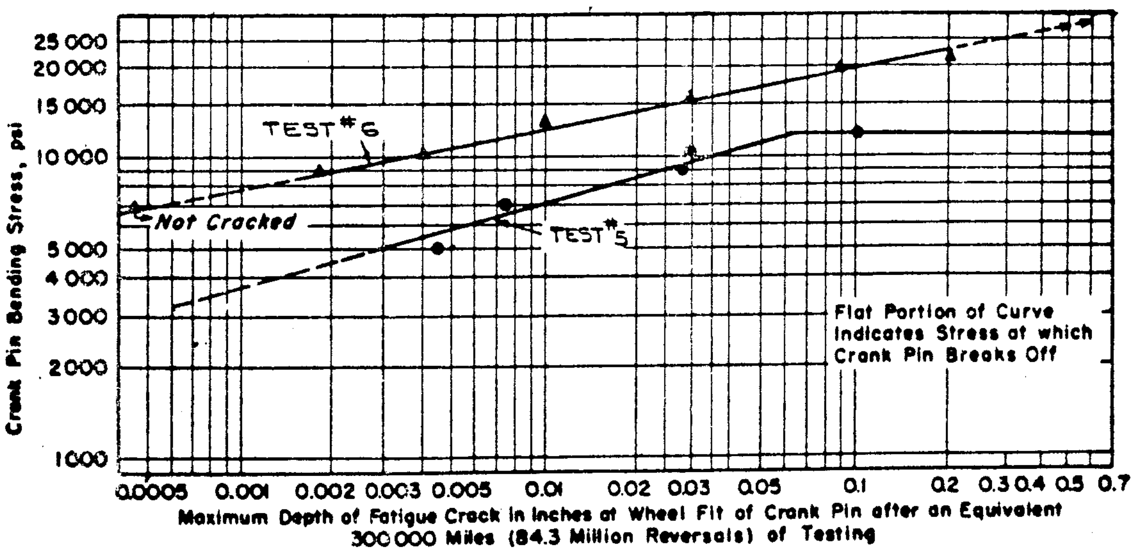


FIGURE 16 – COMPARATIVE PROPAGATION OF FATIGUE CRACKS (1) IN TESTS 5 AND 6.

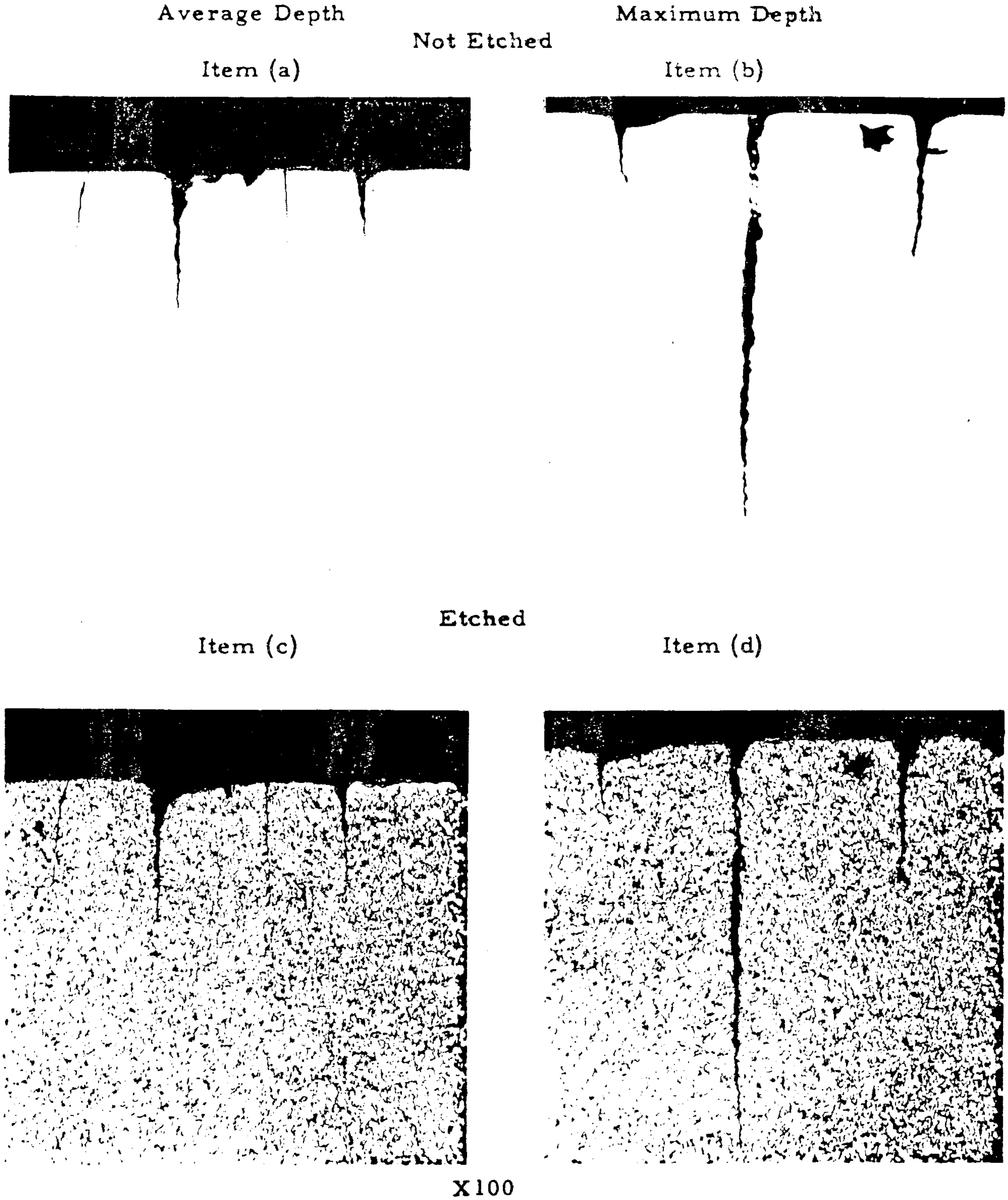


FIGURE 17 – CORROSION FATIGUE CRACKS IN ALLOY 4 SPECIMEN (NOT ROLLED) AFTER 1,035,000 CYCLES AT A STRESS OF 35,000 PSI, IN SEVERN RIVER WATER. TEST 14.

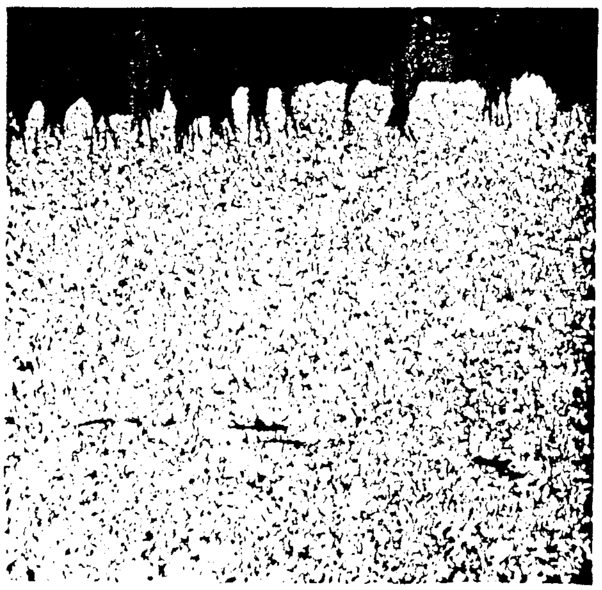
Not Etched

Item (a)

Item (b)

Etched

Item (c)



X100

FIGURE 18 – CORROSION FATIGUE CRACKS IN ALLOY 4 SPECIMEN (DEEP ROLLED) AFTER 24.6 MILLION CYCLES AT A STRESS OF 35,000 PSI, ON SEVERN RIVER WATER. TEST 16.

photomicrographs show clearly the difference in crack propagation in rolled shafts versus non-rolled shafts subject to corrosion fatigue. The ratio of the crack depths in the photomicrographs for rolled and non-rolled shafts is of the same order as that shown in figure 16 for shafts subject to fretting fatigue.

As shown by tests #15 and 16 of table 2 and by reference (12), the benefit derived from surface cold rolling is related to the depth of cold work obtained. The depth of cold work, which is the depth to which plastic deformation occurs, will take place to the maximum depth that the shearing stress induced in the shaft by the action of the "hardening" roller exceeds the yield point in shear of the shaft material. (See figure 20, in the Appendix).

The cold work(plastic deformation) of the shaft material by surface cold rolling results in residual stresses in the shaft. The core metal of the shaft is strained elastically under relatively low tension stress while the surface layer is placed under heavy compressive stress. Test data indicates that the depth of the residual compressive stress is equal to the depth of cold work. Figure 19 shows the magnitude and depth of residual stress found in tests 5 and 6 of figure 15 and table 2. (Figure 19 also shows a radial hardness gradient which indicates the depth of cold work in the shaft and is similar to that shown in figure 12 for the test shaft rolled at Newport News).

These residual compressive stresses in the surface layer of the shaft are attributed by some as the reason for the increased fatigue strength gained by surface cold rolling. (Reference (4) describes tests which support this theory). This reasoning is based on the assumption that steel has much greater fatigue strength under compressive stress than under tensile stress. The theory is advocated that numerous submicroscopic flaws or cracks exist in materials and that these cracks do not propagate in the presence of residual compressive stresses as readily as when no stress or tension is present. (A feature which tends to disprove that residual compressive stresses are responsible for increased endurance strength is the fading of internal residual stresses due to repeated cyclic stressing. Results show that repeated cyclic stressing will reduce residual stresses to only a fraction of their initial value but that no apparent loss of fatigue strength accompanies this reduction).

Other reasons proposed for the gain in endurance strength by surface cold rolling are that the physical and metallurgical properties of the surface skin may be improved or that the specific elastic shear strain energy for the surface grains may approach the higher values possibly characteristic of the grains within the body.

Quite possibly the benefit derived from surface cold rolling may be due to a combination of these reasons.

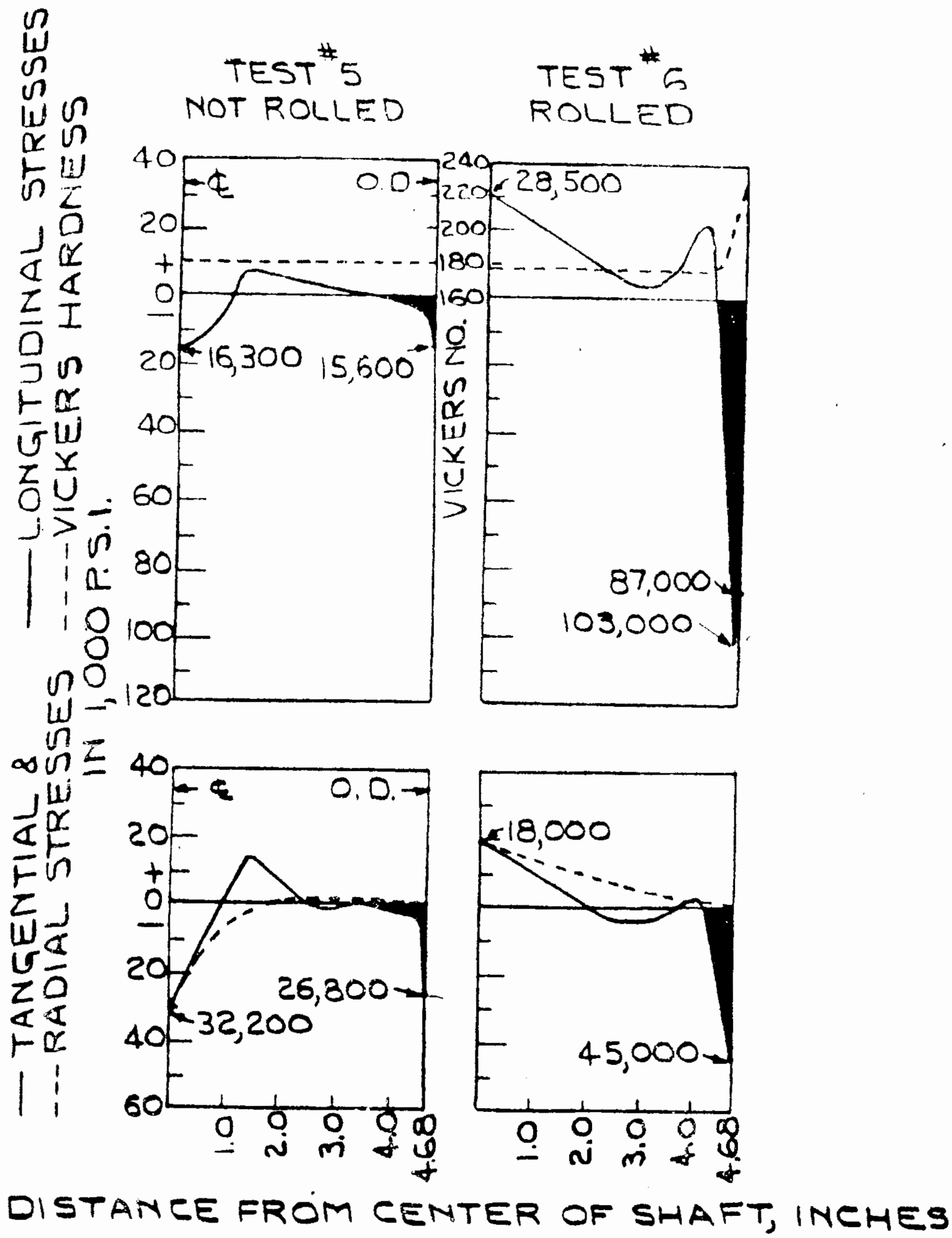


FIGURE 19 – RESIDUAL STRESS OBTAINED IN 9½" DIAMETER CRANK PINS (2) – TESTS 5 AND 6.

The following is evident from this review of test results:

(1) The endurance strength of non-rolled steel shafting subjected to fretting in air is approximately 10,000 PSI and does not vary appreciably between steels with low physical properties and those with high physical properties.

(2) Surface cold rolling steel shafting subjected to fretting in air increases the endurance strength of the shafting about two to three times depending on the depth of cold work.

(3) No endurance limit exists for steel shafts subjected to corrosion fatigue but surface cold rolling increases the number of cycles before fracture at the same stress.

(4) Fatigue cracks initiate at very low stress in both rolled and non-rolled steel shafts subject to fretting but propagate much slower in rolled shafts than in non-rolled shafts.

In applying these test results to full scale marine propeller shafts, it must be kept in mind that the benefit gained with surface cold rolled service shafting may not be of the same proportion as that obtained with test shafts. The scale effects between test shafts and marine propeller shafts have yet to be determined.

Application of Cold Rolling

From the foregoing, it is readily apparent that no question can exist as to the favorable effect of surface cold rolling on fatigue strength and that no harm can be done by cold rolling, if properly done. Questions do exist however, concerning the degree of benefit that the marine industry can derive from surface cold rolling. In discussing this question, an understanding is necessary of the failures experienced by the marine industry with propeller shafts.

Since 1951, merchant ships classed by the American Bureau of Shipping have replaced an average of about 1% of all propeller shafts inspected due to loss at sea and about 18% due to condemnation resulting from cracks in the shaft. A survey (5) of Naval vessels at Puget Sound Naval Shipyard during the period from 1944 to 1954 showed no shafts lost at sea, that about 40% of all shafts inspected had cracks and that about 10% of these cracked shafts were condemned. (The higher casualty rate of merchant shafting over that of Naval shafting is attributed to the difference in ship operation and not the shaft design)

It is theorized that the majority of propeller shaft failures start with fatigue cracks due to fretting under the forward end of the hub and aft end of the sleeve which propagate to failure of the shaft.

The presence of sea water contributes to these shaft failures for those cases where sea water leaks in the hub assembly. The conclusion of Panel M-11, of the Society of Naval Architects and Marine Engineers, based on a survey of tailshaft inspections was, "that shaft corrosion due to sea water ingress is a contributing factor, but is not a dominating factor in tailshaft condemnations" (6).

Surface cold rolling of marine propeller shafts will not eliminat cracks in the shaft due to the low stress required to initiate these cracks. However, rolling will reduce the propagation of the cracks (due to fretting or corrosion) once formed and will result in fewer shaft losses at sea and in fewer shaft condemnations due to cracks.

In addition, surface cold rolling of marine propeller shafts may result in a reduction in the percentage of shafts found with cracks. From the surveys presented above, it can be surmised that about 50% of all shafting inspected has cracks. This would seem to indicate that the design of these shafts results in a bending stress which is very close to the borderline between that which will produce fretting fatigue cracks and that which will not. Tests results, figure 16, show for a rolled shaft that the stress required to initiate cracks is greater than that for a non-rolled shaft.

(It is interesting to note that some owners are installing propeller shafts designed for about twice the strength in bending required by the regulatory bodies. This will result in lowered bending stresses and may eliminate formation of fretting fatigue cracks in the shaft. In the light of the benefit derived from surface cold rolling, it seems an excessive expense. It must be remembered that the increased shaft strength will not prevent the formation of corrosion fatigue cracks).

Since fretting fatigue cracks initiate at such low stresses, it is not recommended that any reduction in design size of propeller shafts be considered as a result of surface cold rolling. Furthermore, the benefit of surface cold rolling could be lost during some future repair requiring-stress relief and, with the possibility of no rolling equipment available and no spare propeller shaft, would necessitate the return to service of the shaft in a "non-rolled" state.

Surface cold rolling propeller shafts makes possible a longer period between inspections of the propeller shaft. Until more is known about the results from rolled service shafting, it is believed that the period between inspections should remain three years.

It is recommended that no relaxation be made in the inspection for cracks in rolled shafts as opposed to non-rolled shafts. Any cracks found in a surface cold rolled shaft should be repaired before returning the shaft to service, for the following reasons:

1. When first found, the cracks in nearly all cases would not have propagated very deep and could be removed by superficial grinding only, with no weld repair. The depth of grinding would probably be such that little loss of surface cold roll benefit would occur. A "peening" process might be developed to restore any rolling benefit lost.

2. If the crack were permitted to go until the next inspection period, it might have progressed deep enough to require welding repairs followed by stress relief and possible resulting loss of rolling benefit.

3. With the possibility of sea water corrosion always present, returning a cracked shaft to service is hazardous.

From the foregoing, it is apparent that definite benefit can be derived by the marine industry from surface cold rolling propeller shafting. Panel M-8 of the Society of Naval Architects and Marine Engineering has recommended (3) that all marine propeller shafts be surface cold rolled. This recommendation is supported by the evidence presented in this paper.

It is not known whether the optimum benefit is being derived using the present procedures. Furthermore, it is not known if laboratory tests indicate accurately the gain that can be obtained with surface cold rolled shafts in service. To properly evaluate the relation of surface cold rolling to the marine industry, the following is needed:

1. Optimum depth of cold work - All surface cold rolled marine propeller shafting, of which we have knowledge, has been cold worked to a theoretical depth of about 0.50 inch. A review of available test data (particularly (12)) reveals that a depth of cold work greater than 0.50 inch might produce greater residual compressive stresses and presumably more resistance to fatigue.

2. Rate of shaft surface speed - Reference (7) shows that a high rate of strain results in a higher yield point. It may be possible that a slower shaft speed, which would produce a lower rate of strain, might result in a slightly greater depth of cold work for the same roller load.

3. Effect on residual compressive stresses in a rolled shaft from the heat of a hot (500° - $600^\circ F$) shaft sleeve when shrunk on the shaft.

4. Rate at which fretting fatigue cracks propagate as the number of cycles increases.

5. Effect of small amounts of sea water over a long period of time on fatigue strength (this would most nearly similate the corrosion condition which can occur in actual service).

6. Procedure for restoring cold work in way of areas ground out for removal of cracks.

7. Service data of surface cold rolled shafts.

It is hoped that the marine industry will cooperate in making available all information possible concerning service data of surface cold rolled propeller shafts.

Acknowledgment

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Rear Admiral A. G. Mumma, Chief of the Bureau of Ships
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Mr. W. C. Stewart, Head, Metallurgy Department U.S. Naval Engineering Experiment Station, Annapolis, Maryland

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13. Bulletin 212, Engineering Experimental Station, University of Illinois, by Thomas and Hoersch

APPENDIX

By reference (10), the greatest principal shearing stress occurs along a line normal to the surfaces of two contacting elastic bodies and is shown in figure 20. The shear stress τ occurs at a depth z. If τ_1 is the shear yield point of the material, then plastic deformation will occur to a depth z_1, assuming that the stress system under the roller at a depth greater than z_1 is negligibly different from that which would exist were the body perfectly elastic. In calculating the depth, z, of the point where the shear stress is τ, three relations are of interest:

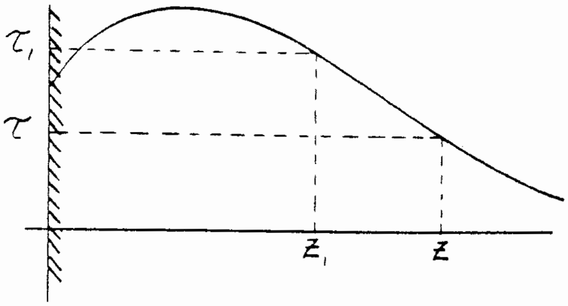

Figure 20 - Variation of principal shearing stress along a line normal to the surface of an elastic body where another body is bought into contact with it.

(A) Load applied at a point

$$\tau = \frac{P}{8\pi}\left[\frac{7-2\mu}{z^2}\right] \tag{1}$$

(B) For a circular contact area

$$\tau = \frac{aE}{2R(1-\mu^2)}\left[\frac{3}{\pi}\,\frac{a^2}{a^2+z^2} + \frac{2}{\pi}(1+\mu)\left(\frac{z}{a}\cot^{-1}\frac{z}{a} - 1\right)\right] \tag{2}$$

Where: a = radius of contact $= \sqrt[3]{\dfrac{3PR(1-\mu^2)}{4E}}$ (3)

$\frac{1}{R}$ = average of the four principal curvatures of the two bodies

$= \frac{1}{4}\left(\frac{1}{r_1} + \frac{1}{r_1'} + \frac{1}{r_2} + \frac{1}{r_2'}\right)$

E = modulus of elasticity

μ = poissons ratio

r_1 = radius of "hardening" roller

r_2 = radius of shaft

r_1' = contour radius of "hardening" roller

r_2' = contour radius of shaft

(C) For elliptical contact area of major semi-axis, c, and minor semi-axis, b

$$\tau = \frac{cE}{R(1-\mu^2)} \cdot f\left(e, \mu, \frac{z}{c}\right) \tag{4}$$

(For derivation see reference (13))

Where:

$$e = b/c$$

$$C = \sqrt[3]{\frac{3PR(1-\mu^2)(e')}{2\pi e^2}} \tag{5}$$

$$(e')^2 = 1-e^2$$

Using equations 4 and 5 to plot τ vs z with P, R, μ and E equal to a constant shows that if e is greater than 0.50 the contact area may be taken as a circle for computing the load.

The condition for e to be greater than 0.50 is that

$$\frac{\frac{1}{r_1'} + \frac{1}{r_2'}}{\frac{1}{r_1} + \frac{1}{r_2}} < 2.85$$

If r_1' = 1.5 inches and r_1 = 4.5 inches the limiting value of r_2 for this condition to be fulfilled is 85 inches.

Equations 1,2 and 3 can be used to determine the roller load for the desired depth of cold rolling, since the shaft sizes of interest are well within the limiting value. Using equations 2 and 3 with e = 1, P = 1000, μ = .3, and E = 30 x 10^6, figure 21 can be plotted for R greater than zero. Equation 1 is used to plot R = 0.

If P = 1000M and is the load which will cause a stress τ_1 at a depth z_1, the stress $\tau_1 = \tau\sqrt[3]{M}$ will exist at depth $z_1 = z\sqrt[3]{M}$. τ is the stress at depth z with a 1000 pound load. To use Figure 21 plot the point τ_1, z_1 where z_1 is the desired depth of cold working and τ_1 is the shear yield point of the shaft material. Connect this point

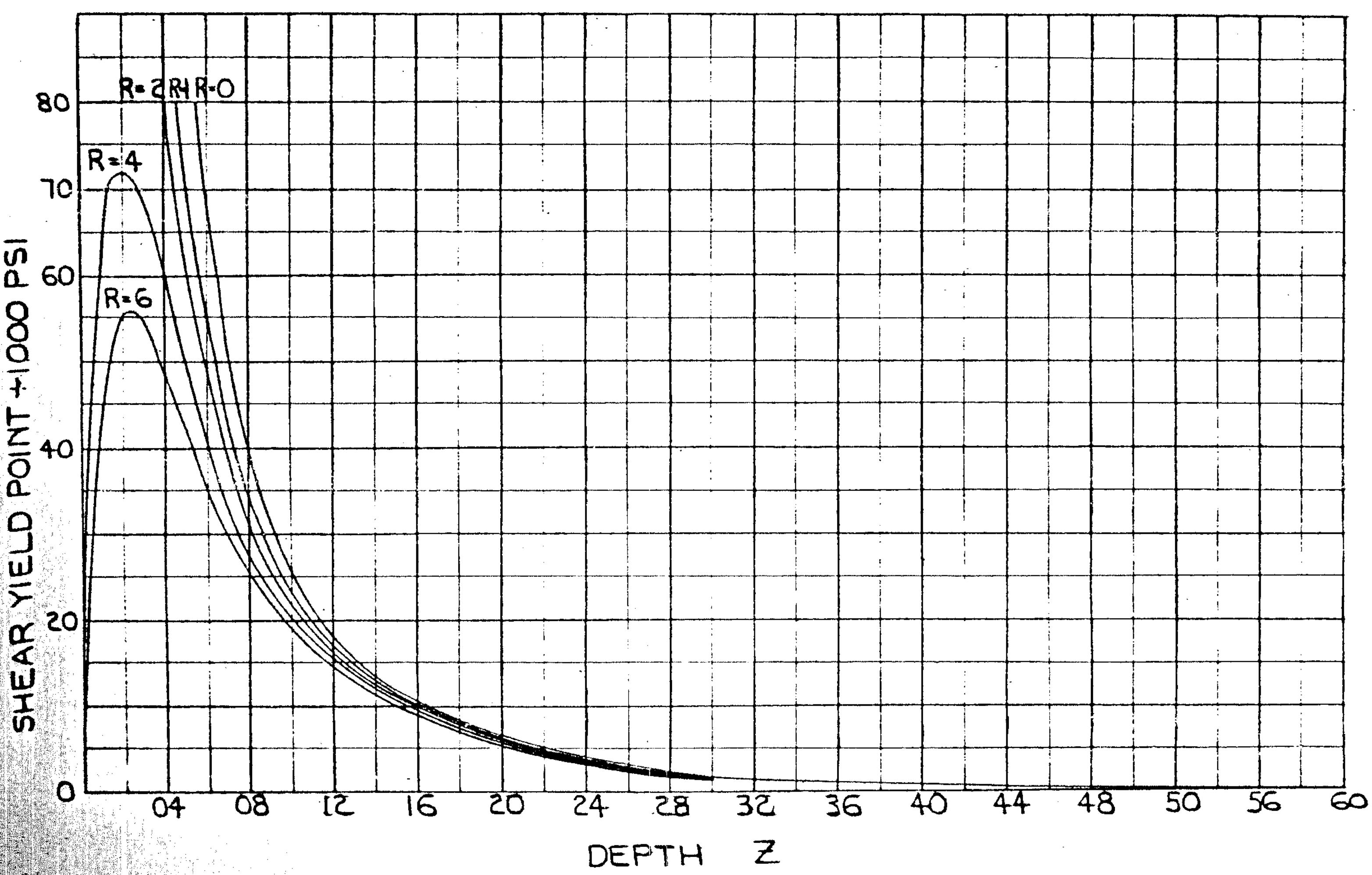

FIGURE 21 – SHEER STRESS AT VARIOUS DEPTH WHEN P (ROLLER LOAD) = 1000 LB. AND $e = 1$.

to the origin with a straight line and where this line crosses the curve of the correct R is the point τ, z. Then $M = \left(\frac{z_1}{z}\right)^3 = \left(\frac{\tau_1}{\tau}\right)^3$ is calculated.

Using Figure 21 as outlined above, Figure 22 can be constructed from which the roller load can be immediately selected for any shaft material. Figure 22 may be used for roller diameters from 8 to 10 inches with a 1-1/2 inch crown radius without introducing any significant error. In constructing Figure 22, the yield point in shear was taken as 0.50 times the yield point in tension.

The minor semi-axis of the contact area for various size shafts is shown in Figure 23 and is used to determine the maximum rate of feed for surface cold rolling. It is evident from Figure 23 that the maximum rate of roller feed along the shaft can not be determined by assuming the contact area is circular. Radius "a" of Figure 23 was computed making this assumption, using equation 3. The value b, is the maximum permissible rate of feed and is the minor semi-axis of the elliptical contact. Reference (11) was used to compute the semi-axis of the elliptical contact shown on Figure 23.

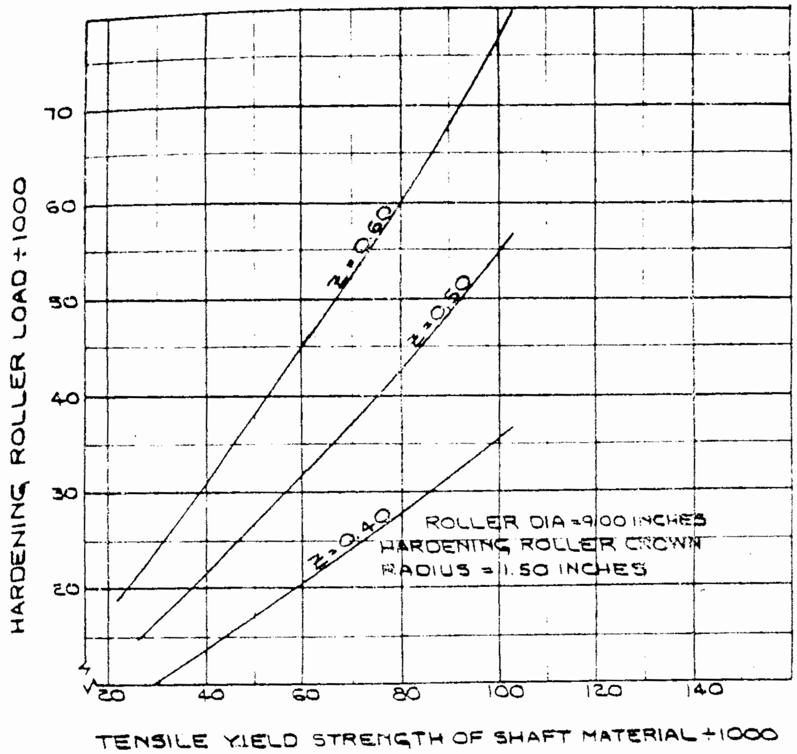

FIGURE 22 – ROLLER LOADS FOR VARIOUS SHAFT MATERIALS USING 9" DIAMETER "HARDENING" ROLLER WITH 1½" CONTOUR RADIUS.

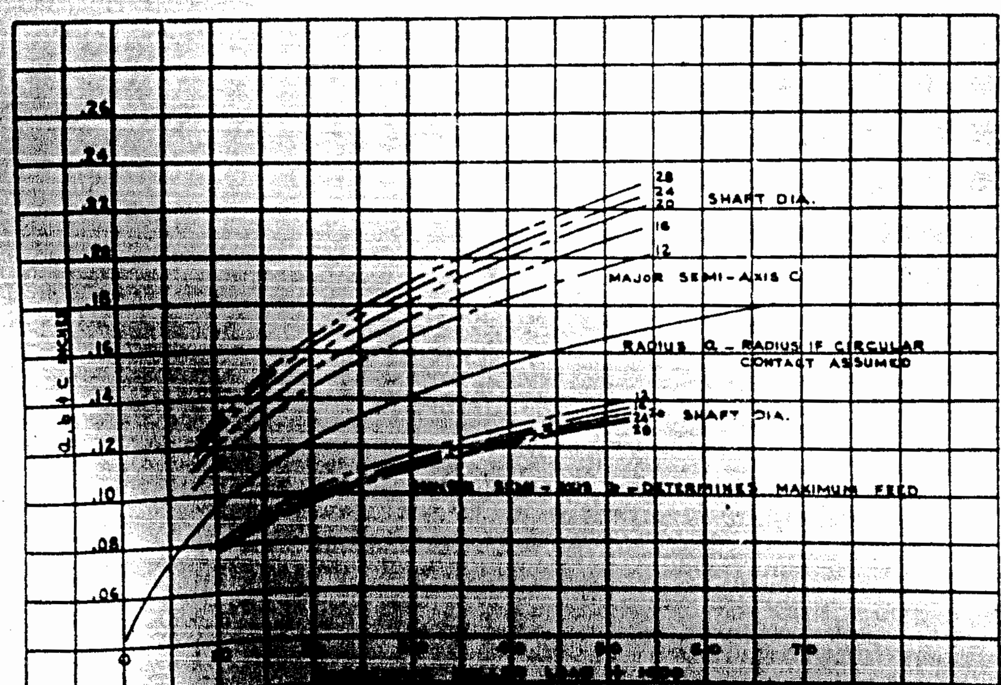

FIGURE 23 – MAJOR AND MINOR AXES OF CONTACT AREA ELIPSE FOR VARIOUS ROLLER LOADS USING A 9" DIAMETER "HARDENING" ROLLER WITH A 1½" CONTOUR RADIUS.