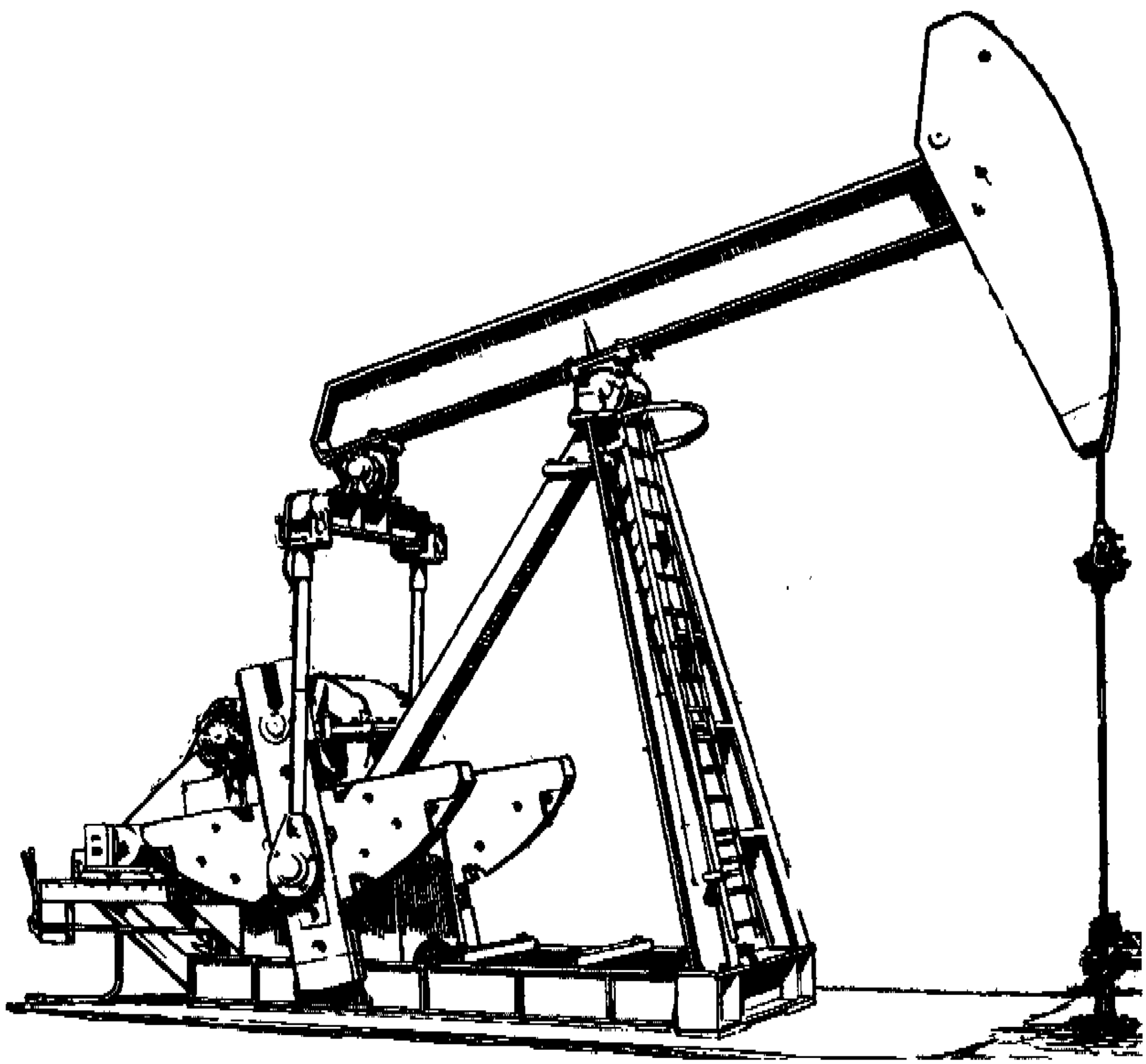
LS Petrochem Equipment Corp.

LS Pumping Unit
Engineering Data Book



F010600

US Tel: (909) 595-3138 Fax: (707) 281-0151 398 S. Lemon Creek Dr. Suite M, Walnut, CA 91789 www.pumpjack.com

TABLE OF CONTENTS

Section 1

Gear Reducer Design

GEAR REDUCER DESIGN

The Darco USA gear reducer is the only pumping unit gear box that presently utilizes a cast steel and forged steel combination of gearing. This according to the foremost gear experts of the world is superior to the combination of ductile cast iron material against steel pinion gears, commonly used by competing pumping unit manufacturers.

In addition the Darco gear train is designed with dual high speed gears and pinions insuring a greater and more equalized loading condition upon operation. The industry's largest shafting and bearings contribute to higher safety factors, while the split gear case design allows operators easier service flexibility.

In an effort to provide an accurate non-prejudiced evaluation of our gear reducer, we have enclosed two papers which were prepared by Mr. Dennis P. Townsend, Manager of Gear Research at the NASA Research Center in Cleveland, Ohio. The first paper titled "Qualifications of the Darco Gear Reducer", was authored by Mr. Townsend; while the second paper includes test results comparing Double Circular Arc gears with that of the more common Involute style gear teeth, this test was performed in China and reviewed for authenticity by Mr. Townsend.

QUALIFICATION OF THE DARCO DOUBLE CIRCULAR ARC GEAR REDUCER PUMPING UNIT

by
Dennis P. Townsend

The circular arc gear was first invented by E. Wildhaber in the United States in the 1920's. It was later invented in Russia in the 1950's by Novikov and is now generally referred to as the Wildhaber-Novikov gear. Each inventor had some different features that make his unique. The Russians began developing the circular arc gear and were soon making production gear reducers using the new tooth design. In the 1960's, the English continued the development of the circular arc gear and began producing a helicopter gear reducer, utilizing the circular arc gear. This unit has been in production for several years and has performed as designed without problems, moreover the designer states that this single circular gear has 1.5 times the torque capacity of an involute gear pair.

Several researchers around the world have conducted research on the single and double circular arc gears. Beginning in the 1970's, the Indians conducted extensive research on circular arc gears and have shown the bending strength and Hertzian compressive strength to be two times that of the involute gears. At the University of Brussels in Belgium, researchers have demonstrated the superiority of the double circular arc gear over the single circular arc gear and determined that the smaller helix angles gave higher bending strength for double circular arc gears. The Japanese have also conducted testing on double circular arc and a similar sim-arc gear unit with very good results. Extensive research and development has been done on circular arc gears in Russia and China, both countries have shown that the double circular arc gear reducer has much higher capacity than the involute reducer. In China recent tests were conducted with equivalent gear reducers of involute and double circular arc gears. In this test the double circular arc reducer was tested for nearly 300 hours at loads a little below two times, and at two times the load capacity of an involute reducer without failure.

The data from researchers in several countries indicates there is no doubt that the double circular arc gear reducer can be made with considerably more torque capacity than an equivalent involute reducer.

THE DARCO/LS INTERNATIONAL DOUBLE CIRCULAR ARC REDUCER DESIGN

The double circular arc reducer design is the result of several years of research and development on circular arc gears. This research and development has led to a better understanding of the design methods for double circular arc gearing. Due to the nature of the loading zone on the teeth, designers have learned how to modify the gear teeth to reduce edge loading, noise and vibration. These methods have been used on involute gears but had to be modified for circular arc gears once the requirements were known. Figure 1 shows a comparison of the contact motion for involute and circular arc gears and illustrates the need for a different type tooth modification for each gear system. Early tests by some researchers did not recognize this requirement and had poor results for the circular arc gearing. Figure 2 shows how bending strength is reduced at lower helix angles and also shows the need for length-wise modification of the tooth to reduce edge loading as the teeth come into contact.

Experimental stress analysis by researchers have shown that the double circular arc bending stress and contact stress is considerably lower than that for involute gears. Figure 4 and Table 1 show the results of photelastic stress evaluation of involute and double circular arc gears. In this comparison the double circular arc gear teeth are loaded at one point only and still show a 30% lower bending stress than the involute tooth. It is concluded from these results that the bending stress would be 50% to 100% improved utilizing the two point contact.

Tests conducted in China on double circular arc and involute gear reducers have shown that the double circular arc gearing can successfully transmit more than two times the torque of an involute reducer with medium hard gears.

The material used in the double circular arc gearing is a steel that has considerably more strength that the ductile iron used in most United States pumping units.

The bearings utilized in the Darco/LS double circular arc unit are the 2300 type which have 50% to 100% more loading capacity than the 5200 series used in other oil well pumping reducers. Since the bearing loads are similar for both type of gear reducers, the 2300 type bearings will have lives that are 4 to 10 times the life of other competing reducers.

Based on the double circular arc gear design, the improved material, and the increased bearing capability, the capacity of the double circular arc gear reducer is at least 50% superior than the competition.

EXPERIENCE OF DOUBLE CIRCULAR ARC GEAR REDUCERS

The circular arc and double circular arc gear drives have gained several years of experience which has shown that they are a durable drive system. Westland Helicopter Company in England has been flying circular arc gears on a helicopter since the late 1960's with very good results and have shown that they can transmit 50% more torque than an involute gear. The English have also developed industrial gear reducers using circular arc gears, while the Russians have been using circular arc reducers since the 1950's with very good results.

The Chinese have utilized circular arc gears for many years and have developed many industrial reducers including oil well pumping units. They have at least one turbine driven high speed double circular arc gear drive operating at 120 m/sec with good results. In addition, the Chinese have tested double circular arc units against involute reducers and have shown a capacity greater than 2 times the involute gear drive. Many double circular arc gear oil well pumping units have been operating in the United States for the past 10 years without gear failures and with proven reliability.

CONCLUSION

* **The capacity of the double circular arc gear reducer has been proven by tests in least four countries to have from 50% to 100% more torque capacity than involute gear reducers.**

* **Ten years of experience in the United States with double circular arc gear driven pumping units have shown this design to be a very reliable and cost effective alternative to the involute type pumping unit.**

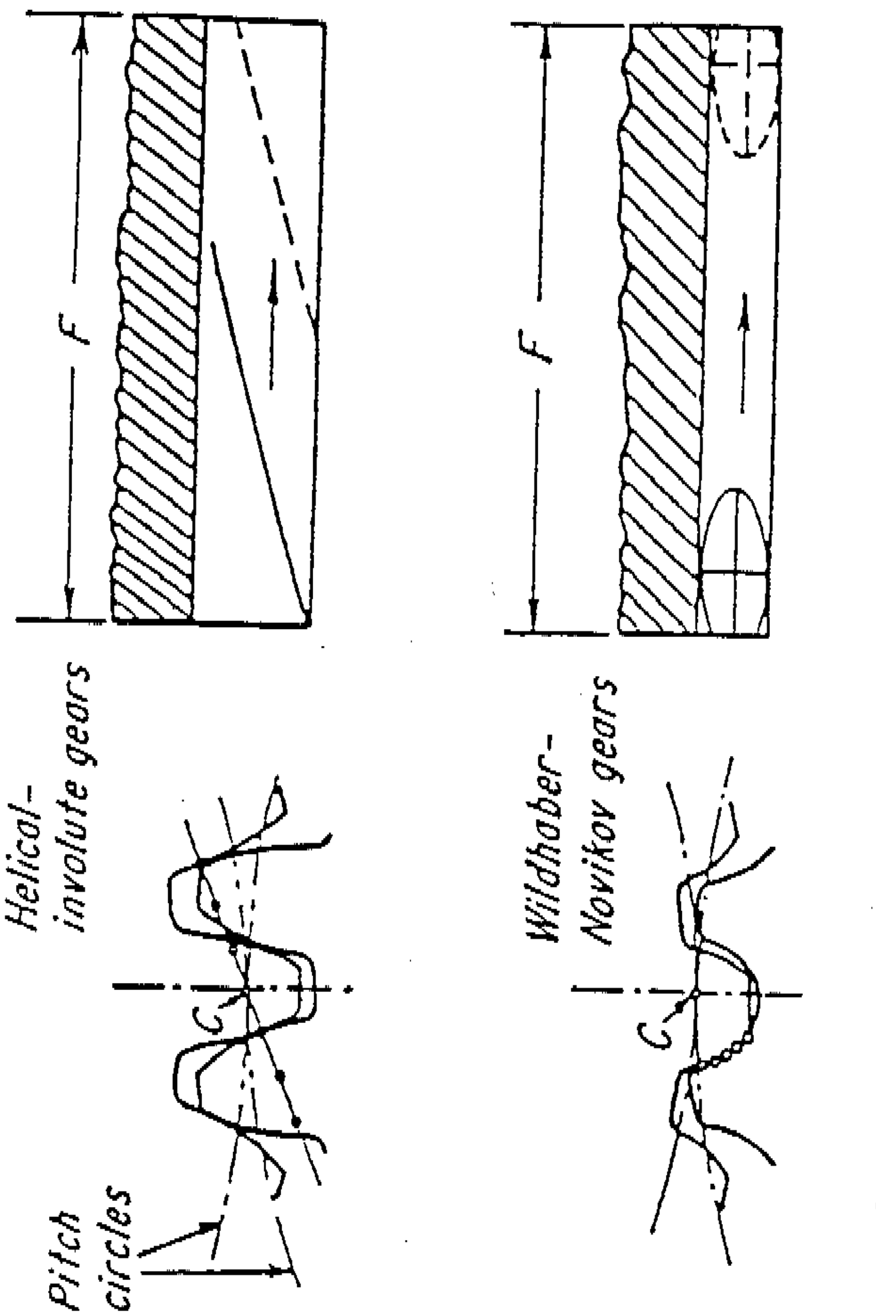
RESUME
MR. DENNIS P. TOWNSEND

Mr. Dennis Townsend is the Manager of Gear Research on the staff of the NASA Lewis Research Center Mechanical Systems Technology Division. He received his Bachelor of Science in Mechanical Engineering in 1952 from the University of West Virginia. Upon graduation, he worked with the Defense Department on the design of electro-mechanical computer systems and General Electric Large Jet Engine Department on the design and development of system components for jet engine fuel lubrication and hydraulic systems. Mr. Townsend joined the NASA Lewis Research Center in 1962 and conducted design analysis and evaluation of nuclear rocket engine components for the nuclear rocket engine program.

Mr. Townsend later joined the Bearing Research Section at NASA LERC where he conducted research and analysis on bearing lubrication. In 1967, he founded the NASA LERC gear and transmission research program and conducted extensive research on gearing and transmissions. His contributions to the gear industry are significant and include advances in gear materials and processes for improved operating temperature and gear life, understanding and analysis of gear lubrication, gear thermal analysis and gear dynamic analysis.

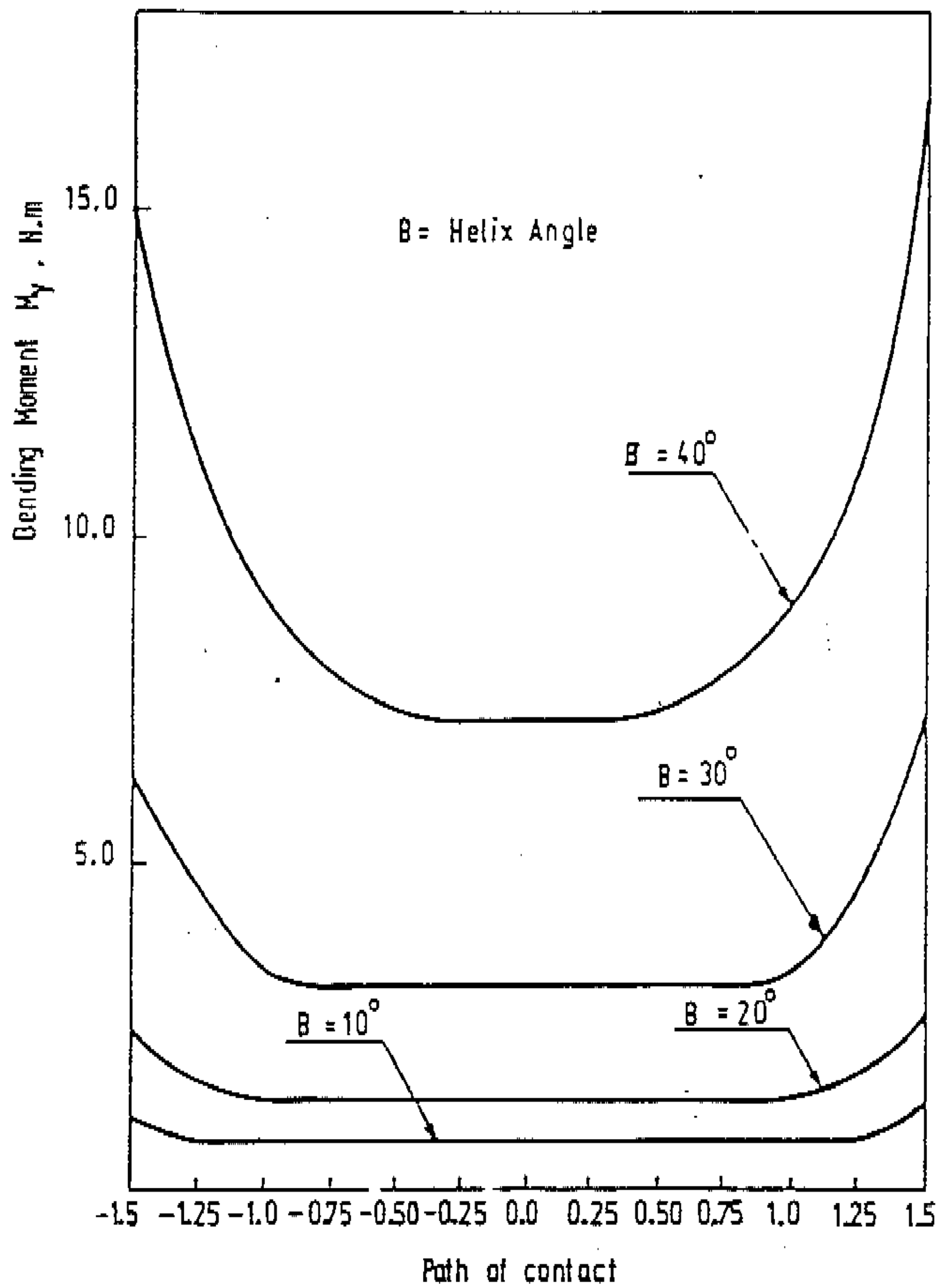
He has authored or co-authored over seventy-five papers in the gear and bearing research area and currently serves as the resident NASA gear consultant for NASA, various United States Military groups, and numerous industrial companies.

Dennis Townsend was Chairman of the Power Transmission and Gearing Committee from 1978-1983, Associated Editor of the Journal of Mechanisms, Transmissions and Automation Design from 1978-1983, awarded the ASME Fellow Award in 1987, and serves as Chairman of the ASME Design Engineering Division for 1989-1991. He currently is re-writing the Gear Handbook of the United States.



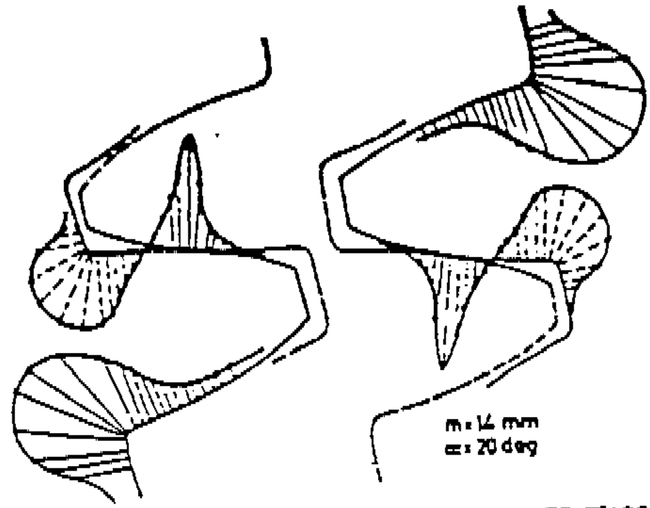
• COMPARISON of helical gear systems.

FIGURE 1.

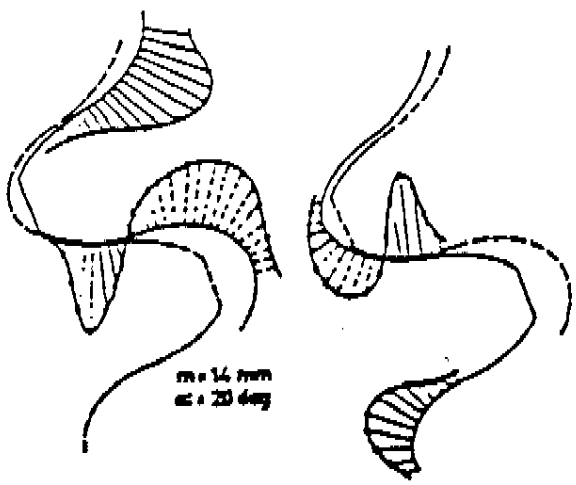


Change of maximum value of bending moment M_y, at built in edge with change of helix angle along the path of contact

FIGURE 2.

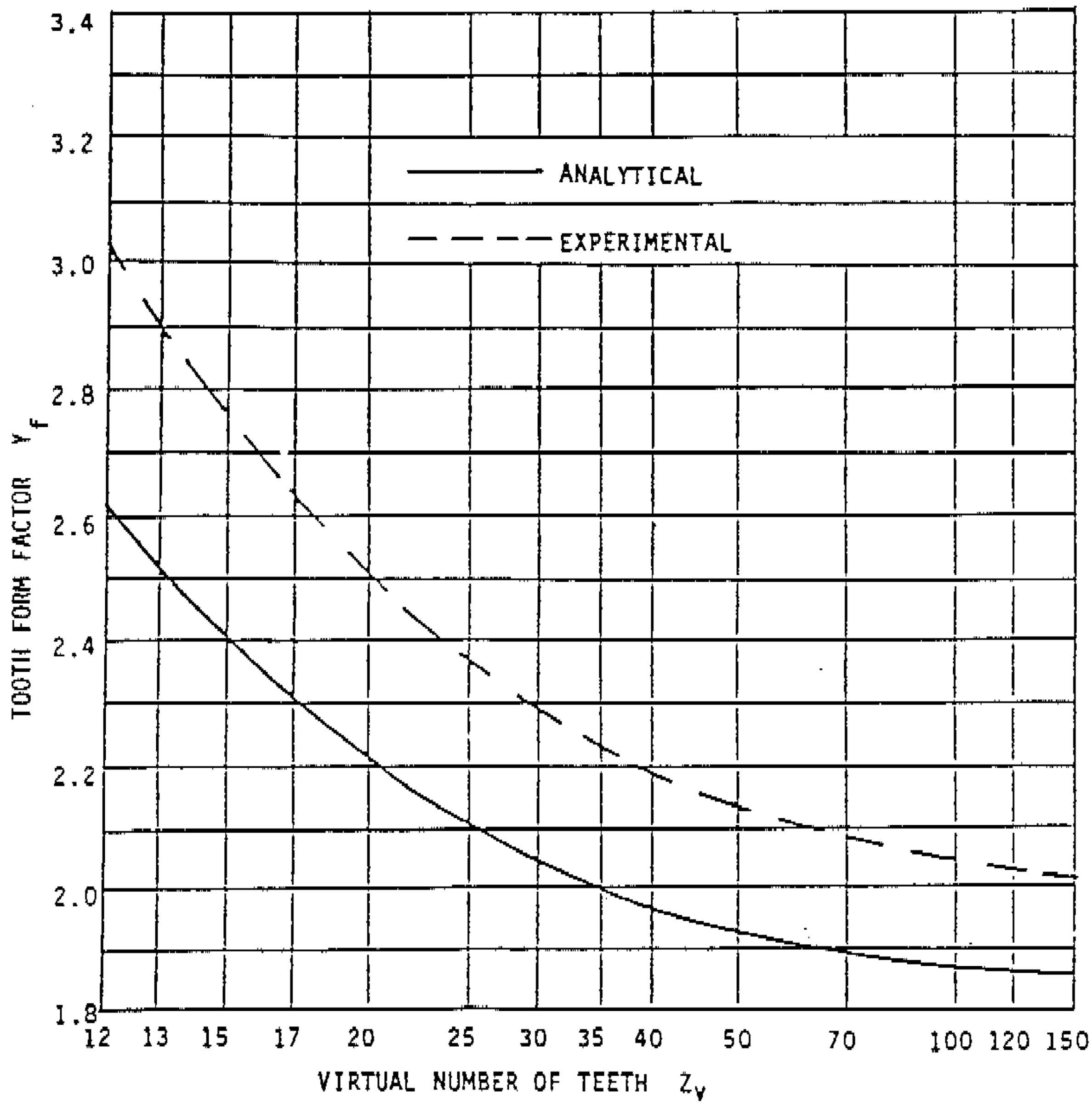


SURFACE STRESS DISTRIBUTION IN EXTERNAL INVOLUTE GEARS



SURFACE STRESS DISTRIBUTION IN WILDHABER-NOVIKOV GEARS
(ADDENDUM-DEDENDUM TYPE)

FIGURE 3.



Tooth Form Factor Y_f for Double-Circular-Arc Gear.

FIGURE 4.

COMPARISON OF BENDING STRESS FOR SINGLE POINT CONTACT

Sl No	Profile	Stress, MPa	
		Tension fillet	
		Pinion	Wheel
1.	Involute, External	571	464
2.	Circular-Arc (Addendum-Dedendum)	390	321

TABLE I

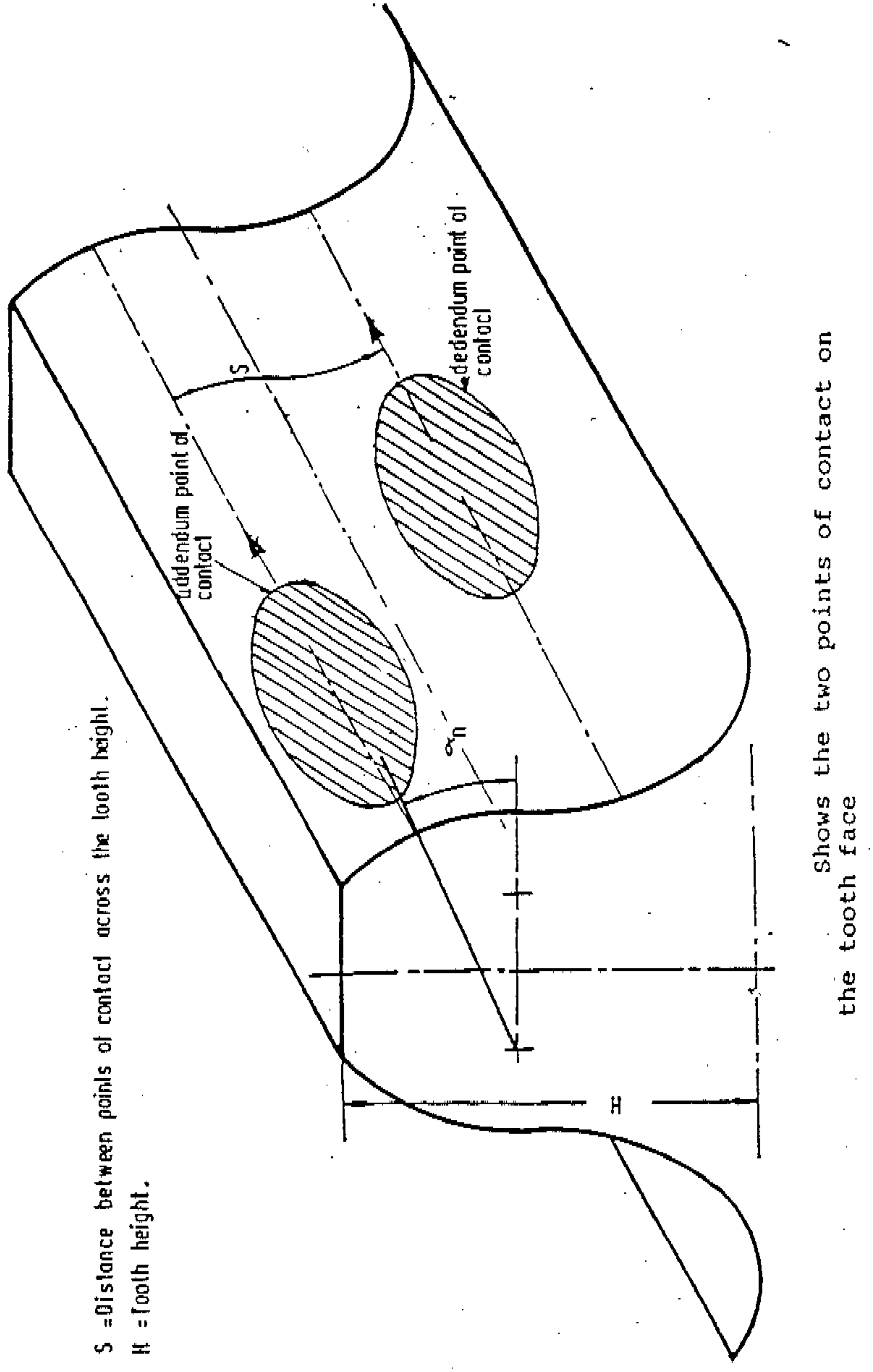

S
addendum point of contact
dedendum point of contact
α_n
H
Shows the two points of contact on the tooth face
S =Distance between points of contact across the tooth height.
H =Tooth height.

FIGURE 5.

ADVANTAGES OF THE DARCO/LS DOUBLE CIRCULAR ARC GEAR REDUCER

* HIGHER TORQUE CAPACITY-50% TO 100% OVER INVOLUTE

* LOW CONTACT STRESSES-CONCAVE/CONVEX CONTACT SURFACES OF THE GEAR PROVIDE LOWER HERTZIAN CONTACT STRESS FOR IMPROVED TORQUE CAPACITY AND LONGER LIFE

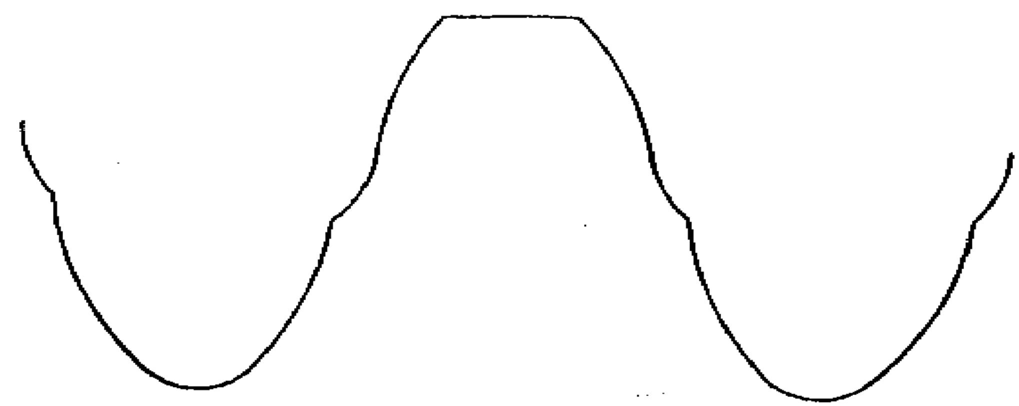
* BENDING STRESS-THE DUAL CONTACT, WIDE TOOTH FORM AND LARGE ROOT RADIUS OF THE DOUBLE CIRCULAR ARC GEAR, PROVIDES A HIGH BENDING TORQUE CAPACITY

* STEEL GEARS HAVE HIGHER STRESS LIMITS THAN DUCTILE IRON GEARS USED BY OTHER OIL WELL PUMPING UNITS

* EXCELLENT EHD OIL FILM IN THE CIRCULAR ARC GEARS PROVIDES BETTER LUBRICATION, LOWER FRICTION, AND IMPROVED EFFICIENCY. TESTS BY AEI IN ENGLAND SHOWED EHD FILM THICKNESS TO BE 3 TO 4 TIMES THAT FOR AN INVOLUTE GEAR

* PROVEN HIGHER CAPACITY OF DOUBLE CIRCULAR ARC GEARS BY RESEARCHERS FROM SEVERAL COUNTRIES SHOWED TESTING RESULTS OF 50% TO 100% GREATER TORQUE CAPACITY

* HIGH CAPACITY ROLLING ELEMENT BEARINGS PROVIDE 4 TO 10 TIMES THE LIFE OF OTHER OIL WELL PUMPING UNIT REDUCERS



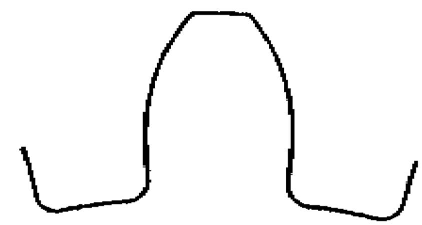
STEEL DOUBLE CIRCULAR ARC GEARS

- STRONG, STABLE, WIDE BASE TOOTH PROFILE
- TWO POINT TOOTH CONTACT
- PITTING, SCUFF, AND FATIGUE RESISTANT

DOUBLE CIRCULAR ARC

* WIDE BASE TOOTH
  STRONGER - MORE STABLE
  LOWER TOOTH ROOT STRESS

* TWO POINT CONTACT
  LESS WEAR
  LOW CONTACT PRESSURE
  GOOD LUBE EHD FILM



INVOLUTE

* NARROWER BASE TOOTH
  WEAKER - MORE DEFL.
  HIGHER TOOTH ROOT STRESS

* SINGLE POINT CONTACT
  MORE WEAR
  HIGH CONTACT PRESSURE
  LESS LUBE EHD FILM

EXPERIENCE OF DOUBLE CIRCULAR ARC GEARING

* 1958 RUSSIAN PRODUCTION UNITS OPERATING IN SEVERAL INDUSTRIES

* 1968 CHINESE PRODUCTION UNITS OPERATING IN MANY INDUSTRIAL APPLICATIONS

* 1962 ENGLISH PRODUCE CIR-ARC REDUCER

* 1967 CHINA STARTS PRODUCTION OF DCA GEAR REDUCERS

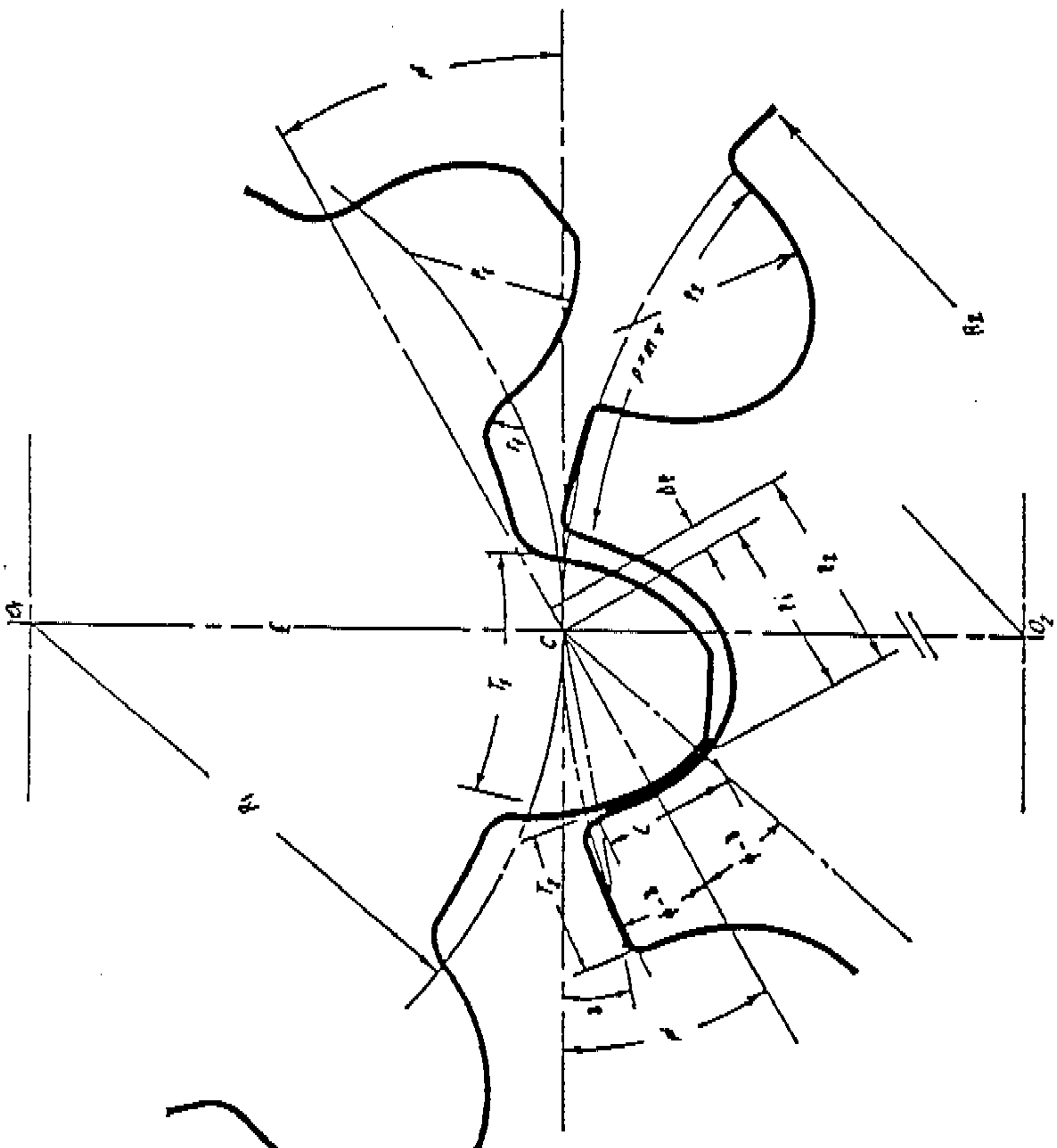
* 1968 WESTLAND HELICOPTER PRODUCES CIRCULAR ARC REDUCER

* 1976 CHINA PRODUCES HIGH SPEED DCA REDUCERS

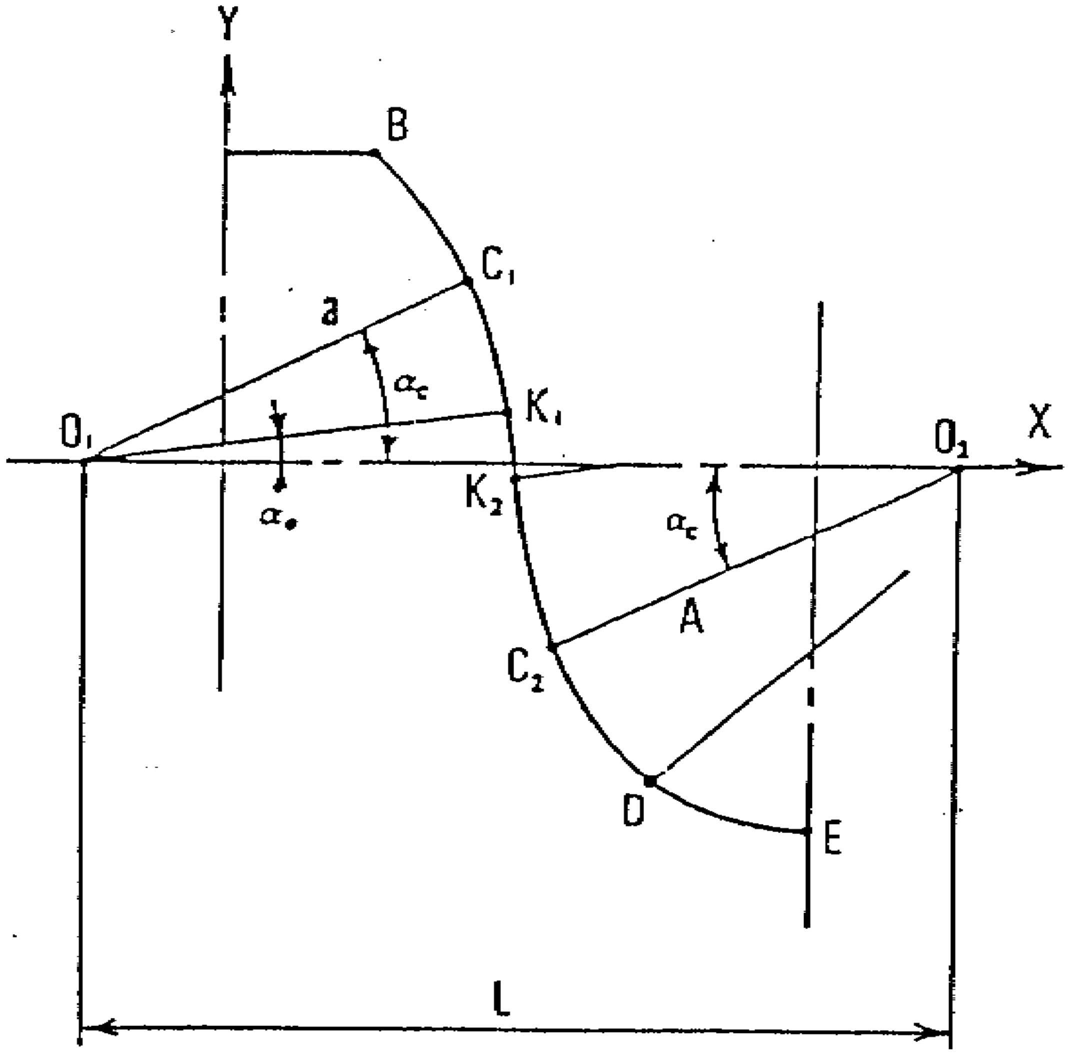
* 1980 CHINA SHIPS DCA GEAR REDUCERS TO UNITED STATES FOR OIL WELL PUMPING UNITS

HISTORY OF DOUBLE CIRCULAR ARC GEAR DEVELOPMENT

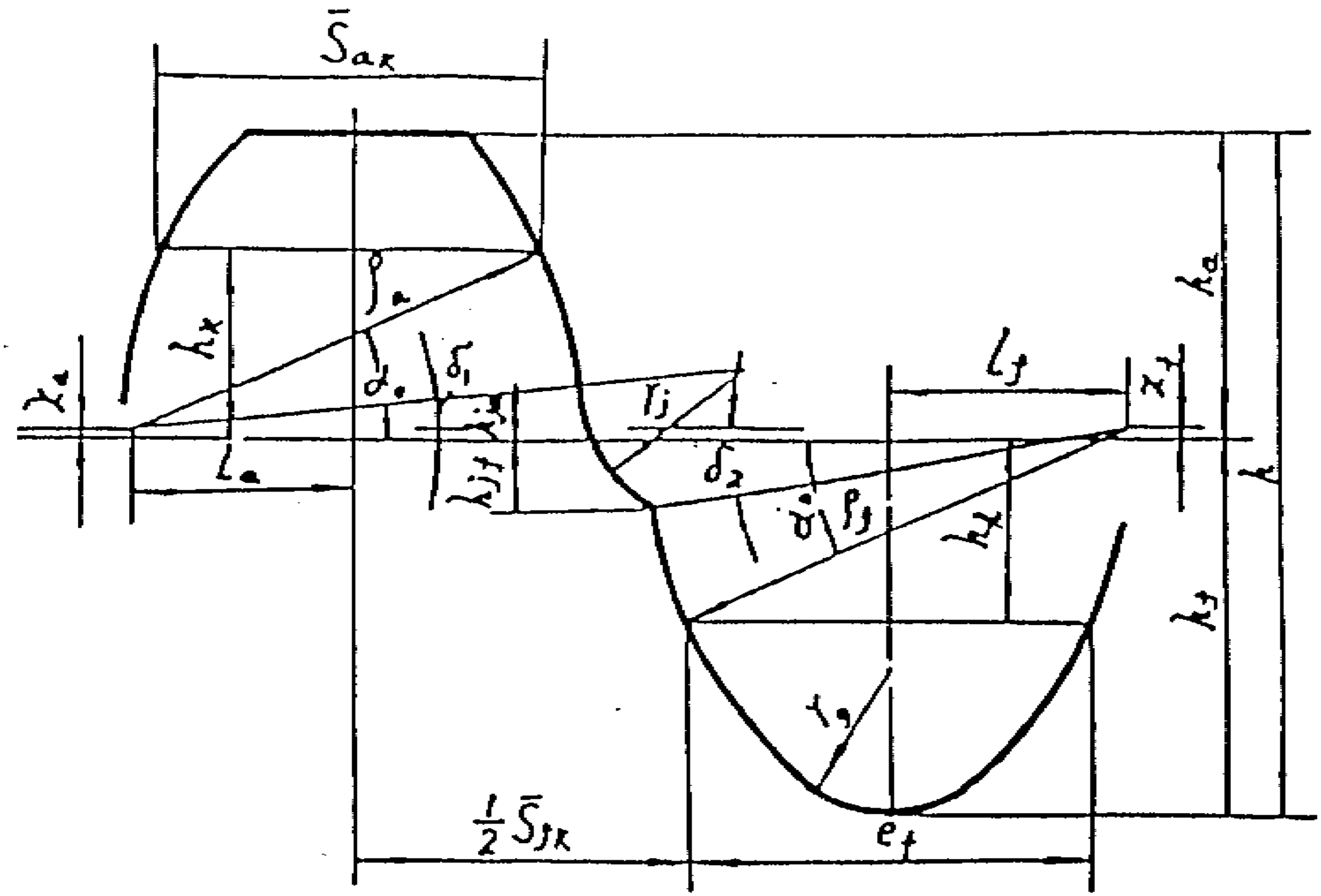
* 1926 WILDHABER U.S. PATENT
* 1956 NOVIKOV RUSSIAN PATENT
* 1958 RUSSIA ISSUED CIR-ARC TOOTH SYSTEM PRODUCTION OF CIR-ARC GEAR DRIVES
* 1958 CHINA PRODUCES CIR-ARC GEAR DRIVES
* 1960 CHINA DEVELOPS DCA DESIGN
* 1960 GERMANY STARTS DEVELOPMENT OF CIR-ARC GEARS
* 1962 ENGLISH PRODUCE CIR-ARC REDUCER
* 1967 CHINA STANDARD FOR CIR-ARC TOOTH FORM
* 1967 CHINA PRODUCES DCA GEAR REDUCERS
* 1968 ENGLAND PRODUCES HELICOPTER CIR-ARC GEAR REDUCERS
* 1970 INDIA BEGINS R&D OF CIR-ARC GEARS
* 1976 CHINA PRODUCES FIRST HIGH SPEED DCA REDUCER
* 1980 CHINA DEVELOPS DCA DESIGN ANALYSIS
* 1981 CHINA ISSUES STANDARD FOR DCA TOOTH FORM



Wildhaber Novikov (Conformal) Gear



Tooth Profile of a Simarc Gear
Combined Involute and Circular–Arc



Basic Rack Tooth Profile of Double–
Circular–Arc Gear Cutting Hob

COMPARISON TEST OF DOUBLE CIRCULAR ARC AND INVOLUTE GEARS

INTRODUCTION

The main component of an oil well pumping unit is the gear reducer. During the developmental stages of the soft tooth flank involute gear reducer in the 1950's, the service life was deemed to be low primarily from tip wear of the gear teeth. In the 1960's, the single circular arc gear was developed as a replacement for the involute gear. This solved the tip wear problem, but contributed to low bending strength resulting in large, heavy gear boxes. In the early 1980's, after ongoing development, the double circular arc gear was used as a replacement for the single circular arc and involute gear. The double circular arc gearboxes showed good results and were then manufactured domestically and abroad in increased quantities.

At that time China's oil industry was producing involute gearboxes with medium hard and hard tooth flanks to reduce the tooth wear and improve the load capacity. Presently the double circular arc reducers have replaced the single circular arc and involute reducers with soft gear teeth and made these earlier designs obsolete.

There is contrasting test data for single and double circular arc gears both in China and abroad. Therefore it was decided to conduct a comparison test between a double circular arc gear reducer with soft tooth flanks and an involute gear reducer with medium hard tooth flanks. The purpose of the test was to compare the load capacity, type of failure, noise, and vibration of the two reducers with the same material, size, and test condition.

PURPOSE AND SIGNIFICANCE OF TESTS

Purpose

a. Verify the load capacity for double circular arc and involute gear reducers.

b. Determine reason of failure of the two types of gear reducers.

c. Determine the vibration, noise, and temperature rise at each load condition.

d. Compare the analytical load capacity with the test results of the double circular arc reducer and use the results to design improved lower cost pumping unit reducers.

Significance

a. Provide actual load capacity comparison between the two types of gear reducers as well as to provide data for conducting destructive tests on reducers.

b. Verify the actual load limit capacity and accuracy of the analytical formulas used for designing future reducers.

c. Institute additional steps to improve the technology of pumping unit gear reducers.

d. Improve technology to allow reduced center distance designs that meet the same technical specifications while saving large amounts of material.

e. Examine the actual load allowance for a double circular arc reducer under existing design conditions for future consideration of modification of the design parameters and machining technology. i.e., lower gear hardness can give higher gear cutting speeds and improved machining efficiency.

f. Provide a elevated understanding of the double circular arc gear and its range of potential applications.

TEST PROCEDURE AND GEAR REDUCER DATA

Test Procedure

There are two basic types of test setups that could be used, the open loop and closed loop or back to back tests. The open loop requires more power and is less economical to run for a long duration test. The closed loop test method circulates the power through the loop of the two gear reducers, with the output and the input shafts of each reducer connected to each other (see Figure 1). A method of applying torque is placed in one of the connecting shafts, usually the high speed or lower torque shaft. The complete set up of the two reducers is driven by an external drive. This method is responsible for supplying only the losses of the system, usually about one tenth or less of the circulating power of the closed loop.

The test setup was composed of one double circular arc reducer and one involute reducer connected at the output shafts by a rigid shaft and couplings. The input shafts were connected with shaft, couplings, torque loader, and torque meter. A 13.4 HP motor and belt drive was used to supply power to the closed loop test system.

Each torque load was set by the torque unit in a static condition. After one hour of operation the torque level had decreased slightly, remaining fairly constant and was recorded as the dynamic torque.

Instrumentation consisted of torque meter, thermocouple for temperature, vibration and noise measurement equipment and strain gauges for torque. The test set up is shown in Figure 1 and the reducer parameters are given in Table 1.

Standard Used To Determine Failures.

1. A gear is considered failed when it has pitting or wear over 80% of the contact surfaces.

2. A gear is considered failed if cracks are found at the gear tooth root using a magnifier.

Procedure For Double Circular Arc Gear Loading Tests

Prior to the start of the load tests, the gears are run for 10 hours at no load and 20 hours at 480 in-lb. From this test the dual contact lines can be checked for correct contact at the pitch line. If the contact is proper the test is continued.

The gear material is a steel with a hardness of HB = 300

Its basic cycle time is:

$$N = \frac{1.5 \times (HB - 30) \times 10^6}{50 - .04HB}$$

$$N = \frac{1.5 \times (300 - 30) \times 10^6}{50 - .04 \times 300} = 1 \times 10^7 \text{cycles}$$

Therefore the time needed for one basic cycle time is:

$$T = \frac{N}{60n} = \frac{1 \times 107}{60 \times 335} = 498 \text{ hours}$$

n = pinion RPM = 335

Tests were performed according to the test program up to the safe loading parameters determined from calculations and investigations of production units (see Table 2). Since the double circular arc reducer had a very high loading capacity, the first two loading steps were run for 20 and 25 hours respectively. After 70 hours at 2070 in-lb load, a few small surface pits were observed. However, these small surface pits disappeared after 193 hours at the 3341 in-lb load condition. This wear in phenomenon indicated that the surface pits were caused by some defects in heat treatment. After the final test of 130 hours at 4113 in-lb which was two times the design load and the limit load for the connecting shaft, the gears showed no signs of surface distress, pitting or cracks in the tooth root.

Eight points were considered for noise measurement. However, only four were chosen because of the interference noise from the drive motor (see Figure 1). The distance from the top surface of the gearbox to the sound pickup is one meter. The frequency spectrum of the main noise value at the highest load was 124 hz. This frequency is caused by the gear mesh frequency. The integral average noise of the double circular arc reducer was 68.1 dB(A) and the maximum single measured noise level was 73 dB(A) (see Table 3).

Reducer vibration levels were measured in three directions, the x, y and z planes for both reducers. The vibration levels in the x direction were heavier than the y and z directions.

The vibration levels of the double circular arc reducer were slightly higher than the involute reducer but the values were generally very low with a maximum value of 2.1 microns. (see Table 4).

Procedure For Involute Gear Loading Tests

The gear material is a medium hard tooth with a hardness of HB = 345.

Its basic cycle time is:

$$N = \frac{1.5 \times (HB\ 30) \times 10^6}{50 - .04HB}$$

$$N = \frac{1.5 \times (345 - 30) \times 10^6}{50 - .04 \times 345} = 1.3 \times 10^7$$

The time required for one basic cycle time is:

$$T = \frac{N}{60n} = \frac{1.3 \times 10^7}{60 \times 335} = 646 \text{ hours}$$

The involute gears used in this test were medium hard and they meet both AGMA and Chinese standards. They have a low safety coefficient and a history of surface distress or micropitting.

The loading tests were performed for the designed torque (see Table 5). The first two loading steps were completed without problems. On the third load test after 200 hours, some surface distress or micropitting was observed. After 200 hours at the fourth load test of 2445 in-lb, heavy surface distress was observed on 80% of the gear teeth covering an area of 30% to 40% of the tooth surface. According to the failure standard, this gear reducer is considered to be failed.

The method used to measure the noise level was the same for both gear reducers and the distance of the microphones from the gear case was the same in both tests. The average value of the noise for the involute reducer was 67 dB(A) with a maximum value of 68 DB(A), which was slightly less than the double circular arc gear reducer (see Table 6).

CONCLUSIONS

Loading Capacity

The double circular arc reducer completed six step loading tests for a total time of 599 hours. The fourth step was 45% overload for 193 hours. The fifth step was 82% overload for 161 hours. The sixth step was 107% overload for 130 hours. The double circular arc reducer completed a total of 484 hours at the 145% to 207% load condition without failure. **It is concluded that the double circular arc gear reducer has at least a 150% over design capacity.**

The involute gear reducer completed four load steps for a total test time of 652 hours. Surface distress and micropitting began to occur at 80% of the design capacity of the reducer and the reducer was destroyed at 200 hours at 106% of the design load capacity. **It is concluded that the involute reducer has a load capacity of only 80% of its design capacity.**

Noise And Vibration Of Two Types Of Reducers

The double circular arc gear reducer measured slightly greater noise levels than the involute reducer of only two dB's. The vibration of the double circular arc reducer measured slightly greater vibration than the involute reducer; however both results were extremely low with a maximum of only 81.5 μin (2.1 μm).

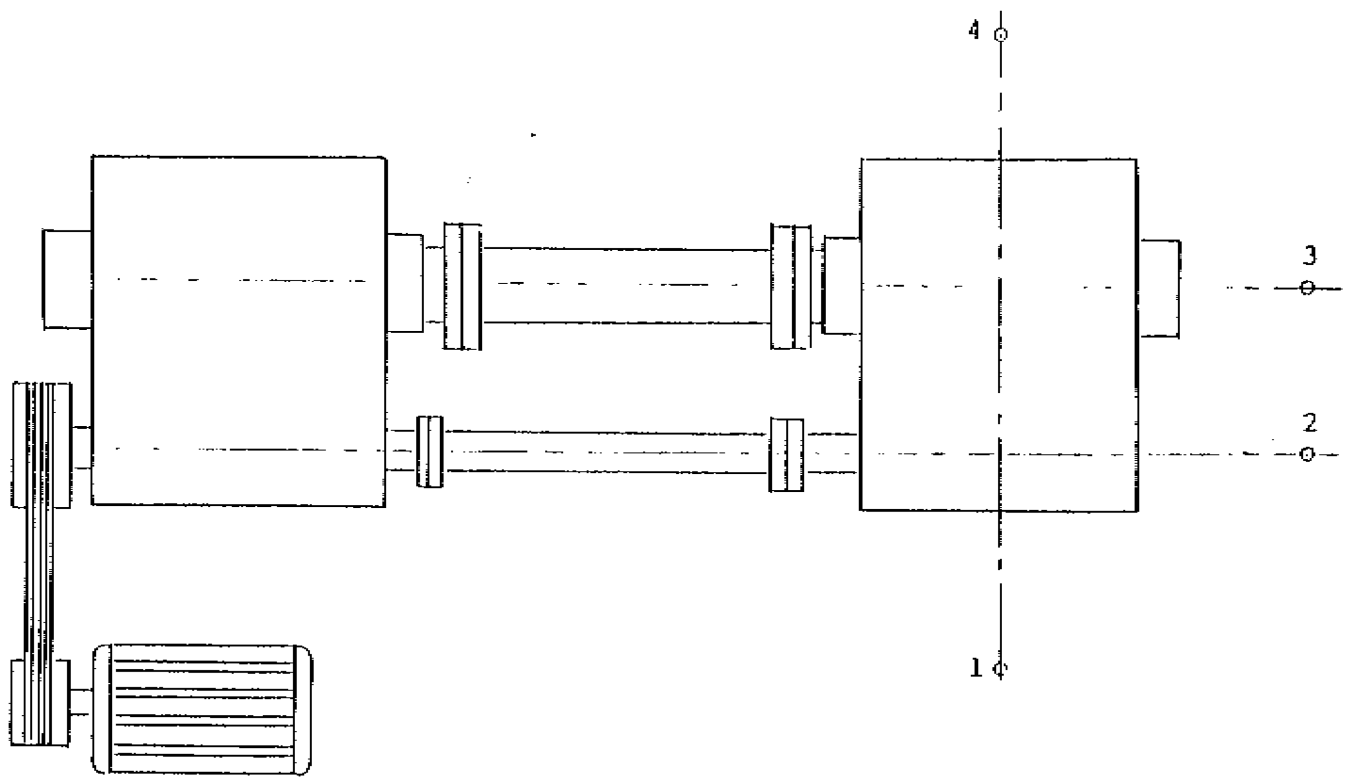


Figure 1. Closed loop test setup showing location of noise measurements

Table 1. Technical Data for Test Gear Reducers

		Involute gear with medium hard tooth flank	Double circular arc gear
Center distance	A	500mm 19.685in	500mm 19.685in
Gear ratio	i	31.73	31.73
Output torque	M_{out}	700 kg-M 60,768 lb-in	700 kg-M 60,768 lb-in
Driving shaft	M_{in}	22 kg-M 1,910 lb-in	22 kg-M 1,910 lb-in
Pressure angle	α_n	20^0	24^0
Tooth number	Z1/Z2 Z3/Z4	18/102 22/112	18/102 22/112
Helix angle	$\beta 1$ $\beta 3$	$25^0 50'31"$ $28^0 21'27"$	$25^0 50'31"$ $28^0 21'27"$
Module	Mn1,2 Mn3,4	3mm/8.47dp 4mm/6.35dp	3mm/8.47dp 4mm/6.75dp
Face width	B1,3 B3,4	45mm/1.77in 50mm/2.36in	45mm/1.77in 60mm/2.36in
Material	Z1.Z3 Z2,Z4	40CrNiMoA 35SiMn	40CrNiMoA 35SiMn
Lubricant		320# extreme pressure industrial gear oil	250# extreme pressure industrial gear oil
Heat treatment Hardness	Z1,Z3 Z2,Z4	HB340-375 HB285-320	HB280-310 HB240-270
Machining		Gear Hobbing	Gear Hobbing

Table 2. Loading Condition of Dual Circular Arc Gear Boxes

	Loading Number						Total
	1	2	3	4	5	6	
Input Static Torque Kg-m lb-in	12.15 1055	17.8 1545	23.84 2070	34.48 2993	41.91 3639	47.38 4113	
Input Dynamic Torque kg-m lb-in	10.66 7925	16.71 1451	20.1 1745	31.93 2772	40.12 3483	45.51 3951	
Percentage of Full Load %	48%	76%	91%	145%	182%	207%	
Cycles	$4x10^5$	$5x10^5$	$1.4x10^6$	$3.8x10^6$	$3.2x10^6$	$2.6x10^6$	
Operating time (hour)	20	25	70	193	161	130	599
Condition of tooth Surface	Two Contact Bands	Bright Normal	Normal	Normal	Normal	Normal	

Table 3. Record of Noise Measurement for Double Circular Arc Gear Reducer
Unit dB(A)

	Measuring Point					
Load	1	2	3	4	Average	Background Noise
12.15 kg-m 1055 lb-in	68.5	68	67	66.5	67.5	
23.84 kg-m 2070 lb-in	64	64	64	64	64	53
34.48 kg-m 2993 lb-in	71	66.5	66.5	65.5	68.1	51
41.92 kg-m 3639 lb-in	73	68	68	57.4	66.6	56
47.38 kg-m 4113 lb-in	70.5	67.5	65.5	61	67.3	50

Note: The background noise modifications have been made for the data of each point in Table 4.

Table 4. Record of Vibration Measurement Units (μ m)

Vibration of double circular arc gear reducer				Vibration of involute gear reducer			
Load	Z	X	Y	Load	Z	X	Y
23.84 kgf-m 2070 lb-in	1.47	0.63	0.84	16.72 kgf-m 1452 lb-in	0.84	0.7	0.98
34.48 kgf-m 2993 lb-in	1.4	1.82	1.26	21.63 lgf-m 1878 lb-in	0.56	0.28	0.42
41.92 kgf-m 3639 lb-in	0.91	1.47	1.19	28.17 kgf-m 2445 lb-in	0.42	0.28	0.42
47.38 kgf-m	1.33	2.1	1.4				

Table 5. Loading State of Involute Gear Reducer

	Loading Number				Total
	1st	2nd	3rd	4th	
Static Torque kg-m lb-in	13.19 1145	16.72 1452	21.63 1878	28.17 2446	
Dynamic Torque kg-m lb-in	9.5 825	14.62 1269	18.75 1628	23.24 2018	
Percentage of full load	43%	66%	80%	106%	
Operating time (hour)	92	160	200	200	652
Cycles	$1.84x10^6$	$3.2x10^6$	$4x10^6$	$4x10^6$	
Condition of Tooth Surface	Smaller Contact Area	Bright Tooth face Little Wear	Some Surface Distress	Considerable Surface Distress	

Table 6. Record of Noise Measurement for Involute Gear Reducer Unit dB(A)

Load	Measuring Point				Average	Background Noise
	1st	2nd	3rd	4th		
16.72 kg-m 1406 lb-in	65	62	60	60	62.2	50
21.63 kg-m 1878 lb-in	64.5	61.5	60.5	60	62	48
28.17 kg-m 2446 lb-in	66	62	60.5	60	62.8	52

Note: The background noise modifications were made for the data of each point in Table 4.

Section 2

Walking Beam Design

ANALYSIS OF THE DARCO WALKING BEAM

by

Robert H. Gault

The API Specification 11E, Sixteenth Edition, October 1, 1988 is used to determine the proper rating of the walking beams. Attached are copies of Section 2, Pumping Unit Structures, (Pages 5-9), which covers the sizing of walking beams and other structural members.

The walking beams which are presently furnished with the units are fabricated from steel plates. Since these are fabricated beams, the note in SPEC 11E, at the bottom of paragraph 2.4, Page 6 is pertinent and will be addressed.

The specifications for the walking beams of the selected units are as follows:

Pumping Unit Size	Beam Hgt. in. d	Beam Wgt. lb/ft	Wdth Flg. in. b	Thck Flg. in. t	Thck Web in.	Ix $in.^4$	Sx $in.^3$	Iy $in.^4$	J $in.^4$
114-119-86	17.8	109	11.8	0.71	0.47	1568	176	194	10.6
160-173-100	22.0	137	13.4	0.79	0.47	3049	277	316	11.3
228-213-100	27.2	110	9.9	0.94	0.55	3954	290	152	7.3
320-256-120	27.2	145	14.2	0.94	0.63	5456	401	448	10.9
456-256-144	27.6	160	14.2	1.10	0.63	6348	460	524	14.7
640-305-168	33.1	192	15.8	1.10	0.71	10647	643	723	20.5
912-365-192	33.1	220	15.8	1.30	0.78	12235	739	854	27.5

The walking beam ratings in API Spec 11E are based on ASTM A36 structural steel. This indicates a minimum yield strength of 36,000 psi. The physical and chemical tests of the walking beam material by MQS Inspection, Inc. show the beam material of the Darco units to be a higher grade of steel than the minimum specified. It has a yield strength of 64,400 psi. It is classified as 1518 steel.

The API specification makes no provisions for rating beams of higher quality steel, so calculations first will be made using the actual strength of the beam and then calculations will be made using the API formulas.

CALCULATIONS WITH ACTUAL BEAM SPECIFICATIONS (SAMPLING)

The formula for calculating beam stress is:

$$Smax = M/Sx$$

And for calculating Safety Factor:

$$SF = Sy / Smax$$

where:

Smax = Maximum stress in upper and lower fibers, psi
M = Maximum moment at support point, in. lbs.
Sx = Elastic section Modulus, in^3
Sy = Yield Strength of material, psi

Pumping Unit Size	Front Working Center	Beam Rating in/lbs	Beam Moment in/lbs	Sx In^3	Maximum Stress psi	Safety Factor SF
114-119-86	111	11900	1320900	176	7495	8.58
160-173-100	129	17300	2231700	277	8050	7.99
228-213-100	127	21300	2705100	291	9304	6.93
320-256-120	154	25600	3942400	401	9826	6.55
456-256-144	181	25600	4633600	460	10072	6.39
640-305-168	181	30500	5520500	643	8580	7.50
912-365-192	207	36500	7555500	739	10219	6.30

The above table shows that the actual strength of the walking beams exceed the maximum tensile and compressive stress loads, at full beam rating, with Safety Factors in excess of 6.30.

Since these are fabricated beams, the question of stresses at critical sections and stress concentrating factors must be addressed. Fabricated beams have one principle advantage. Since they are constructed of flat steel plate, they have a much more uniform cross section than rolled beams. The tolerances for rolled beams include allowances for camber, sweep, beam out of square, flanges not parallel, different flange thicknesses, web off center, beam height, flange width, etc. These tolerances are necessary because of variations in the rolling process. To get beams with the quality required for walking beam material from rolled beams, it has always been necessary to select the best possible beams available and then do additional processing at the point of manufacture. With fabricated beams, constructed from flat plate, it is easier to obtain the highest quality beam.

It is understood that the flanges are welded to the web with complete penetration welds. Since the weld material is of higher strength than the beam material, the joint will be stronger than the beam itself. Further, the stress at this point will be less than in the outermost fibers of the beam flanges. Since the beams are constructed with continuous uniform dimensions there will be no stress concentration factors.

CALCULATIONS WITH API SPEC 11E BEAM SPECIFICATIONS

The maximum Allowable Stress in Pumping Unit Walking Beams are listed in Table 2, Page 6 of the API SPEC 11E. In Paragraph 1 of this table the allowable "tensile stress in extreme fibers in bending, psi" for ASTM A36 beams is 11,000 psi. This is a safety factor of 3.27 (36000/11000). Since the yield strength of the Darco beam material is 64,400 psi, it will have a safety factor of 5.85 (64400/11000).

From Paragraph 2, the formula for allowable "compressive stress in extreme fibers in the bending, psi" is:

$$f_{cb} = \frac{6,000,000}{\frac{1 \times d}{b \times t}}$$

Using this API formula, which is based on a minimum yield strength of only 36,000 psi the allowable compressive bending stress is shown in the following table. For the Darco beams with a yield strength of 64,400, the theoretical allowable compressive bending stress is also shown. API SPEC 11E, Page 6, Table 2.1, Paragraph 3 limits the allowable stress to the smaller of the theoretical allowable bending stress of 11,000 psi.

Pumping Unit Size	Actual Beam Stress	API Theo. Allow Comprs Stress	API Theo. Safety Factor	Darco64 Theo. Allow Comprs Stress	Darco Theo. Safety Factor	API Actual Allow Stress
114-119-86	7495	25441	3.39	45511	6.07	11000
160-173-100	8050	22380	2.78	40035	4.97	11000
228-213-100	9304	16163	1.74	28914	3.68	11000
320-256-120	9826	19119	1.95	34202	3.48	11000
456-256-144	10072	18760	1.86	33560	3.33	11000
640-305-168	8560	17405	2.03	31136	3.62	11000
912-365-192	10219	17986	1.76	32175	3.15	11000

This table shows that the design of the Darco beams is quite conservative. The allowable stress formula has a built in safety factor of 3.27 in bending compression. Using this API formula the beams have extra safety factors because they are larger than the minimum size required. When the formula is adjusted for the extra yield strength of the Darco beams the extra safety factors are quite large. Since all of the above beams have an allowable much greater than 11,000 psi they are all limited to 11,000 psi.

The formula for determining the rating of the walking beam is shown in API SPEC 11E, Page 5, Paragraph 2.4,. It is:

$$W = \frac{f_{cb} \times S}{L}$$

where:

W = Walking beam rating in pounds of polished rod load
f_{cb} = Compressive stress in bending in pounds per square inch
S = Section modulus of the beam in cubic inches
L = Greater of the front or rear working center

Using this API Beam Rating Formula the following table was calculated:

Pumping Unit Size	Actual Beam Stress	Actual Beam Rating	Allow Beam Rating	Extra Safety Factor
114-119-86	7495	11900	17464	1.47
160-173-100	8050	17300	23637	1.37
228-213-100	9304	21300	25181	1.18
320-256-120	9826	25600	28658	1.12
456-256-144	10072	25600	27955	1.09
640-305-168	8580	30500	39099	1.28
912-365-192	10219	36500	39286	1.08

This table shows that the walking beams could be rated with higher load ratings than they are, even when no allowance is made for the stronger steel used in the beam.

In summary: the walking beams on the Darco units greatly exceed the requirements of API SPECIFICATION 11E.

RESUME

MR. ROBERT H. GAULT

Mr. Robert Gault graduated from Oklahoma University with a Bachelor of Science Degree in Mechanical Engineering. He was employed by Bethlehem Steel Corporation, Supply Division and served in various capacities prior to retiring as Manager of Product Development.

Mr. Gault also was an Associate Professor of Petroleum Engineering at Texas Tech University. He is currently an instructor for Oil & Gas Consultants International, specializing in artificial lift-sucker rod pumping worldwide. The following countries and their petroleum industries are currently utilizing his expertise: The United States of America, Australia, Argentina, Canada, Columbia, Ecuador, India, Peru, Oman, Sumatra, Singapore and Venezuela.

He also is an engineering consultant, specializing in pumping unit design analysis and sucker rod pumping problems for the oil industry. He is responsible for the design of the Bethlehem "BG" and CMI/Baker Torquemaster special geometry pumping units. He is a holder of numerous patents and publications in the field of artificial lift.

Mr. Gault is a recipient of the SWPSC "John C. Slonneger" Award for outstanding achievement in the field of artificial lift. He is also a recipient of the 1985 Society of Petroleum Engineers' "Production Engineer of the Year" award.

He currently serves a Secretary for the American Petroleum Institute Committee on the Standardization of Production Equipment. He is a member of the Task Group to update the recommended practice for designing sucker rod pumping systems (RP11L) and the Task Group on motor performance. He has held numerous past leadership positions within both the Society of Petroleum Engineers and the American Petroleum Institute.

Specification for Pumping Units

API SPECIFICATION 11E (SPEC 11E)
SIXTEENTH EDITION, OCTOBER 1, 1989

American Petroleum Institute
1220 L Street, Northwest
Washington, DC 20005

SECTION 2
PUMPING-UNIT STRUCTURES

2.1 Scope. This section covers:

a. Standardization of specific structure sizes in combination with established reducer sizes as given in Section 3.

b. Walking beam design, with specific rating formula.

c. Design loads and limiting working stresses on other structural components are also included.

NOTE: *Only loads imposed on the structure and/or gear reducer by the polished rod load are considered in this specification. Additional loads on the pumping unit imposed by add-on devices attached to the reducer, walking beam, or other structural components are not part of this specification. These would include such devices as compressors, stroke increasers, etc.*

2.2 No dimensional requirements, other than stroke length, are established. Rating methods are given only for polished-rod capacities; however, allowable working stresses of other structural components for a given polished rod capacity are defined.

Other design criteria such as bearing design, braking capacity, etc., are also established.

2.3 Standard Pumping-Unit Series. It is recommended that pumping units furnished to this specification adhere to the gear reducer rating, structure capacity, and stroke length as given in Table 2.2, although the combinations of these items that make up the pumping unit designation need not be identical to those in the table. The particular combinations in the table are typical but combinations other than those listed are acceptable under this standard.

NOTE: *It is the spirit and intent of above provision, that any manufacturer having authority to use the API monogram on pumping-unit structures, may not represent a structure carrying the monogram or for which the letters API or the words "American Petroleum Institute" are used in its description as having a rating of any kind or size other than provided above. This applies to sales information as well as to structure markings.*

2.4 Walking Beam. The following formula shall be used for rating conventional walking beams as shown in Fig. 2.1.

$$W = \frac{f_{cb}S}{L}$$

Wherein:

W = walking-beam rating in pounds of polished-rod load.

L = greater of l_f or l_r

f_{cb} = compressive stress in bending in pounds per square inch. See Table 2.1 for maximum allowable stress.

S = section modulus in cubic inches. The gross section of the rolled beam may be used except that holes or welds are not permissible on the tension flange in the critical zone. See Fig. 2.1.

l_f = distance from centerline of saddle bearing to centerline of well in inches. See Fig. 2.1.

l_r = distance from centerline of saddle bearing to centerline of equalizer bearing in inches. See Fig. 2.1.

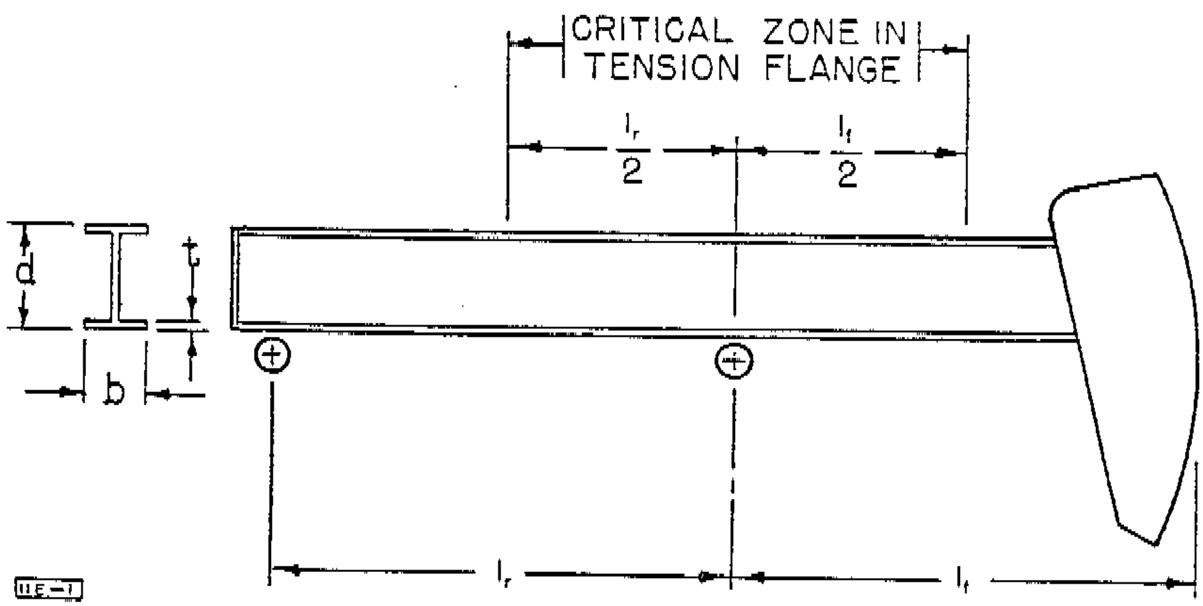


FIG. 2.1
WALKING-BEAM ELEMENTS

TABLE 2.1
MAXIMUM ALLOWABLE STRESSES IN PUMPING-UNIT WALKING BEAMS
(See Fig. 2.1)

1	2 Stress	3 Symbol	4 ASTM A36 Structural Steel
1	Tensile stress in extreme fibers in bending, psi.	f_{tb}	11,000
2	Compressive stress in extreme fibers in bending, psi. (May not exceed values on line 3)	f_{cb}	$\dfrac{{}^{*}6{,}000{,}000}{\frac{ld}{bt}}$
3	Maximum compressive stress in bending, except as limited by equation on line 2, psi.	f_{cb}	11,000
4	Minimum yield strength of material, psi.		36,000

*In the quantity $\frac{l}{b}\frac{d}{t}$:

l = longest laterally unbraced length of beam, inches (longer of l_r or l_f, see Fig. 2.1).
d = depth of beam section, inches.
b = width of compression flange, inches.
t = thickness of compression flange, inches.

NOTE: *The formula given in Par. 2.4 is based on the conventional beam construction using a single rolled section. With unconventional construction or built-up sections, due regard shall be given to change in loading, to checking stresses at all critical sections, and to the existence of stress concentrating factors.*

2.5 The working stress, f_{cb} for the beam rating formula given in Par. 2.4, shall be determined from Table 2.1. For standard rolled beams having cross sections symmetrical with the horizontal neutral axis, the critical stress will be compression in the lower flange. The maximum value of this stress, f_{cb}, is the smaller of the values determined from lines 2 and 3 of Table 2.1.

2.6 Unit Rotation. Viewed from the side of the pumping unit with the well head to the right, crank rotation is defined as either clockwise or counterclockwise.

2.7 Design Loads for All Structural Members Except Walking Beams. For all pumping unit geometries, and unless otherwise specified, use the maximum load exerted on the component in question by examining the loads on the component at each 15° crank position on the upstroke of the pumping unit. Use polished rod load W, for all upstroke crank positions. (See Par. 2.1c)

For units with bi-directional rotation and non-symmetrical torque factors, the direction of rotation for

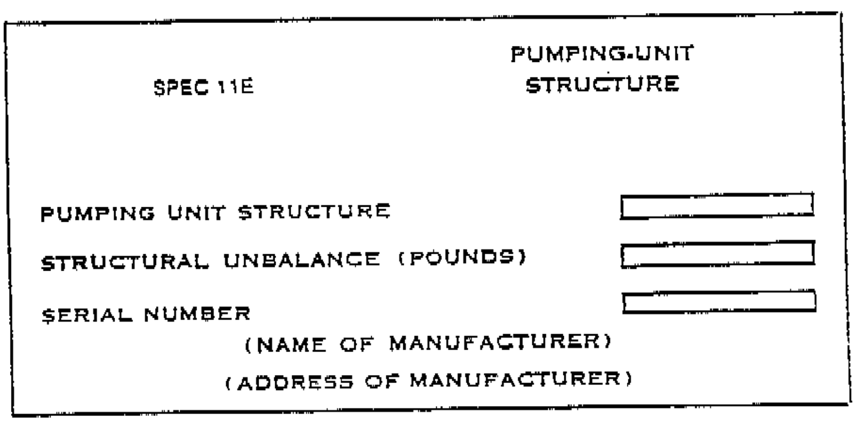


FIG. 2.2
PUMPING-UNIT STRUCTURE NAME PLATE

NOTE: *Structural unbalance is that force in pounds required at the polished rod to hold the beam in a horizontal position with the pitmans disconnected from the crank pins. This structural unbalance is considered positive when the force required at the polished rod is downward, and negative when upward. The minus (—) sign shall be stamped on the name plate when this value is negative.*

design calculations shall be that which results in the highest loading on structural components.

Due consideration shall be given to the direction of loading on all structural bearings and on the structural members supporting these bearings.

NOTE: *Allowable stress levels are based on simple stresses without consideration of stress risers. Adequate stress concentration factors shall be used when stress risers occur.*

2.8 Design Stresses for All Structural Members Except Walking Beam, Bearing Shafts and Cranks.

a. Design stresses for all structural components shall be a function of the yield strength of the material, Sy, psi.

b. Components subjected to simple tension or compression and non-reversing bending shall have a limiting stress of .3 Sy. If stress risers occur in critical zones of tension members, the limiting stress shall be .25 Sy.

c. Components subjected to reverse bending shall have a limiting stress of .2 Sy.

d. The following formula shall be used for all components acting as columns:

$$W_z = \frac{a\,Sy}{4}\left[1 - \frac{Sy}{4n\,\pi^2 E}\left(\frac{l}{r}\right)^2\right]$$

Wherein:

W_z = maximum applied load on column, lbs.

a = area of cross section, sq. in.

Sy = yield strength of material, psi

n = end restraint constant (assume = 1)

E = modulus of elasticity, psi

l = unbraced length of column, in.

r = radius of gyration of section, in.

$\frac{l}{r}$ shall be limited to a maximum of 90. For $\frac{l}{r}$ values of 30 or less, columns may be assumed to be acting in simple compression (See Par. 2.8b).

2.9 Shafting. All bearing shafts as well as other structural shafting shall have limiting stresses as outlined in Par. 3.8 in the reducer section of this specification.

2.10 Hanger. Wirelines for horseheads shall have a minimum factor of safety of five when applied to the breaking strength of the wireline.

For allowable stresses on carrier bar, end fittings, etc., see Par. 2.8b and 2.8c.

2.11 Brakes. Pumping unit brakes shall have sufficient braking capacity to withstand a torque exerted by the cranks at any crank position with a maximum amount of counterbalance torque designed by the manufacturer for the particular unit involved. This braking torque to be effective with the pumping unit at rest under normal operating conditions with the well disconnected.

NOTE: *The pumping unit brake is not intended as a safety stop but is intended for operational stops only.*

When operations or maintenance are to be conducted on or around a pumping unit, the position of the crank arms and counterweights should be securely fixed in a stationary position by chaining or other acceptable means.

2.12 Horseheads. Horseheads shall be either hinged or removable to provide access for well servicing.

Horseheads shall be attached to the walking beam in such a manner as to prevent falling off due to a high rod part or other sudden load changes.

The distance from the pivot point of the horsehead to the tangent point of the wireline on the horsehead shall have a maximum dimensional tolerance at any position of the stroke of the following values:

±½ in. for stroke lengths to 100 in.
±⅝ in. for stroke lengths 100 in. to 200 in.
±¾ in. for stroke lengths of 200 in. and longer

2.13 Cranks. All combined stresses in cranks shall be limited to a maximum value of .15 Sy.

2.14 Structural Bearing Design. Structural bearing shafts may be supported in sleeve or anti-friction bearings.

a. Anti-Friction Bearings.

For bearings subject to oscillation or rotation use the bearing load ratio formula:

$$R_l = k\,\frac{C_l}{W_l}$$

Where:

R_l = bearing load ratio
k = 1 for bearings rated at 33-1/3 rpm and 500 hours or
k = 3.86 for bearings rated at 500 rpm and 3000 hours
C_l = bearing manufacturer's specific dynamic rating in lbs.
W_l = maximum load on bearing in lbs.

For bearings subject to oscillation only use an R_l value of 2.0 or greater.

For bearings subject to rotation use an R_l value of 2.25 or greater.

TABLE 2.2
PUMPING UNIT SIZE RATINGS

1	2	3	4
Pumping Unit Size	Reducer Rating, in.-lb	Structure Capacity, lb	Max. Stroke Length, in.
6.4—32—16	6,400	3,200	16
6.4—21—24	6,400	2,100	24
10—32—24	10,000	3,200	24
10—40—20	10,000	4,000	20
16—27—30	16,000	2,700	30
16—53—30	16,000	5,300	30
25—53—30	25,000	5,300	30
25—56—36	25,000	5,600	36
25—67—36	25,000	6,700	36
40—89—36	40,000	8,900	36
40—76—42	40,000	7,600	42
40—89—42	40,000	8,900	42
40—76—48	40,000	7,600	48
57—76—42	57,000	7,600	42
57—89—42	57,000	8,900	42
57—95—48	57,000	9,500	48
57—109—48	57,000	10,900	48
57—76—54	57,000	7,600	54
80—109—48	80,000	10,900	48
80—133—48	80,000	13,300	48
80—119—54	80,000	11,900	54
80—133—54	80,000	13,300	54
80—119—64	80,000	11,900	64
114—133—54	114,000	13,300	54
114—143—64	114,000	14,300	64
114—173—64	114,000	17,300	64
114—143—74	114,000	14,300	74
114—119—86	114,000	11,900	86
160—173—64	160,000	17,300	64
160—143—74	160,000	14,300	74
160—173—74	160,000	17,300	74
160—200—74	160,000	20,000	74
160—173—86	160,000	17,300	86
228—173—74	228,000	17,300	74
228—200—74	228,000	20,000	74
228—213—86	228,000	21,300	86
228—246—86	228,000	24,600	86
228—173—100	228,000	17,300	100
228—213—120	228,000	21,300	120
320—213—86	320,000	21,300	86
320—256—100	320,000	25,600	100
320—305—100	320,000	30,500	100
320—213—120	320,000	21,300	120
320—256—120	320,000	25,600	120
320—256—144	320,000	25,600	144
456—256—120	456,000	25,600	120
456—305—120	456,000	30,500	120
456—365—120	456,000	36,500	120
456—256—144	456,000	25,600	144
456—305—144	456,000	30,500	144
456—305—168	456,000	30,500	168
640—305—120	640,000	30,500	120
640—256—144	640,000	25,600	144
640—305—144	640,000	30,500	144
640—365—144	640,000	36,500	144
640—305—168	640,000	30,500	168
640—305—192	640,000	30,500	192
912—427—144	912,000	42,700	144
912—305—168	912,000	30,500	168
912—365—168	912,000	36,500	168
912—305—192	912,000	30,500	192
912—427—192	912,000	42,700	192
912—470—240	912,000	47,000	240
912—427—216	912,000	42,700	216
1280—427—168	1,280,000	42,700	168
1280—427—192	1,280,000	42,700	192
1280—427—216	1,280,000	42,700	216
1280—470—240	1,280,000	47,000	240
1280—470—300	1,280,000	47,000	300
1824—427—192	1,824,000	42,700	192
1824—427—216	1,824,000	42,700	216
1824—470—240	1,824,000	47,000	240
1824—470—300	1,824,000	47,000	300
2560—470—240	2,560,000	47,000	240
2560—470—300	2,560,000	47,000	300
3648—470—240	3,648,000	47,000	240
3648—470—300	3,648,000	47,000	300

b. Sleeve Bearings.

The design of sleeve bearings is beyond the scope of this specification. It shall be the responsibility of the pumping unit manufacturer to design sleeve bearings, based on available test data and field experience, which are comparable in performance to anti-friction bearings designed for the same operating loads and speeds.

2.15 Installation Markings. Clearly defined and readily usable markings shall be provided on the end cross members of the base to indicate the vertical projection of the walking beam centerline. The markings shall be applied with a chisel, punch, or other suitable tool.

2.16 Marking.* Each pumping-unit structure shall be provided with a name plate substantially as shown in Fig. 2.2. At the discretion of the manufacturer, the name plate may contain other non-conflicting and appropriate information, such as model number or lubrication instructions.

2.17 In order that the torque on a reducer may be determined conveniently and accurately from dynamometer test data, manufacturers of pumping units shall provide, on request of the purchaser, stroke and torque factors for each 15-deg position of the crank. An approved form for the submission of these data is shown in Appendix A.

*Users of this specification should note that there is no longer a requirement for marking a product with the API monogram. The American Petroleum Institute continues to license use of the monogram on products covered by this specification but it is administered by the staff of the Institute separately from the specification. The policy describing licensing and use of the monogram is contained in Appendix H, herein. No other use of the monogram is permitted.

SECTION 3
PUMPING-UNIT REDUCERS

3.1 SCOPE

Applicability. This Specification is applicable to enclosed speed reducers wherein the involute gear tooth designs include helical and herringbone gearing. This Specification is intended primarily for beam-type pumping units.

Limitations. The rating methods and influences identified in this Specification are limited to single and multiple stage designs applied to oilfield pumping units, in which the pitch-line velocity of any stage does not exceed 5000 feet per minute and the speed of any shaft does not exceed 3600 revolutions per minute.

3.2 RESPONSIBILITY

Gear Reducer Designers. Professionals using this Specification should realize that it is quite difficult to identify and offer solutions to all the influences affecting a gear reducer. For this reason, it is recommended that this Specification be used by engineers with significant experience in mechanical systems.

Reducers rated under this Specification, and properly applied, installed, lubricated and maintained, shall be capable of safely carrying the rated peak torque under normal oilfield conditions.

Rating Factors. The allowable stress numbers in this Specification are maximum allowed values. Less conservative values for other rating factors in this Specification shall not be used.

Metallurgy. The allowable stress numbers, s_{ac} and s_{at}, included in this Specification are based on commercial ferrous material manufacturing practices. Hardness, tensile strength, and microstructure are the criteria for allowable stress numbers. Reasonable levels of cleanliness and metallurgical controls are required to permit the use of the allowable stress numbers contained in this Specification.

Residual Stress. Any material having a case-core relationship is likely to have residual stresses. If properly managed, these stresses will be compressive and will enhance the bending strength performance of the gear teeth. Shot peening, case carburizing, nitriding, and induction hardening are common methods of inducing compressive prestress in the surface of the gear teeth.

Section 3

Metallurgical Analysis

MQS

Formerly MAGNAFLUX Quality Services
MQS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

LABORATORY NO. 59074-4-5 thru

DATE REPORTED: 5-16-89
REVISED REPORT 5-24-89*
PURCHASE ORDER NO. 0048

SHIPPER NO. ---

DARCO USA INC.
10835 Geary Avenue
P.O. Box 4207
Santa Fe Springs, CA 90670-8207

SAMPLE SUBMITTED: Steel

MATERIAL SPECIFICATION: ---

TEST METHOD: ---

QUANTITATIVE CHEMICAL ANALYSES

Element	% High Speed Pinion	% Crank Pin	% Bolt
Carbon	.41	.41	.35
Manganese	.81	.69	.60
Silicon	.28	.30	.24
Phosphorous	.017	.017	.016
Sulfur	.008	.013	.016
Chromium	.82	.30	.91
Nickel	1.63	.05	.05
Molybdenum	.25	.01	.02
Copper	.07	.07	.11
Iron	Rem.	Rem.	Rem.
Hardness	RC 22	RB 98	RC 25

(*). Revised report to remove dash number and spell out Element per client request

By: [signature] V. Philli
MQS INSPECTION, INC.

CHEMICAL • MECHANICAL • METALLURGICAL • FAILURE ANALYSIS • NONDESTRUCTIVE TESTING

MQS

Formerly MAGNAFLUX Quality Services

MQS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

DARCO USA INC.
10135 Geary Avenue
P.O. Box 4207
Santa Fe Springs, CA 90670-1207

Attn: Purchasing Agent

LABORATORY NO. 44081-4-1

DATE REPORTED: 1-20-92

PURCHASE ORDER NO. 0648

SHIPPER NO. DS #100

SAMPLE SUBMITTED: Steel

MATERIAL SPECIFICATION: ---

TEST METHOD: ---

QUANTITATIVE CHEMICAL ANALYSIS

Slow Speed Crank Shaft

Element	%
C	.46
Mn	.62
P	.011
S	.023
Si	.29
Cr	.15
Ni	.08
Mo	.01
Cu	.10
Fe	Rem.

ANALYSIS INDICATES SAMPLE IS C1045 STEEL

By: Scott Hauser Scott Hauser
MQS INSPECTION, INC.

M&P 5

CHEMICAL • MECHANICAL • METALLURGICAL • FAILURE ANALYSIS • NONDESTRUCTIVE TESTING

Formerly MAGNAFLUX Quality Services

MOS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

LABORATORY NO. 44081-5-1

DATE REPORTED: 1-6-92

PURCHASE ORDER NO. 0648

SHIPPER NO. DS # 100

SAMPLE SUBMITTED: ---

MATERIAL SPECIFICATION: ---

TEST METHOD: ---

DARCO USA INC.
10135 Geary Avenue
P.O. Box 4207
Santa Fe Springs, CA 90670-1207

Attn: Purchasing Agent

METALLURGICAL REPORT

Sample Identification

Slow Speed Crank Shaft

Test Required

To perform hardness test.

Test Results

Hardness, Rockwell C

36-37

By: Ben Wood
MOS INSPECTION, INC.

MQS

Formerly MAGNAFLUX Quality Services

MQS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

DARCO USA INC.
10135 Geary Avenue
P.O. Box 4207
Santa Fe Springs, CA 90670-1207

Attn: Frank Burson

LABORATORY NO. 79064-4-1 thru

DATE REPORTED: 7-17-89

PURCHASE ORDER NO. 0069

SHIPPER NO. ___

SAMPLE SUBMITTED: Steel

MATERIAL SPECIFICATION: ___

TEST METHOD: ___

QUANTITATIVE CHEMICAL ANALYSES

Element	% -1 Equalizer Beam	% -2 Equalizer to Pitman Arm Housing	% -3 Pitman Arm
C	.21	.27	.13
Mn	.51	.64	.49
Si	.24	.37	.25
P	.016	.026	.011
S	.024	.019	.025
Cr	.01	.17	.01
Ni	.01*	.14	.01
Mo	.01*	.07	.01*
Cu	.05	.13	.02
Fe	Rem.	Rem.	Rem.

(*) less than

-1 ANALYSIS INDICATES SAMPLE IS 1020 STEEL
-2 ANALYSIS INDICATES SAMPLE IS 1025 STEEL
-3 ANALYSIS INDICATES SAMPLE IS 1015 STEEL

By: V. Phillips [signature] V. Philli
MQS INSPECTION, INC.

CHEMICAL • MECHANICAL • METALLURGICAL • FAILURE ANALYSIS • NONDESTRUCTIVE TESTING

JUL 19 1989

MQS

Formerly MAGNAFLUX Quality Services
MQS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

DARCO USA INC.
P.O. Box 4207
Santa Fe Springs, CA 90670-1207

Attn: Frank Burson

LABORATORY NO. 79064-6-1
DATE REPORTED: 7-14-89
PURCHASE ORDER NO. 0069
SHIPPER NO. ---
SAMPLE SUBMITTED: Steel Coupons
MATERIAL SPECIFICATION: ---
TEST METHOD: ASTM E8

CERTIFIED TEST REPORT

			Yield Strength		Ultimate Strength		Elongation		Reduction Area	
Identification Number	Stressed Dimension	Stressed Area	Actual Load Pounds	Pounds per Sq. In.	Actual Load Pounds	Pounds per Sq. In.	Inches	Percent	Reduced Dimension	Percent
ROOM TEMPERATURE TENSILE TEST										
Equalizer Beam										
1	.2475	.0481	2100	43700	3275	68100	.32	32	.157	60
.qualizer to Pitman Arm Housing										
2	.2473	.0480	2350	49000	3875	80700	.31	31	.167	54
Pitman Arm										
3	.251	.0495	1900	38400	2910	58800	.34	34	.138	70
MAXIMUM REQUIREMENTS										
MINIMUM REQUIREMENTS				---		---		---		---

YIELD STRENGTH DETERMINE AT: 0.2% Offset
ELONGATION GAGE LENGTH: 1 inches
SPEED OF TESTING: 0.05 in/min
HEAT TREATED AS FOLLOWS:
WITNESSED BY:

FRACTURE CODE:
(F) Flaw present.
(G) Broke outside gage mark.
(g) Broke through gage outside middle half length.

By: C. Mc
MQS INSPECTION, INC.

CHEMICAL • MECHANICAL • METALLURGICAL • FAILURE ANALYSIS • NONDESTRUCTIVE TESTING

P

JUN 28 1

MQS

Formerly MAGNAFLUX Quality Services

MQS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

DARCO U.S.A. INC
10135 Geary Avenue
Santa Fe Springs, CA. 90670-1207

Attn: William F. Wheeler

LABORATORY NO. 69341-4-1

DATE REPORTED: 6-27-89

PURCHASE ORDER NO. 0061

SHIPPER NO. ----

SAMPLE SUBMITTED: Steel Coupo

MATERIAL SPECIFICATION: ----

TEST METHOD: ----

QUANTITATIVE CHEMICAL ANALYSIS

Pumping Unit Walking Beam

ELEMENT	%
CARBON	.16
MANGANESE	1.28
SILICON	.30
PHOSPHOROUS	.025
SULFER	.009
CHROMIUM	.02
NICKEL	.01*
MOLYLIDENUM	.01*
COPPER	.01*
IRON	Rem.

ANALYSIS INDICATES SAMPLE IS 1518 STEEL

*Less Than

By: Victor Phillips
MQS INSPECTION, INC.

JUN 28 19

MQS

Formerly MAGNAFLUX Quality Services
MQS INSPECTION, INC.
6800 EAST WASHINGTON BOULEVARD
LOS ANGELES, CALIFORNIA 90040
TELEPHONE (213) 724-3811

DARCO USA INC.
10135 Geary Avenue
Santa Fe Springs, CA 90670-1207

Attn: W. F. Wheeler

LABORATORY NO. 69341-6-1
DATE REPORTED: 6-26-89
PURCHASE ORDER NO. 0061
SHIPPER NO. ---
SAMPLE SUBMITTED: Pumping Unit Walking Beam Section
MATERIAL SPECIFICATION: ---
TEST METHOD: ASTM E8

CERTIFIED TEST REPORT

			YIELD STRENGTH		ULTIMATE STRENGTH:		ELONGATION		REDUCTION AREA	
IDENTIFICATION NUMBER	STRESSED DIMENSION	STRESSED AREA	ACTUAL LOAD POUNDS	POUNDS PER SQ. IN.	ACTUAL LOAD POUNDS	POUNDS PER SQ. IN.	INCHES	PERCENT	REDUCED DIMENSION	PERCENT
ROOM TEMPERATURE TENSILE TEST										
	.506	.2011	12950	64400	16300	81100	.60	30	.305	6[illegible]
MAXIMUM REQUIREMENTS										
MINIMUM REQUIREMENTS				----		---		---		---

YIELD STRENGTH DETERMINE AT: Halt of the dial
ELONGATION GAGE LENGTH: 2 inches
SPEED OF TESTING: 0.05 in/min
HEAT TREATED AS FOLLOWS:
WITNESSED BY:

FRACTURE CODE:
(F) Flaw present.
(G) Broke outside gage mark.
(g) Broke through gage outside middle half length.

By: C. Mc
MQS INSPECTION, INC.

CHEMICAL • MECHANICAL • METALLURGICAL • FAILURE ANALYSIS • NONDESTRUCTIVE TESTING

Section 4

Bearing Content

BEARING CONTENT

MODEL	PITMAN	SADDLE	EQUALIZER	HIGH SPEED	INTERMEDIATE	LOW SPEED
114	22316C	NJ2228E	22326C	NU2313E	NU2317E	22230E
160	22316C	NJ2228E	22326C	NU2315E	NU2320E	22232C
228	22318C	NJ2228E	22326C	NU2317E	NU2322E	23136C
320	22322C	NJ2330E	22330C	NU2319E	NU2324E	32238C
456	22326C	NJ2334E	22336C	NU2322E	NU2326E	32240C
640	22328C	NJ2336E	22340C	NU2324E	NU2330E	22244C
912	22328C	NJ2336E	22340C	NU2324E	NU2330E	22244C

All bearings shown above are SKF or equivalent and can be purchased via bearing distributors nationwide.

Bearing L-10h life comparisons are listed in this section and utilize American Manufacturing's previous data on its products and that of Lufkin Industries.

BEARING LIFE COMPARISONS

The L-10h "Fatigue Life" of a bearing refers to the basic rating life in operating hours. Listed below are L-10h comparisons of the three primary pumping unit manufacturer's gear reducer bearings.

Formula:

$$L10h = \frac{1,000,000}{n \times 60} \left(\frac{C}{P}\right)^{\frac{10}{3}}$$

n = Rotational speed of bearing, RPM
C = Basic load rating of bearing
P = Radial load on bearing

Data derived from the Link Belt and SKF technical journals.

API Size 114D Reducer

High Speed Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2313	24750	1797	580.4	179,836
Lufkin	5213	24300	1797	580.4	169,167
American	5215	29500	1797	580.4	322,875

Intermediate Bearing Comparison:

Darco	2317	66825	5359	107.1	699,696
Lufkin	5216	33300	5359	107.1	68,642
American	5218	46200	5359	107.1	204,449

API Size 160D Reducer

High Speed Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2315	41175	2474	580.4	337,987
Lufkin	5215	29500	2474	580.4	111,223
American	5217	41100	2474	580.4	335,940

Intermediate Bearing Comparison:

Darco	2320	99000	7402	107.1	883,784
Lufkin	5217	41100	7402	107.1	47,174
American	5218	46200	7402	107.1	69,668

BEARING LIFE COMPARISONS

API Size 228D Reducer

High Speed Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2317	66825	3250	600.7	660,759
Lufkin	5216	33300	3250	600.7	64,823
American	5218	46200	3250	600.7	193,072

Intermediate Bearing Comparison:

Darco	2322	153450	9000	106.3	2,000,153
Lufkin	5218	46200	9000	106.3	36,586
American	5221	66500	9000	106.3	123,191

API Size 320D Reducer

High Speed Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2319	51525	4249	606.3	112,615
Lufkin	5218	46200	4249	606.3	78,285
American	5219	53000	4249	606.3	123,725

Intermediate Bearing Comparison:

Darco	2324	178200	13209	107.0	910,452
Lufkin	5220	59900	13209	107.0	24,043
American	5224	88300	13209	107.0	87,653

API Size 456D Reducer

High Speed Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2322	153450	5783	476.5	1,949,088
Lufkin	5220	59900	5783	476.5	84,729
American	5221	66500	5783	476.5	120,046

Intermediate Bearing Comparison:

Darco	2326	210600	16365	83.5	996,866
Lufkin	5224	88300	16365	83.5	61,104
American	5226	94000	16365	83.5	75,271

BEARING LIFE COMPARISONS

API Size 640D Reducer

High Speed Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2324	178200	7081	468.1	1,662,980
Lufkin	5224	88300	7081	468.1	160,102
American	5224	88300	7081	468.1	160,102

Intermediate Bearing Comparison:

	Brg #	C	P	RPM	L10h
Darco	2330	427500	19647	85.8	5,586,772
Lufkin	5226	94000	19647	85.8	35,848
American	5228	117000	19647	85.8	74,357

API Size 912D Reducer

The Darco bearings are identical to that of our 640D model.

Section 5

Client References

Client List

Aera Energy
Agip (Egypt)
Amoco
Arco
BP
Cal-Tex (Indonesia)
Chevron
CNPC (China National Petroleum Corp.)
El Paso
Exxon USA
Fina Oil & Chemical
Kelt Energy
Maxus
Mobil Oil
Neste Oy
ONCG (India)
Pacific Enterprises
Perenco (France)
Petrobras (Brazil)
Phillips Petroleum
Pioneer Resources
Repsol (Venezuela)
Santa Fe Energy Resources
Shell PDO (Oman)
Tecpetrol S.A.
Texaco
Total
Unocal
YPF (Argentina)
Additional references upon request