INTRODUCTION TO PUMPING UNIT TECHNOLOGY

PRESENTED BY

EARL H. SIGMAN, P.E.

MANAGER OF ENGINEERING

OF

MORGAN PETROLEUM EQUIPMENT

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SLIDE/TAPE PRESENTATION IS DEDICATED TO A BETTER UNDERSTANDING OF PUMPING UNIT TECHNOLOGY.

A SPECIAL THANKS TO THE SEVERAL COMPANIES DETAILED IN THE BIBLIOGRAPHY FOR THEIR PERMISSION TO INCORPORATE MATERIAL INTO THIS SLIDE PRESENTATION.

THE TITLE OF THIS LECTURE IS "INTRODUCTION TO PUMPING UNIT TECHNOLOGY" AND I THINK IT IS IMPORTANT FOR EVERYONE INVOLVED IN THE MANUFACTURE OF PUMPING UNITS, ALSO KNOWN AS PUMPJACKS, TO KNOW A LITTLE BIT ABOUT THE LOCATION OF OIL, THE METHODS OF DRILLING AND A LOT ABOUT THE DOWNHOLE EQUIPMENT.

SOME OF YOU ARE PROBABLY QUITE KNOWLEDGEABLE ON THIS SUBJECT, BUT AS THIS LECTURE IS FOR THE BEGINNERS, WE WILL QUICKLY REVIEW THIS DOWNHOLE SITUATION.

THE PURPOSE OF THE PUMPJACK IS TO STROKE THE DOWN HOLE PUMP. ANYTHING THAT AFFECTS THE DOWNHOLE PUMP AFFECTS THE PUMPJACK ON THE SURFACE. THE MAJOR MISTAKE MADE BY PUMPING UNIT MANUFACTURERS IS NOT TO UNDERSTAND THE DOWNHOLE LOADING.

BEING A MECHANICAL ENGINEER, I AM NOT KNOWLEDGEABLE ABOUT WHERE TO DRILL FOR OIL, BUT - IT IS MY UNDERSTANDING THAT OIL IS TRAPPED BENEATH THE SURFACE OF THE GROUND IN POROUS ROCKS OR SAND, WHICH DOES NOT ALLOW THE GAS, THE OIL OR THE WATER TO ESCAPE TO THE SURFACE.

OIL AND GAS ARE CONSIDERED TO BE FLUIDS WHICH MIGRATE THROUGH THE POROUS ROCK OR SAND ZONES UNTIL THEY REACH SOME KIND OF GEOLOGICAL FORMATION TRAP WHICH TRAPS THE FLUID AND CAUSES A POOL, OR RESERVOIR TO FORM.

THREE OF THESE TRAPS ARE A FAULT, AN ANTICLINE AND A STRATIGRAPHIC TRAP. SOME OF WHICH ARE ILLUSTRATED ON PAGE 3.

THE FAULT IS CAUSED BY A FRACTURE IN THE CAP DENSE ROCK STRUCTURE AND A SLIPPING OF THIS ROCK TO BLOCK THE FLOW OF THE FLUIDS IN THE POROUS ROCK. THE OIL AND GAS GATHER AT THIS LOCATION, ESPECIALLY WHEN IT HAS AN UPHILL SLOPE.

THE ANTICLINE IS A RISING OF THE ROCK STRUCTURE CREATING A PEAK TOWARD THE SURFACE AND AS THE OIL AND GAS FLOAT ON THE WATER, IT COLLECTS IN THIS PEAK.

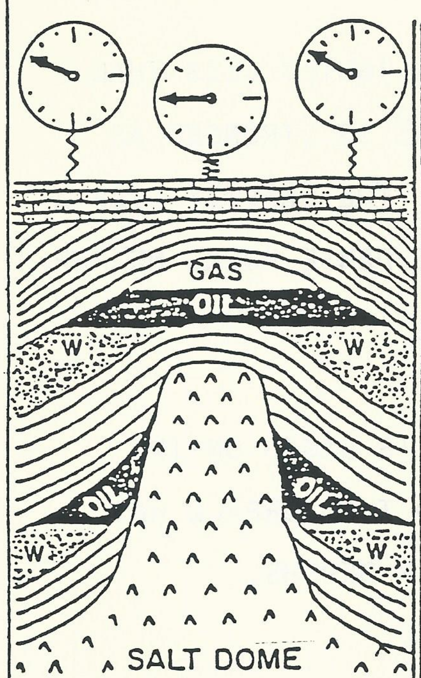
THE STRATIGRAPHIC TRAP IS A RISING OF THE DENSE ROCK BELOW THE POROUS ROCK OR SANDS TO MEET WITH THE DENSE CAPROCK, SHALE, OR CLAY ABOVE THE POROUS ROCK TO FORM A POCKET FOR OIL AND GAS TO COLLECT IN. THESE TRAPS ARE NOT STRUCTURAL FAULT TRAPS BUT ARE FORMED BY THE NATURAL SEDIMENTARY PROCESS.

SALT DOMES RISING UP FROM DEEP WITHIN THE EARTH CAUSE THESE TYPES OF FORMATIONS TO FORM AS SHOWN ON PAGE 3.

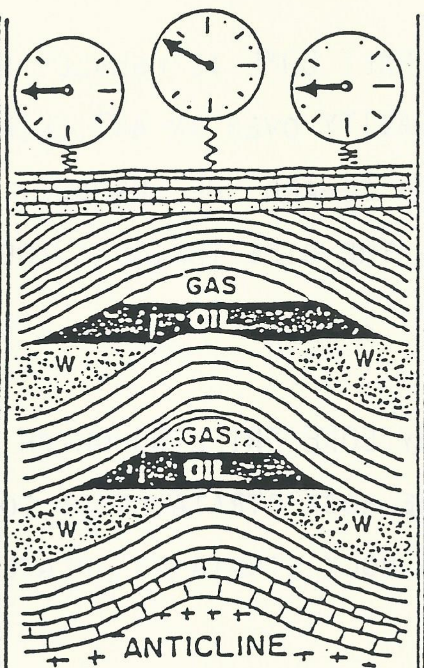
THERE ARE SEVERAL METHODS TO LOCATE THESE GEOLOGICAL TRAPS. THE FOREMOST SCIENTIFIC INSTRUMENT IS THE SEISMOGRAPH.

THE SEISMOGRAPH CAN BE USED TO FIND THESE GEOLOGICAL STRUCTURES BOTH ON WATER AND LAND. MEASURING GRAVITY CAN ALSO INDICATE THESE GEOLOGICAL STRUCTURES.

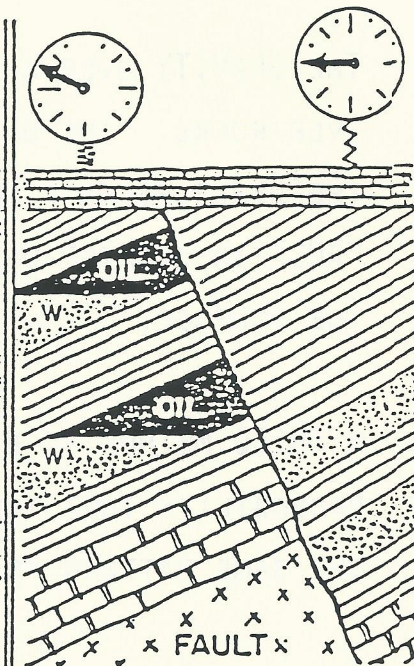
GRAVITY METER



Gravity slightly less over salt dome; salt lighter than rocks.

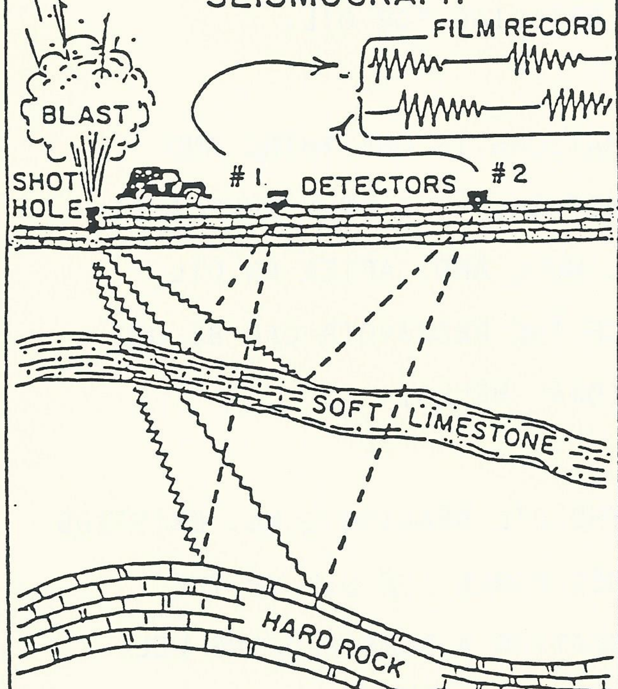


Gravity stronger over anticline; denser basement rocks nearer surface.



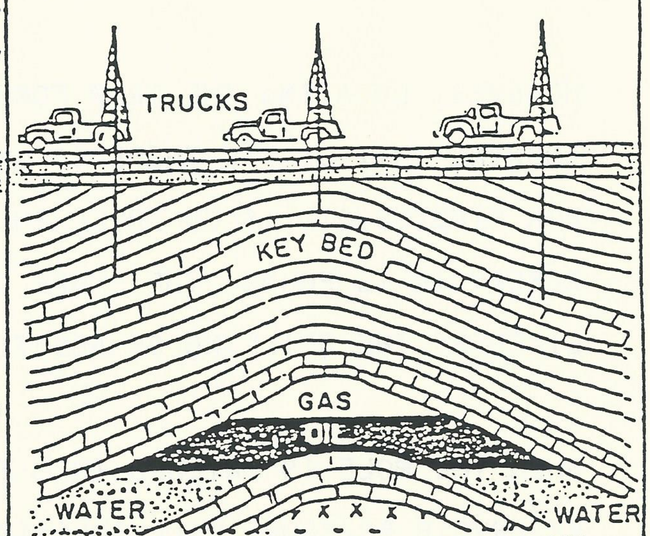
Gravity greater on left side of fault; dense rocks are nearer surface there.

SEISMOGRAPH



The seismograph is the foremost scientific instrument now in use for the location of hidden structures which may contain oil or natural gas.

CORE-DRILLING



Core-drilling is based on the probability that folds found in rock strata close to surface contain about same pattern at greater depths. By drilling a series of cores over a given area, a reasonably accurate picture of sub-surface geology can be obtained.

THE GRAVITY OVER A SALT DOME IS USUALLY SLIGHTLY LESS THAN OVER ROCKS. THE GRAVITY OVER AN ANTICLINE IS STRONGER AS THE DENSE ROCKS RISE TOWARD THE SURFACE. THE GRAVITY OVER A FAULT IS GREATER ON THE SIDE THAT HAS THE DENSE ROCKS NEARER THE SURFACE.

A THIRD METHOD IS BY CORE DRILLING WHICH IS BASED ON THE PROBABILITY THAT THE ROCK STRATA CLOSE TO THE SURFACE HAS THE SAME PATTERN AS ROCK STRATA AT GREATER DEPTHS.

AS I SAID BEFORE, THE FLUID MIGRATES THROUGH THE ROCK FORMATIONS UNTIL IT REACHES SOME KIND OF A TRAP WHERE IT WILL GATHER IN A RESERVOIR, USUALLY UNDER PRESSURE, WHICH BECOMES THE IDEAL LOCATION FOR DRILLING FOR OIL.

HOWEVER, LOCATING THE TRAP FORMATIONS IS ONE THING AND KNOWING THAT OIL IS THERE IS ANOTHER. THE ONLY TRUE TEST FOR FINDING OIL IS TO DRILL THE HOLE AND, AFTER AN OIL RESERVOIR IS FOUND, THE SHAPE OF THE RESERVOIR CAN BE OUT-LINED BY THE DRILLING OF ADDITIONAL WELLS.

WHEN THE HOLE IS DRILLED INTO THE OIL BEARING ZONE, EXISTING RESERVOIR PRESSURE WILL SOMETIMES FORCE THE OIL UP THE DRILLED HOLE TO THE SURFACE, CREATING A FREE-FLOWING WELL. SOMETIMES THE OIL WILL GO PARTIALLY UP THE HOLE AND HAVE TO BE PUMPED EVEN IF THEY ARE INITIALLY FREE-FLOWING; MOST IN THE MIDDLE EAST ARE FREE-FLOWING.

THE NATURAL PRESSURE OF THE RESERVOIR WHICH DRIVES THE FLUIDS TO THE SURFACE IS A WATER DRIVE, WHICH IS SIMPLY WATER UNDER PRESSURE WITH ENOUGH VOLUME TO LIFT THE OIL WHICH FLOATS ON WATER; A DISSOLVED GAS DRIVE, WHICH IS GAS UNDER PRESSURE DISSOLVED IN THE OIL AND WITH ENOUGH VOLUME TO DRIVE THE OIL AND GAS TO THE SURFACE WHEN THE RESERVOIR IS PIERCED AND THE GAS CAN EXPAND; AND A GAS CAP DRIVE, WHICH IS A POCKET OF GAS OVER THE FLUID WHICH HAS ENOUGH PRESSURE AND VOLUME TO EXPAND AND DRIVE THE OIL DOWNWARD OR LATERALLY TO A WELL AND THEN TO THE SURFACE.

A WELL SHOULD BE PUMPED AS HARD AS POSSIBLE AND STILL BE ABLE TO FILL THE DOWN HOLE PUMP. THE TERM "PUMPED OFF" IS TAKEN TO MEAN A WELL WHOSE PRODUCING PRESSURE AND VOLUME IS DRAWN DOWN TO ITS LOWEST PRACTICAL VALUE AND STILL SAFELY FILL THE PUMP. OBVIOUSLY, IT IS A VERY COMPLEX SUBJECT, BUT WITH THESE FEW THOUGHTS IN MIND, WE WILL PROCEED TO THE DOWNHOLE DISCUSSION.

A DOWNHOLE PUMP IS FILLED BY THE PRESSURE DIFFERENTIAL OF THE RESERVOIR. WHEN THE RESERVOIR OIL LEVEL AND PRESSURE DROP BELOW ITS CAPACITY TO FILL THE PUMP, A CONDITION OF INCOMPLETE PUMP FILLAGE OCCURS. THIS IS VERY DAMAGING TO THE OIL PUMPING SYSTEM, SEE "FLUID POUND", PAGE 72.

WHAT IS HAPPENING IS THAT WE ARE CREATING AN AIR OR GASEOUS FLUID SPACE BELOW OUR PUMP PLUNGER AND AS WE LOWER OUR PLUNGER TO PICK UP A NEW OIL LOAD, WE ARE MOVING THE PLUNGER DOWNWARDS THROUGH A COMPRESSIBLE GASEOUS FLUID INSTEAD OF A NONCOMPRESSIBLE OIL AND WE IMPACT THE NONCOMPRESSIBLE OIL FLUID LEVEL SOMEWHERE DURING THE DOWN STROKE.

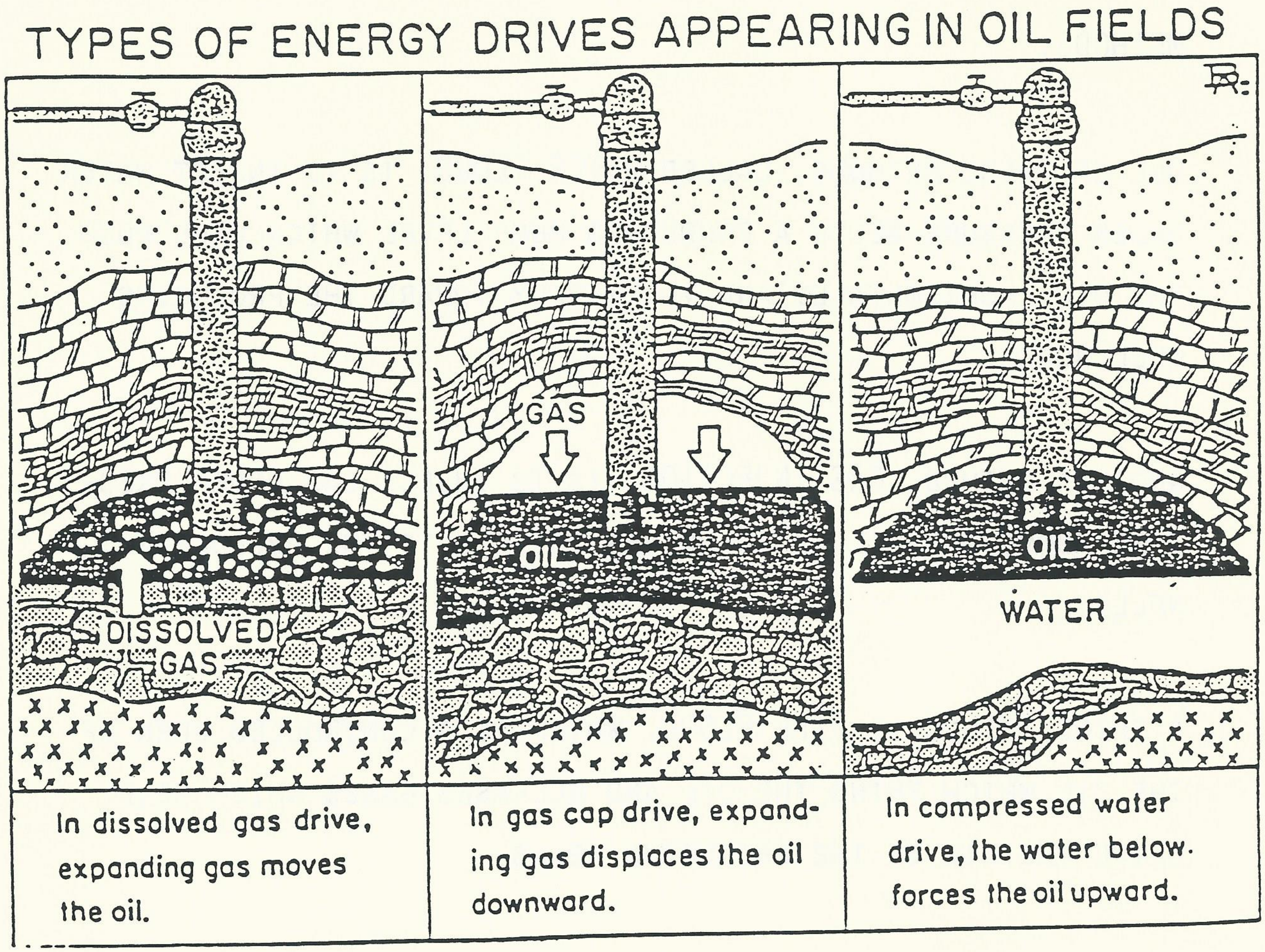
NOW OIL IS AN INCOMPRESSIBLE FLUID AND IF WE HIT IT WITH THE FULL SPEED OF OUR DOWN STROKE, WE CAUSE A SHOCK WAVE WHICH SHOOTS FROM THE PUMP PLUNGER UP THE SUCKER RODS AND INTO THE SURFACE PUMPJACK. THIS SHOCK WAVE IS DAMAGING TO THE DOWN-HOLE PUMP, THE SUCKER RODS, THE TUBING AND THE SURFACE PUMP.

IN THIS CASE (OF FLUID POUND), THE STROKE, STROKES PER MINUTE OR THE PUMP SIZES SHOULD BE CHANGED, OR A TIMER ADDED TO ALLOW THE RESERVOIR PRESSURE TO REFILL THE WELL CASING WITH OIL, MINIMIZING THIS PROBLEM. EVEN SO, OVER A PERIOD OF TIME, THE RESERVOIR PRESSURE CAN BE REDUCED TO A POINT WHERE THE FLUID DOES NOT REPLACE ITSELF FAST ENOUGH TO BE PUMPED ECONOMICALLY. (SOME COMPANIES PUMP AS LOW AS ½ BARREL PER DAY.)

THESE OLD RESERVOIRS CAN BE REJUVENATED AND THE PROCESSES USED ARE USUALLY BROKEN DOWN INTO TWO CATEGORIES:

ENHANCED RECOVERY PROCESSES - WHICH ARE NORMALLY CHEMICAL PROCESSES, AND SECONDARY RECOVERY METHODS.

Fig. 16.1 *Natural pressures that have helped concentrate oil in the ground also help drive it out. (From American Petroleum Institute.)*



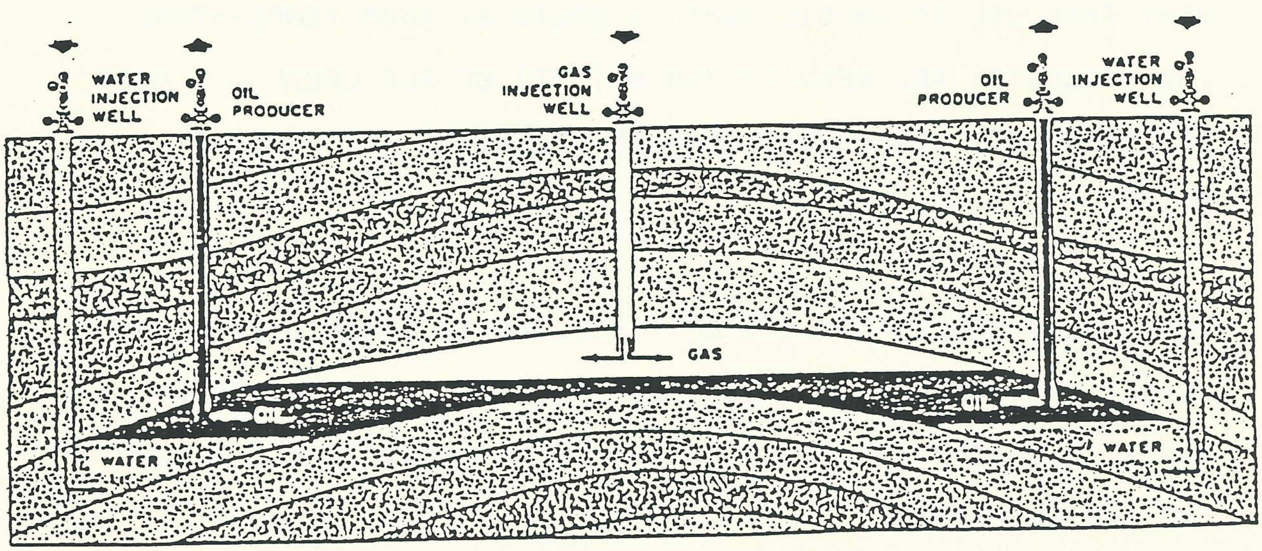


Fig. 16.2 *Gas and water are pumped into the oil-bearing formation, forcing oil toward the well. Usually the gas method is used first; later, water drive is employed. (From American Petroleum Institute.)*

WE WILL DISCUSS, BASICALLY, ONLY THE SECONDARY RECOVERY METHODS.

ONE OF THESE METHODS, CALLED WATER FLOOD, IS TO INJECT WATER UNDER PRESSURE BELOW A PRODUCING PUMP LEVEL WHICH WILL PUSH THE OIL, HORIZONTALLY AND VERTICALLY TOWARD THE PRODUCING WELL BORES.

ANOTHER METHOD, GAS INJECTION, IS TO INJECT GAS UNDER PRESSURE ABOVE THE OIL WHICH DRIVES THE OIL TO THE PRODUCING WELLS.

A THIRD METHOD IS FIRE FLOOD, WHICH IS A CONTROLLED BURN OF THE OIL WHICH THINS THE OIL AND RELEASES GASES WHICH HELP DRIVE THE OIL TO THE PRODUCING WELLS.

THE CRUDE OIL WHICH IS REMOVED FROM THE GROUND RANGES FROM A VERY THIN OIL TO AN OIL THAT IS SOLID AT ROOM TEMPERATURE. THE VISCOSITY AND GRAVITY (OR WEIGHT) OF THE CRUDE OIL MUST BE TAKEN INTO CONSIDERATION WHEN SELECTING THE PUMPING SYSTEM.

VERY HEAVY CRUDE (LOW API GRAVITY) CAN BE THINNED BY STEAM INJECTION, COMBUSTION, POSSIBLY CHEMICAL ADDITIVES OR BY ADDING HIGH GRAVITY CRUDE TO THE POINT WHERE IT WILL FLOW AND CAN BE PUMPED.

I BELIEVE SOME OF THE LAWS FOR SPACING OF WELLS HAS SOMETHING TO DO WITH SHARING THE COST OF GAS INJECTION, WATER FLOOD, ETC., AS WELLS IN SURROUNDING AREA WILL BE AIDED BY THE PROCESS.

WHOEVER IS SELECTING THE PUMPING UNIT SIZE SHOULD GIVE SOME THOUGHT AS TO WHAT WILL HAPPEN TO THE WELL IN THE FUTURE.

THE TWO MAIN METHODS OF DRILLING A WELL IN THE UNITED STATES ARE CABLE TOOL METHOD AND ROTARY DRILLING.

THE CABLE TOOL METHOD CAN BE COMPARED TO THE WAY A ROCK CUTTER DRILLS A HOLE THROUGH A SLAB OF ROCK. THE CHISEL IS HIT WITH A HAMMER WHICH IN TURN CHIPS OUT A HOLE IN THE ROCK.

THE CABLE TOOL DRILLING METHOD IS SIMILAR IN THAT A HEAVY BIT IS SUSPENDED ON THE END OF A CABLE AND THE BIT IS MOVED IN AN UP AND DOWN MOTION TO APPLY THE CHIPPING ACTION TO THE ROCK BEING DRILLED.

THE LOOSE ROCK CHIPS ARE REMOVED BY USING A BAILER ON A WIRE LINE. THIS METHOD OF DRILLING IS QUITE POPULAR IN THE LOW PRESSURE SHALLOW AREAS OF THE APPALACHIANS, BUT THE ROTARY METHOD HAS REPLACED CABLE TOOL DRILLING IN MOST OTHER AREAS IN THE UNITED STATES .

THE DRILLING OF A WELL BY ROTARY METHODS CAN BE COMPARED TO THE USE OF A BRACE AND BIT TO DRILL A HOLE IN WOOD. THREE SEPARATE OPERATIONS MUST BE CARRIED OUT AT THE SAME TIME: THE DRILL STEM AND BIT MUST BE ROTATED, THE BIT MUST BE LOWERED AS THE FORMATION DRILLS OUT FROM UNDER IT, AND THE CUTTINGS MUST BE REMOVED SO THE BIT CAN GET IN CONTACT WITH THE UN-DRILLED PORTION OF THE FORMATION.

THE DRILL STEM FORMS THE LINK BETWEEN THE BIT AND SURFACE. IT IS NOT, HOWEVER, A SOLID, ONE-PIECE SHAFT. AS THE WELL BORE DEEPENS, MORE JOINTS OF DRILL PIPE ARE ADDED AT THE DERRICK FLOOR. THE JOINT IS USUALLY ABOUT 30 FT. IN LENGTH. THIS EVER-LENGTHENING DRILL STRING IS ATTACHED TO THE KELLY AND SWIVEL ROTATED BY THE ROTARY TABLE AND IS THE ROTATING SHAFT WHICH TURNS THE BIT AT THE BOTTOM OF THE HOLE. THE DRILL STEM IN THE HOLE OPERATES AS A STRONG STRING OF STEEL - TWO, THREE, OR EVEN FIVE MILES LONG.

DURING THE DRILLING OPERATION WITH A ROTARY RIG, A MUD PUMP FORCES FLUID MUD DOWN THE CENTER OF THE DRILL PIPE AND DRILL BIT, FLUSHING OUT CUTTINGS, COOLING AND WEIGHTING THE BIT, AND CONTROLLING THE WELL WHEN ENCOUNTERING UNEXPECTED DANGEROUS GAS PRESSURE.

A PIPE, CALLED A WELL CASING, IS INSERTED INTO THE HOLE AND EVENTUALLY CEMENTED IN PLACE. THE CEMENT IS PUMPED DOWN THROUGH THE CENTER OF THE PIPE UNDER PRESSURE AND ALLOWED TO

COME UP AROUND THE OUTSIDE OF CASING TOWARD THE SURFACE, DRIVING OUT THE DRILLING FLUIDS AHEAD TO IT AND FILLING THE GAP BETWEEN THE WELL CASING AND THE DRILLED HOLE.

THE CASING USUALLY IS IN MULTIPLE SECTIONS. A SURFACE CASING WILL USUALLY BE INSTALLED TO 300 FT. OR POSSIBLY UP TO 4000 FT. IN LENGTH WHICH WILL BE CEMENTED IN PLACE. THIS IS TO PREVENT FRESH WATER SANDS FROM BEING CONTAMINATED WITH OIL, GAS, MUD OR SALT WATER.

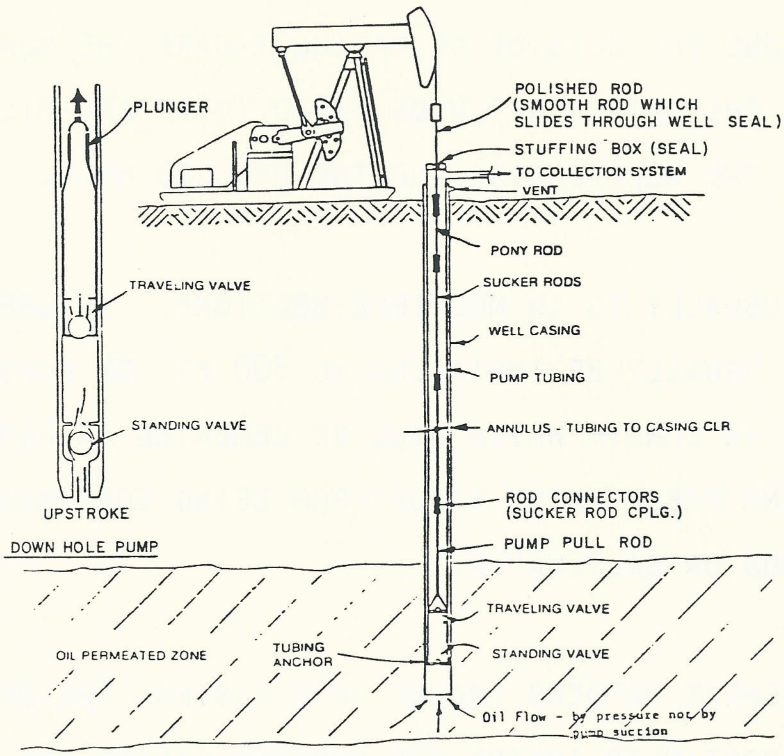
AFTER THE CEMENT HARDENS AROUND THIS CASING, THE DRILLING PROCESS IS CONTINUED INSIDE THE SURFACE CASING AND ANOTHER CASING STRING IS RUN INSIDE THE SURFACE CASING.

WE NOW HAVE THE HOLE DRILLED AND THE HOLE LINED WITH A WELL CASING WHICH IS CEMENTED TO THE ROCK FORMATIONS.

AT THIS TIME, THE CASING IS PERFORATED TO ALLOW THE FLUID TO ENTER THE WELL. THIS IS ACCOMPLISHED BY LOWERING A "GUN" TO THE DESIRED LEVEL AND FIRING BULLETS OR SHAPED CHARGES TO PERFORATE THE CASING. SOMETIMES HYDRAULIC PRESSURE IS USED TO FRACTURE THE ROCK FORMATION AND ACID CAN BE USED TO EAT PASSAGES INTO SOME TYPES OF FORMATIONS.

DOWN THE CENTER OF THIS WELL CASING, WE RUN ANOTHER STRING OF PIPE CALLED THE PRODUCTION TUBING INTO WHICH CAN BE RUN A DOWNHOLE PUMP. THE DOWNHOLE PUMP IS NORMALLY A SIMPLE

WHAT HAPPENS DOWN THE HOLE



1. TRAVELING VALVE CLOSES AT START OF PLUNGER UPSTROKE.

2. RODS PICK UP LOAD GRADUALLY AS THEY STRETCH.

3. WHEN RODS ARE FULLY LOADED (STRETCHED DROPPING PRESSURE IN PUMP), STANDING VALVE OPENS (FROM RESERVOIR PRESSURE), ALLOWING OIL TO ENTER. TRAVELING VALVE PLUNGER LIFTS COLUMN TO SURFACE.

4. ON DOWNSTROKE, RODS SHORTEN AS LOAD IS RELEASED; STANDING VALVE THEN CLOSES (FROM PRESSURE BUILD UP IN PUMP) AND TRAVELING VALVE OPENS (UNANCHORED TUBING STRETCHES AS TRAVELING VALVE OPENS AND LOAD IS TRANSFERRED TO TUBING).

PLUNGER PUMP. THERE ARE MORE THAN ONE KIND, BUT FOR THIS DISCUSSION, WE WILL TALK ONLY ABOUT A SIMPLE DIAGRAM BASIC DESCRIPTION.

THE FOLLOWING IS THE DOWNHOLE PART NOMENCLATURE WHICH WE WILL BE USING IN THIS DISCUSSION, REALIZING IN ACTUALITY THAT WE ARE ONLY SKIMMING THE SURFACE OF THIS TECHNOLOGY.

1. WELL CASING - THE PIPE WHICH WALLS THE DRILLED HOLE AND IS CEMENTED TO THE ROCK STRUCTURE.

2. PUMP TUBING - THE PIPE WHICH IS SET IN THE CENTER OF CASING FROM THE GROUND (WELL HEAD) TO BELOW OIL LEVEL, THROUGH WHICH, THE FLUID PRODUCTION FLOWS.

3. ANNULUS - THE CLEARANCE BETWEEN THE PRODUCTION TUBING AND THE WELL CASING.

4. TUBING ANCHOR - AN ATTACHMENT FOR FASTENING THE PRODUCTION TUBING TO THE WELL CASING TO PREVENT TUBING FROM STRETCHING AND CONTRACTING AS LOAD CHANGES.

5. OIL PERMEATED ZONE - POROUS ROCK OR SANDS IN WHICH THE FLUID IS LOCATED.

6. DOWNHOLE PUMP - THE PUMP LOCATED IN THE TUBING WHICH, WHEN STROKED UP AND DOWN, LIFTS THE OIL.

7. SUCKER RODS - IS A STRING OF STEEL RODS BETWEEN THE SURFACE OF THE GROUND AND IS ATTACHED TO THE DOWNHOLE VALVE ROD.

8. VALVE ROD - IS THE VERY BOTTOM ROD CONNECTING THE SUCKER ROD STRING TO THE PUMP PLUNGER.

9. PONY ROD - THE SECOND AND BOTTOM RODS OF THE SUCKER ROD STRING CONNECTING THE POLISHED ROD AND THE VALVE ROD TO THE SUCKER RODS - NONSTANDARD LENGTH TO ADJUST PUMP PLUNGER SPACING.

10. POLISHED RODS - THE POLISHED ROD IS THE SPECIAL TOP ROD OF THE SUCKER ROD STRING, POLISHED TO SLIDE THROUGH THE ROD SEAL IN THE STUFFING BOX.

11. ROD COUPLINGS - THE SCREWED JOINT WHICH CONNECTS ONE ROD TO ANOTHER, BY MEANS OF A THREADED COLLAR.

12. SEAL - THE TOP OF THE WELL IS SEALED TO THE POLISHED ROD TO MINIMIZE FLUID AND GAS FROM ESCAPING, BY MEANS OF PACKING IN OF A STUFFING BOX.

13. VENT - THE ANNULUS IS VENTED (USUALLY INTO THE FLOW LINE) TO CONTROL GAS PRESSURE WITHIN THE WELL, ALLOWING THE GAS TO BYPASS THE PUMP.

14. STANDING VALVE - A BALL AND SEAT CHECK VALVE IN THE BOTTOM OF THE PUMP TO ALLOW FLUID TO ENTER THE PUMP ON THE UPSTROKE.

15. TRAVELING VALVE - A BALL AND SEAT CHECK VALVE WHICH MOVES UPWARDS WITH THE SUCKER RODS SUPPORTING THE COLUMN OF OIL.

WE WILL ELABORATE ON THESE COMPONENT PARTS AS WE CONTINUE OUR DISCUSSION.

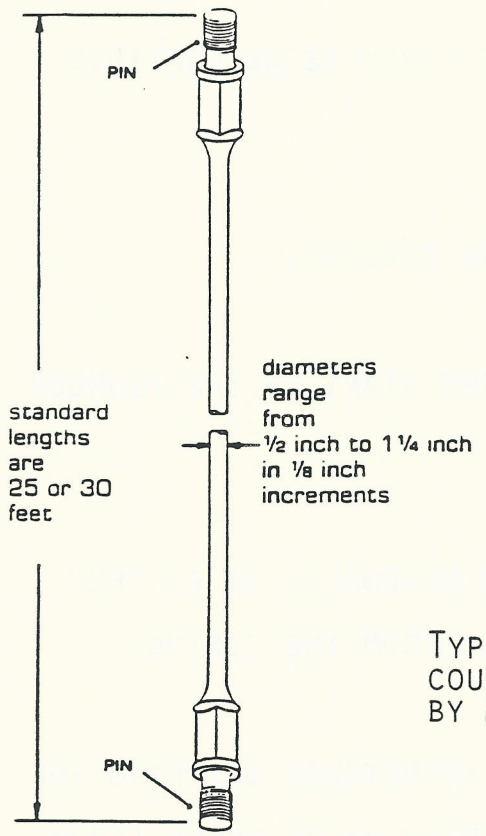
BASICALLY, THE DOWNHOLE PUMP WORKS AS FOLLOWS:

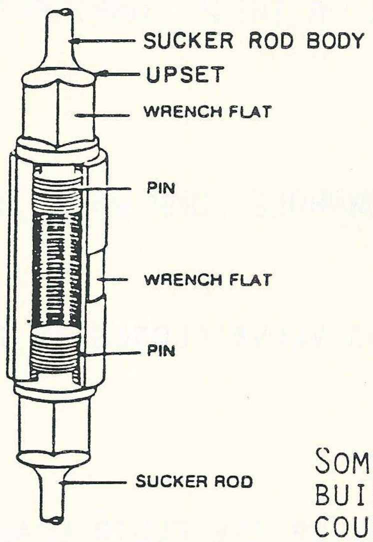
1. THE TRAVELING VALVE CLOSES AT THE START OF THE PLUNGER UPSTROKE.

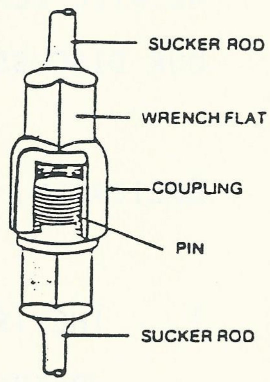
2. THE RODS PICK UP THE FLUID LOAD GRADUALLY AS AS THEY STRETCH, REMOVING THE FLUID LOAD FROM THE TUBING.

3. WHEN THE RODS ARE FULLY LOADED (STRETCHED DROPPING THE PRESSURE IN THE PUMP), THE STANDING VALVE OPENS (FROM THE RESERVOIR PRESSURE), ALLOWING OIL TO ENTER. THE TRAVELING VALVE PLUNGER LIFTS THE COLUMN OF OIL TO THE SURFACE AND RESERVOIR PRESSURE FILLS THE PUMP AS THE PLUNGER RISES.

TYPICAL SUCKER ROD COMPONENTS







SOME RODS HAVE COUPLINGS BUILT INTO ONE END WHEN THE COUPLING FAILS ENTIRE ROD MUST BE REPLACED. (SELDOM USED)

TYPICAL ROD COUPLING JOINT USED BY MOST COMPANIES.

TYPICAL ROD USED BY MOST COMPANIES HAS A TREADED PIN ON BOTH ENDS. IF A COUPLING FAILS, THE RODS, SOMETIMES, DO NOT HAVE TO BE REPLACED.

4. ON DOWNSTROKE, THE RODS SHORTEN AS THE LOAD IS RELEASED, THE STANDING VALVE CLOSES AND THE TRAVELING VALVE OPENS (FROM PRESSURE BUILD UP IN THE PUMP) AND THE LOAD IS TRANSFERRED FROM THE RODS TO THE TUBING.

LET US LEAVE THE DISCUSSION ON THE OPERATION OF THE DOWNHOLE PUMP FOR LATER ON WHEN WE DISCUSS SIZING OF THE PUMPJACK AND CONTINUE ON NOW WITH PART DEFINITIONS.

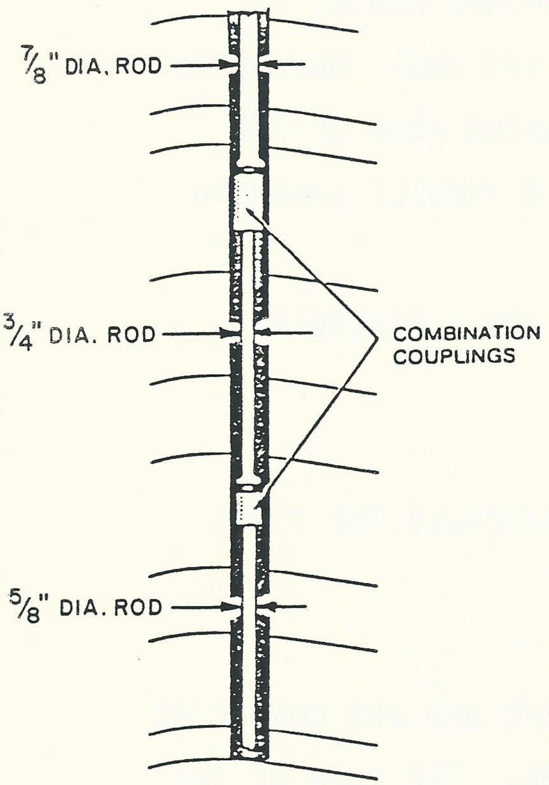
A TYPICAL SUCKER ROD IS 25 OR 30 FT. LONG AND RANGES IN DIAMETER FROM 1/2 INCH TO 1-1/4 INCH IN 1/8 INCH INCREMENTS. IT IS USUALLY MADE FROM ROLLED STEEL RANGING FROM 85,000 MINIMUM TO A 115,000 PSI MINIMUM ULTIMATE TENSILE STRENGTH.

THE EXCEPTIONS ARE SURFACE HARDENED ROD AND A FIBERGLASS ROD.

WE WILL BE LIMITING OUR DISCUSSION TO BASICALLY THE STEEL RODS.

THE RODS ARE NORMALLY THREADED ON EACH END AND ARE CONNECTED TOGETHER BY MEANS OF A THREADED CONNECTOR. THE ENDS OF THE ROD NEAREST THE CONNECTOR ARE SWELLED TO REDUCE STRESS CONCENTRATION AT THE CHANGE IN SECTION FROM THE SMOOTH ROD TO THE THREADED END.

SUCKER ROD SIZE NO. DEFINITION
See API RP11L



EXAMPLE OF
#75 TAPERED
ROD STRING

API ROD NO.	ROD DIAMETERS	API ROD NO.	ROD DIAMETERS
44	all 1/2"	86	1", 7/8", 3/4"
54	5/8, 1/2"	87	1", 7/8"
55	all 5/8"	88	all 1"
64	3/4, 5/8, 1/2"	96	1 1/8", 1", 7/8", 3/4'
65	3/4, 5/8"	97	1 1/8", 1", 7/8"
66	all 3/4"	98	1 1/8, 1"
75	7/8, 3/4, 5/8"	99	all 1 1/8"
76	7/8, 3/4"	107	1 1/4", 1 1/8", 1", 7/8"
77	all 7/8"	108	1-1/4", 1-1/8", 1"
85	1", 7/8", 3/4", 5/8"	109	1 1/4", 1 1/8"

Rod strings can be made either the same size rods or made of different sizes to reduce the weight. The largest sizes are on top and the changes are made at the proper intervals to maintain safe working stress levels.

In wells that are deeper than 2000 ft., the rods are usually designed as a tapered rod string which reduces the weight of the rods and the load on the surface pumping unit. A tapered rod string consists of rods of different diameters. We put the smaller rods at the bottom of the well. As we come up the well, the load is increasing on the rod by the dead weight of the rod itself, so that at certain percentage locations, as we come toward the surface, we increase the diameter of the rod string to match the load increase caused by the rod weight, and maintain fairly consistent stress levels from the downhole pump up to the pumpjack.

The rod strings are identified by numbers like 87; these numbers refer to the largest and smallest rod sizes in eighths of an inch.

An 87 rod string will consist of 8 divided by 8 = 1 in. diameter rods and a 7 divided by 8 = 7/8 in. diameter rods. If you have an API rod number such as 85, not only would the end rods be 1" and 5/8", but all the numbers in between the 8 and the 5 would be used.

All the numbers in between, in other words, would be an 8, 7, 6 and 5 rod string which would consist of 1", 7/8", 3/4" and 5/8" rods, in percentages as spelled out by the API Rod String Design. It is important that you know this because

THE ROD STRING AFFECTS THE STRUCTURAL LOAD ON A PUMPJACK FROM THE VARYING WEIGHTS AND HELPS DETERMINE THE PUMPJACK SIZE.

SOME OF THE EXCEPTIONS TO THE DESIGN ARE FIBERGLASS RODS. FIBERGLASS RODS ARE GOOD IN TENSION AND BAD IN COMPRESSION. THEREFORE, ON THE BOTTOM OF THE WELL DOWN NEAR THE PUMP, THE FIRST ROD (CALLED SINKER BARS), ARE ALWAYS STEEL TO ADD WEIGHT. THE FIBERGLASS RODS ARE UP TOWARD THE SURFACE AND CAN BE TAPERED. THE SINKER BARS ARE SIZED TO ADD ENOUGH WEIGHT TO MAINTAIN A TENSION STRESS IN THE FIBERGLASS.

THE ADVANTAGE OF FIBERGLASS RODS IS DECREASED LOAD ON THE SURFACE PUMP, ALLOWING YOU TO USE IN SOME CASES A SMALLER PUMP, LESS ENERGY, AND THEY ARE CORROSION RESISTANT.

THE DISADVANTAGE TO THE FIBERGLASS RODS IS THAT IT HAS AN INCREASED STRETCH WHICH CAUSES YOU TO USE A LONGER STROKE PUMPING UNIT. ADDITIONALLY, IF YOU HAD SEVERE FLUID POUND, HIT THE BOTTOM HARD ENOUGH, OR HAD SOME OTHER PROBLEM WHICH CAUSED THE PUMP TO HANG UP IN THE WELL, WHEN THE LOAD ON THE SUCKER RODS IS REVERSED DURING THE DOWNSTROKE, SINCE THEY ARE NOT NEARLY AS GOOD IN COMPRESSION AS THEY ARE IN TENSION YOU COULD HAVE PREMATURE ROD FAILURES.

SUCKER ROD FAILURES NORMALLY OCCUR AT THE JOINTED CONNECTION OF THE RODS AND MOSTLY ARE DUE TO IMPROPER ROD MAKE-UP. THE

RODS SHOULD BE SCREWED TOGETHER WITH ENOUGH TORQUE TO CREATE A CLAMPING FORCE GREATER THAN THE FORCE THE RODS WILL SEE IN OPERATION.

THIS CAN BE ACCOMPLISHED USING A POWER TONG WHICH USES MEASURED TORQUES FOR EACH SIZE ROD OR A METHOD SIMILAR TO THE "TURN OF THE NUT METHOD" USED IN TIGHTENING STRUCTURAL BOLTS.

THE JOINT IS TURNED A MEASURED DISTANCE FROM A HAND-TIGHT CONDITION WHICH PRELOADS THE JOINT.

BOTH METHODS ARE DESCRIBED IN THE API SPECIFICATIONS.

DURING OPERATION OF THE PUMPING SYSTEM IT IS ALMOST IMPOSSIBLE TO PREVENT THE SUCKER ROD COUPLINGS FROM WEARING AGAINST THE PRODUCTION TUBING THROUGH THE LENGTH OF THE STROKE.

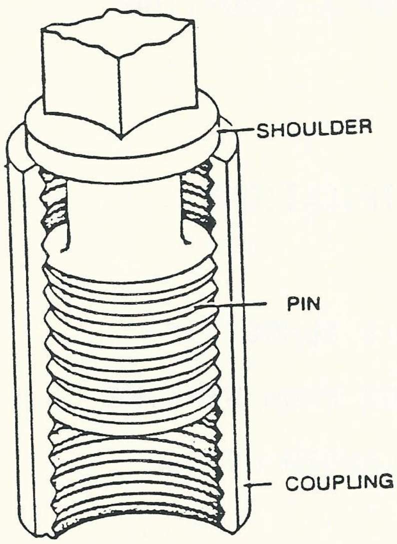
CONSEQUENTLY, NEAR THE ROD JOINT COUPLINGS, DIFFERENT TYPES OF GUIDES CAN BE ATTACHED TO THE RODS TO SPACE THE COUPLING WEAR FROM THE TUBING AND PREVENT WEAR ON THE COUPLING. AS MOST FAILURES OCCUR IN THE COUPLING JOINT, IT IS IMPORTANT TO PREVENT WEAR.

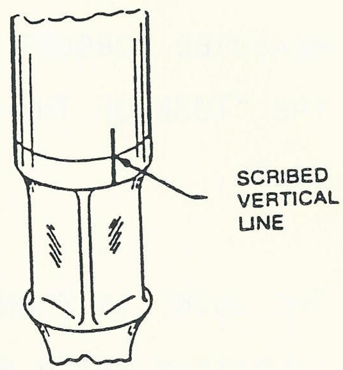
ANOTHER WAY TO INCREASE THE TUBING LIFE FROM WEAR IS, AFTER A PERIOD OF TIME, TO CHANGE THE LENGTH OF THE PONY RODS AT

SUCKER ROD JOINTS

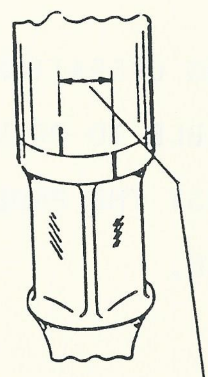
SEE API SPECIFICATION No. RP-11BR

Sucker rod joints are torqued to give a preload stress higher then the operating stress





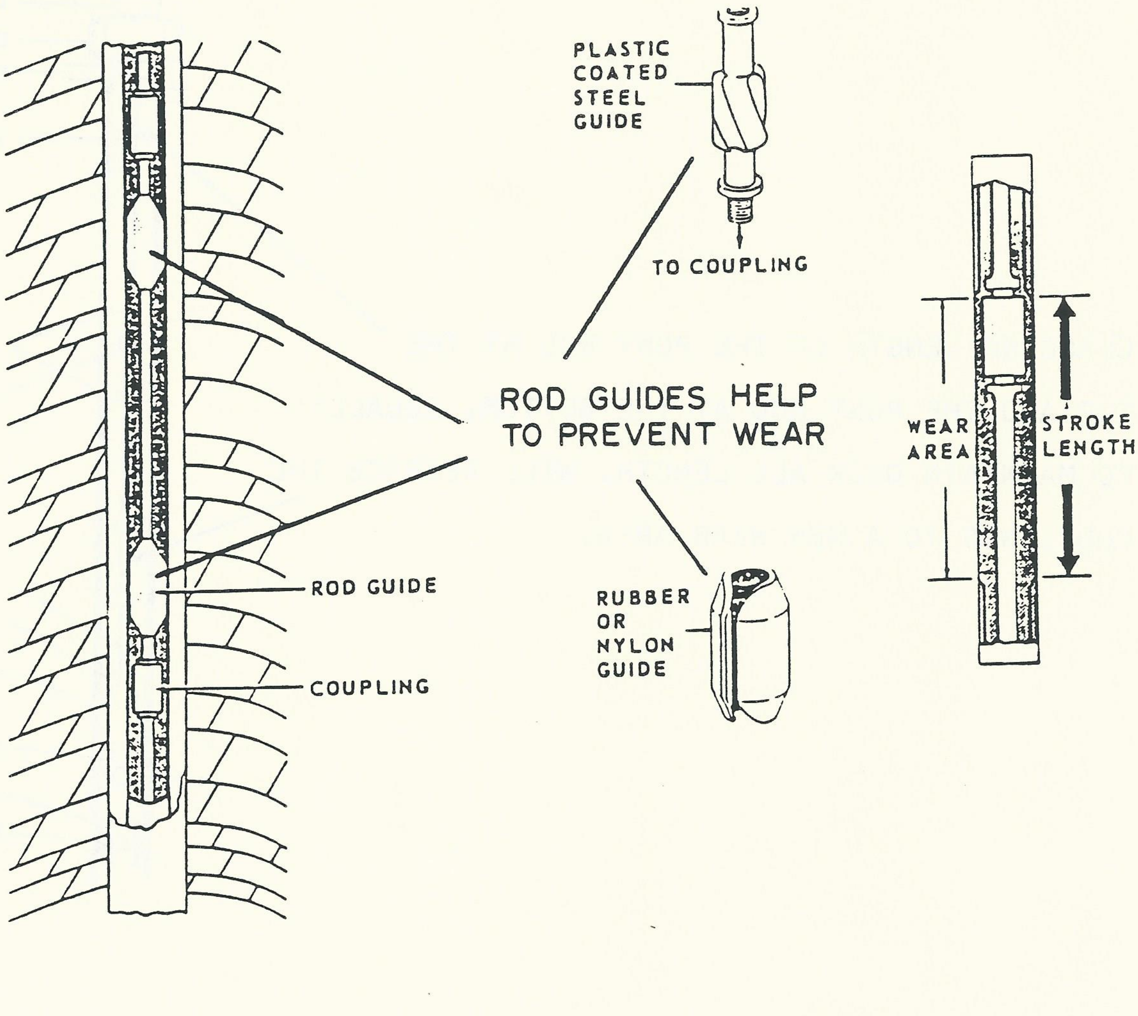
Initial setting to be hand tight



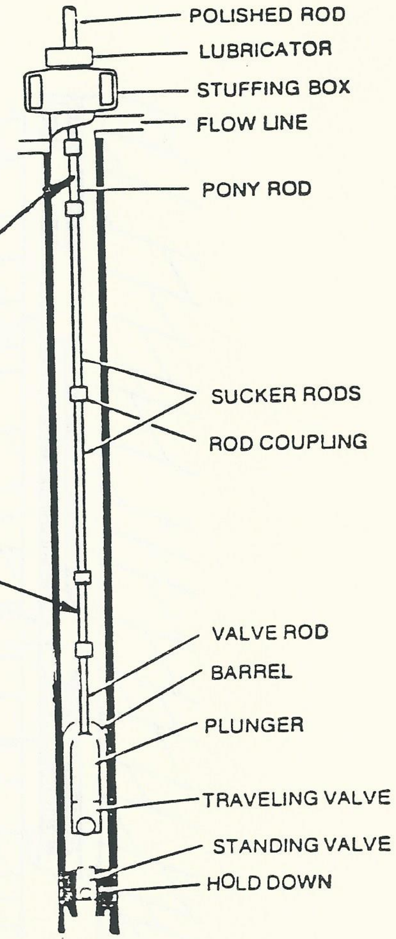
Turn from hand tight the recommended amount for the rod size.

NOTE: over one half of the sucker rod failures occur at the joint and are usually caused by improper joint make up

SUCKER ROD COUPLINGS WEAR AGAINST THE PRODUCTION TUBING THROUGH THE LENGTH OF THE STROKE.



RESPACING ROD COUPLINGS FOR A NEW TUBING WEAR PATTERN



CHANGING LENGTH OF THE PONY ROD AT THE TOP AND THE PONY ROD AT THE BOTTOM, EQUALLY TO MAINTAIN OVER ALL LENGTH, WILL RESPACE THE COUPLINGS TO A NEW WEAR AREA.

THE TOP AND BOTTOM OF THE SUCKER RODS EQUALLY TO MAINTAIN OVERALL LENGTH SO AS NOT AFFECT THE PLUNGER SPACING IN THE PUMP. THIS WILL RESPACE THE COUPLING WEAR TO A DIFFERENT LOCATION ON THE PRODUCTION TUBE.

SOME WELLS ARE SUBJECT TO PARAFFIN BUILD-UP. THIS BUILD-UP CAN REDUCE THE AMOUNT OF FLUID FLOW AND CREATE UNNEEDED LOAD ON THE SUCKER RODS AND THE PUMPING SYSTEM. CONSEQUENTLY, SOMETIMES, PARAFFIN SCRAPERS ARE ADDED TO THE RODS TO MINIMIZE THE PARAFFIN BUILD-UP ON THE WALLS OF THE TUBING. SOME SCRAPERS ARE SPIRAL AND DO NOT NEED TO BE ROTATED.

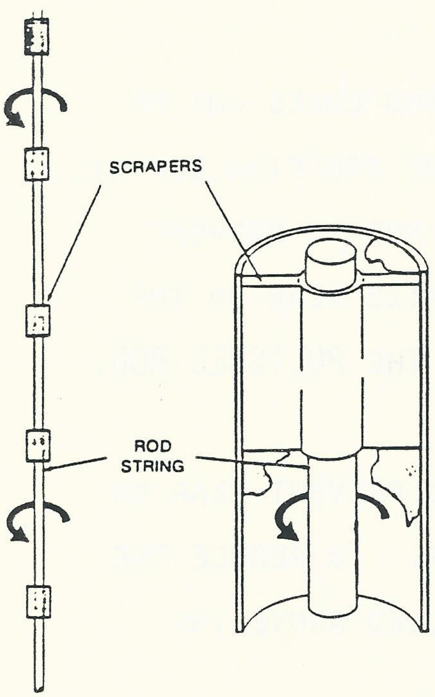
BOTH THE PARAFFIN BUILD-UP AND THE TUBING SCRAPERS NEED TO BE ACCOUNTED FOR IN THE SELECTION OF THE PUMPING UNIT.

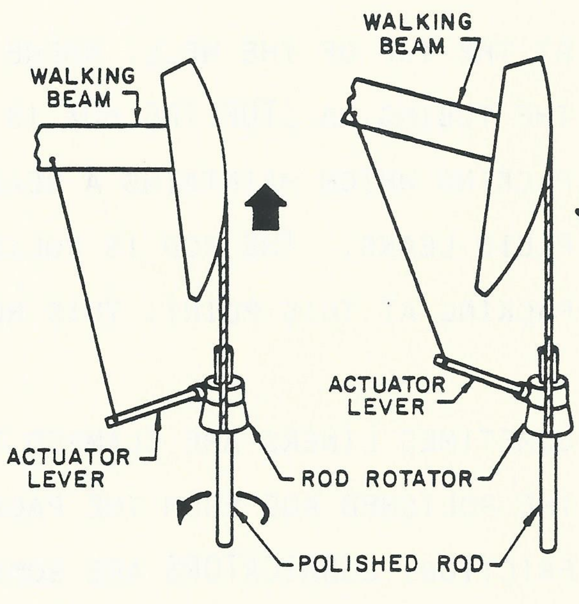
AT THE TOP OF THE WELL, WHERE THE SUCKER ROD COMES OUT OF THE TUBING, A STUFFING BOX IS LOCATED. THE STUFFING BOX HAS PACKING WHICH MAINTAINS A SEAL AROUND THE ROD TO PREVENT FLUID LEAKS. THE ROD IS POLISHED TO MINIMIZE WEAR ON THE PACKING AT THIS POINT; THIS ROD IS CALLED THE POLISHED ROD.

SOMETIMES LINERS ARE CLAMPED TO THE ROD TO PREVENT WEAR ON THE POLISHED ROD FROM THE PACKING FRICTION. TO REDUCE THE FRICTION, LUBRICATORS ARE SOMETIMES INSTALLED ABOVE THE STUFFING BOX.

TUBING SCRAPERS

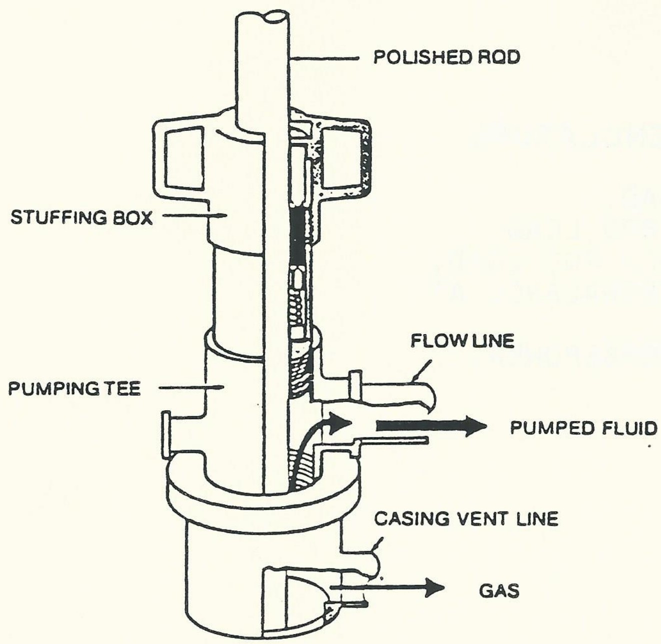
- Scrapers attached to the rods are used to prevent paraffin build-up.
- Flat Scrapers need to be rotated.
- A system of levers and gears turns the rods slightly.
- Usually spiral scrapers do not need to be rotated.

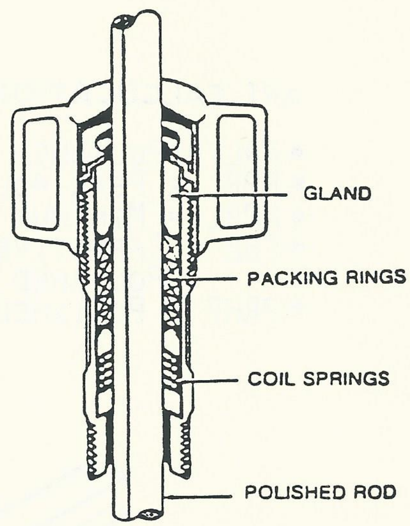


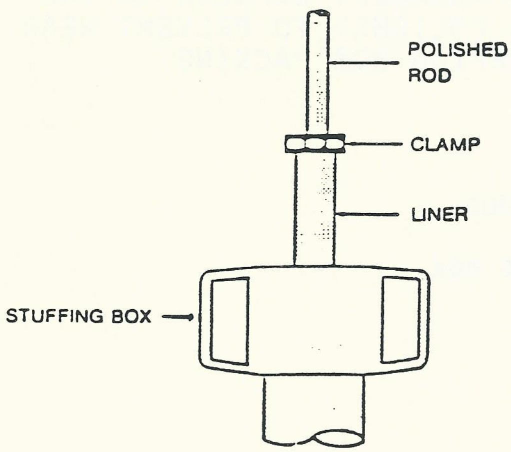


SEALING OF THE WELL

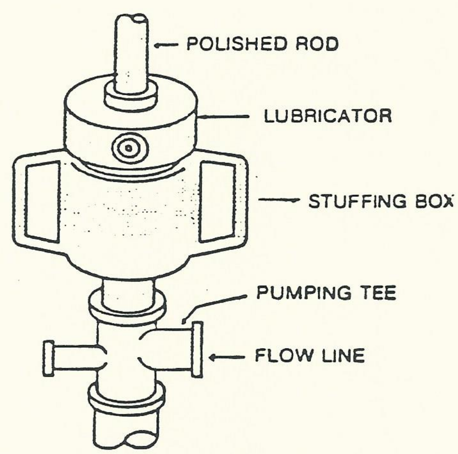
A stuffing box is installed above the well tubing to prevent fluid losses around the polished rod.







Sometimes liners are clamped to the polished rod to prevent wear on the rod from the packing friction.

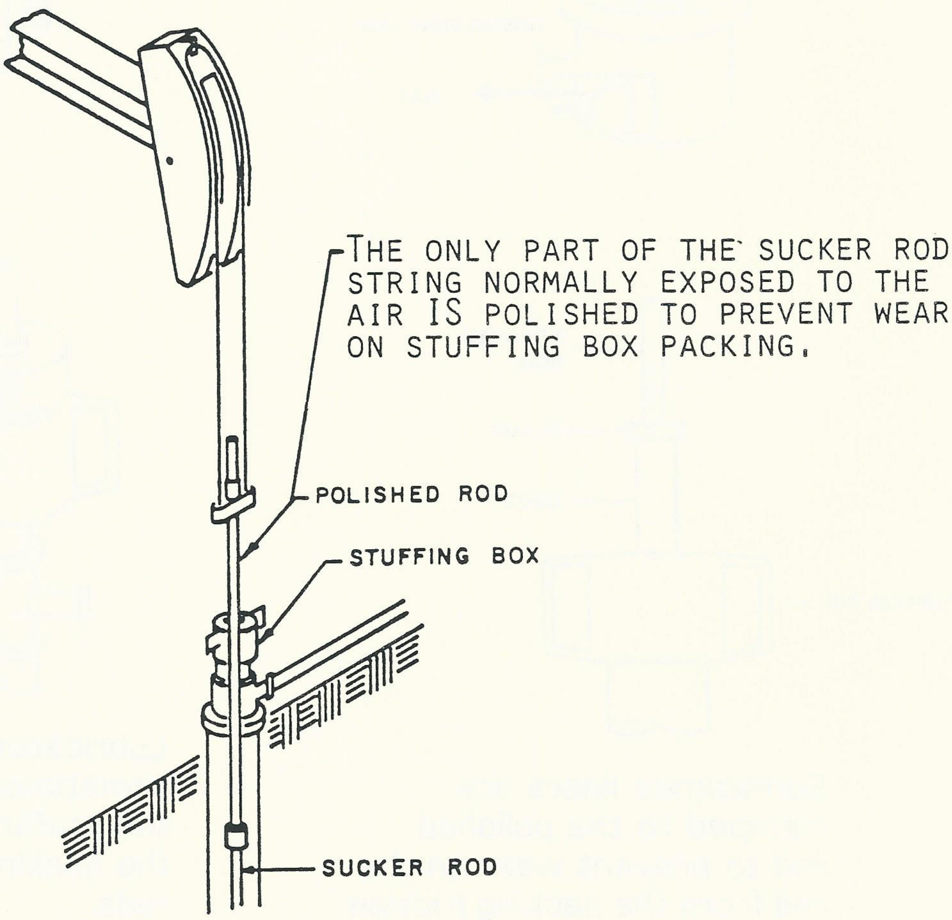


Lubricators are sometimes installed above the stuffing box to decrease the packing friction on the rods.

THE API CALCULATIONS, AS FAR AS LOADING IS CONCERNED, REFER TO LOADS AT THE POLISHED ROD WHICH IS THE TOP ROD OF THE SUCKER ROD STRING AND IS POLISHED TO MINIMIZE WEAR ON THE STUFFING BOX PACKING.

API CALCULATION NOMENCLATURE

- PRL = POLISHED ROD LOAD.
- PPRL = PEAK POLISHED ROD LOAD.
- MPRL = MINIMUM POLISHED ROD LOAD.
- CBE = EFFECTIVE COUNTERBALANCE AT POLISHED ROD.
- PRHP = POLISHED ROD HORSEPOWER.



THE PACKING CAN BE ADJUSTED OR PRESSURED AT THE POLISHED ROD BY TIGHTENING OR LOOSENING THE PACKING GLAND NUT. TOO TIGHT PACKING CAN INCREASE THE GEARBOX LOAD, REQUIRED POLISHED ROD HORSEPOWER AND DECREASE THE FREE-FALL SPEED OF THE RODS AND INCREASE POWER CONSUMPTION.

LET US DISCUSS THE DIFFERENT TYPES OF COUNTERBALANCED BEAM PUMPING UNITS.

THERE ARE ACTUALLY THREE OR FOUR DIFFERENT TYPES OF PUMPING UNITS AND THERE IS AN ADDITIONAL STACK OF U.S. PATENTS ON SURFACE PUMPING UNITS APPROXIMATELY A FOOT THICK, BUT THE UNIT THAT SURVIVED OVER MANY YEARS IS THE API CONVENTIONAL BEAM PUMPING UNIT, SOMETIMES REFERRED TO AS A PUMPJACK.

THE PURPOSE OF A PUMPJACK IS TO PROVIDE AN UP AND DOWN MOTION TO THE SUCKER ROD STRING TO OPERATE THE DOWNHOLE PUMP.

THE API BEAM UNIT CAN BE DIVIDED INTO THREE CLASSES:

CONVENTIONAL CLASS 1 LEVER SYSTEM
FRONT MOUNTED CLASS 3 LEVER SYSTEM
AIR BALANCED UNIT CLASS 3 LEVER SYSTEM

THE PRINCIPAL DIFFERENCE BETWEEN THE CLASS 3 LEVER SYSTEM AND THE CONVENTIONAL UNIT IS THE SPEED OF THE UPSTROKE AND

DOWNSTROKE. THE CONVENTIONAL UNIT ACCELERATES FASTER AND HAS A FASTER UPSTROKE THAN THE CLASS 3 LEVER SYSTEM. THE CLASS 3 SYSTEM HAS A FASTER DOWNSTROKE THAN THE CONVENTIONAL UNIT. BOTH UNITS START AT THE BOTTOM OF THE STROKE SIMULTANEOUSLY AND END UP TOGETHER. BUT THE DIFFERENT SPEEDS MAKE THESE UNITS HAVE DIFFERENT CHARACTERISTICS WHICH MAKES ONE PREFERRED OVER THE OTHER, ON DIFFERENT WELL PARAMETERS.

ON HEAVY PUMP LOADS, SHORT STROKES, THE CONVENTIONAL UNIT IS PREFERRED. ON LIGHT PUMP LOADS, LONG STROKES, THE CLASS 3 IS PREFERRED. WELLS WITH FLUID OR GAS POUND WOULD POSSIBLY PREFER THE CONVENTIONAL UNIT.

THE CLASS 3 LEVER SYSTEM CAN ALSO BE A CRANK BALANCE SYSTEM. IHE CLASS 3 GEOMETRY CAN ALSO BE COUNTERBALANCED BY THE USE OF AIR CYLINDERS WHICH REDUCE THE TOTAL UNIT WEIGHT.

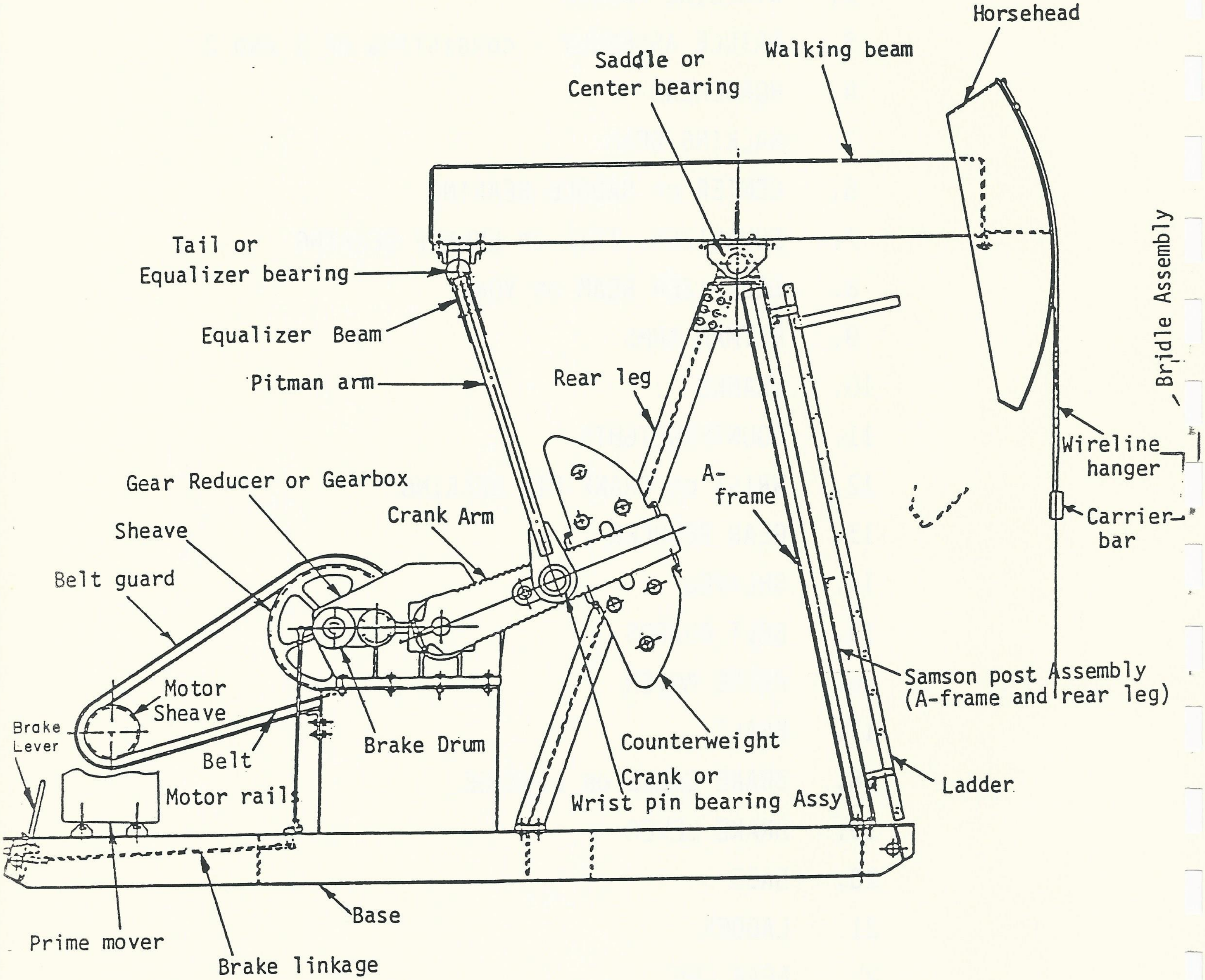
IN ADDITION TO THESE API UNITS, THERE ARE MANY OTHER DESIGNS, SUCH AS HYDRAULIC WHICH CAN BE NOTHING MORE COMPLICATED THAN A HYDRAULIC CYLINDER MOUNTED OVER THE RODS THAT LIFTS THE RODS UP AND DOWN.

THE RODS CAN ALSO BE MOVED BY MEANS OF WIRE ROPE, SHEAVES AND WINCHES. THERE ARE MANY TYPES OF NEW AND EXPERIMENTAL UNITS BEING DEVELOPED AND PATENTS APPLIED FOR WHICH STROKE THE DOWNHOLE PUMP.

THE NOMENCLATURE FOR THE CRANK BALANCED PUMPING UNIT IS AS FOLLOWS:

1. CARRIER BAR
2. WIRELINE HANGER
3. BRIDLE ASSEMBLY - CONSISTING OF 1 AND 2
4. HORSEHEAD
5. WALKING BEAM
6. CENTER OR SADDLE BEARING
7. EQUALIZER, TAIL OR EVENER BEARING
8. EQUALIZER BEAM OR YOKE
9. PITMAN ARMS
10. CRANKS
11. COUNTERWEIGHTS
12. WRIST OR CRANK PIN BEARING
13. GEAR REDUCER
14. SHEAVES
15. BELT GUARDS
16. PRIME MOVER
17. BRAKE
18. BRAKE CABLE OR LINKAGE
19. BRAKE LEVER
20. BASE
21. LADDER
22. REAR LEG
23. A-FRAME
24. SAMSON POST ASSEMBLY - CONSISTING OF 22 AND 23

STANDARD PUMPING UNIT NOMENCLATURE



FOR THE PURPOSE OF THIS DISCUSSION TODAY, WE ARE PRINCIPALLY TALKING ABOUT ONLY THE CLASS 1 LEVER SYSTEM, KNOWN AS THE PUMPJACK OR AS A COUNTERBALANCED PUMPING UNIT.

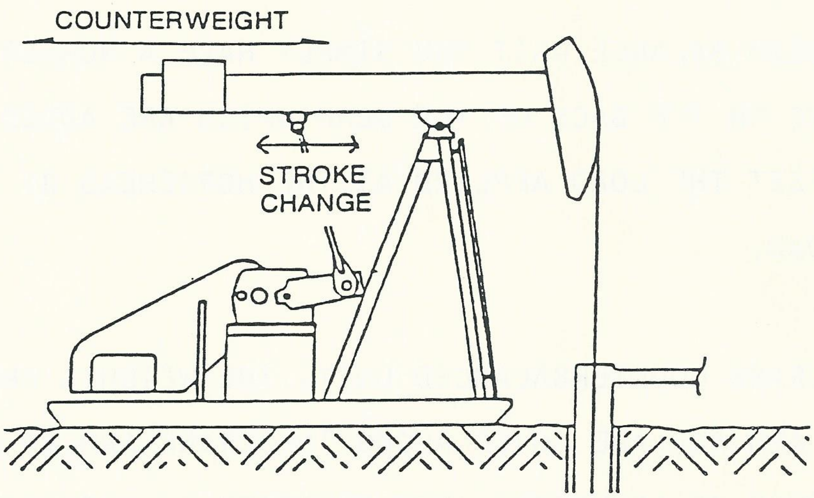
THE CONVENTIONAL PUMPING UNIT OF THE CLASS 1 LEVER SYSTEM CAN BE BROKEN DOWN INTO BEAM BALANCE UNITS AND CRANK BALANCE UNITS.

ON A BEAM BALANCE UNIT YOU SIMPLY HAVE A NUMBER OF COUNTER-WEIGHTS ON THE BACK OF THE BEAM WHICH ARE ADDED OR REMOVED TO OFFSET THE LOAD APPLIED AT THE HORSEHEAD BY THE SUCKER ROD LOAD.

ON A CRANK COUNTERBALANCED UNIT, THE WEIGHTS ARE ON THE CRANK ARM AND ARE MOVED ALONG THE CRANK ARM TO OBTAIN THE SAME TYPE OF COUNTERBALANCE FUNCTION AS OBTAINED ON THE BEAM BALANCE UNIT. THERE ARE SOME ADVANTAGES TO THE ROTARY COUN-COUNTERWEIGHTS IN THAT THEY ARE APPLIED HARMONIOUSLY WITHOUT SHOCK. ON THE CRANK BALANCE, THERE ARE USUALLY THREE WRIST PIN POSITIONS. BY MOVING THE WRIST PIN FROM ONE HOLE TO ANOTHER, WE CHANGE THE LENGTH OF THE STROKE. USUALLY, ON THE BEAM BALANCE UNIT, THE LENGTH OF THE STROKE IS CHANGED BY CHANGING THE POSITION OF THE TAIL BEARING OR SADDLE BEARING.

WHEN YOU HEAR THE EXPRESSION "AMOUNT OF COUNTERBALANCE" - THEY ARE NOT TALKING ABOUT THE ACTUAL WEIGHT OF THE COUNTER-

CONVENTIONAL PUMPING UNIT
BEAM BALANCED



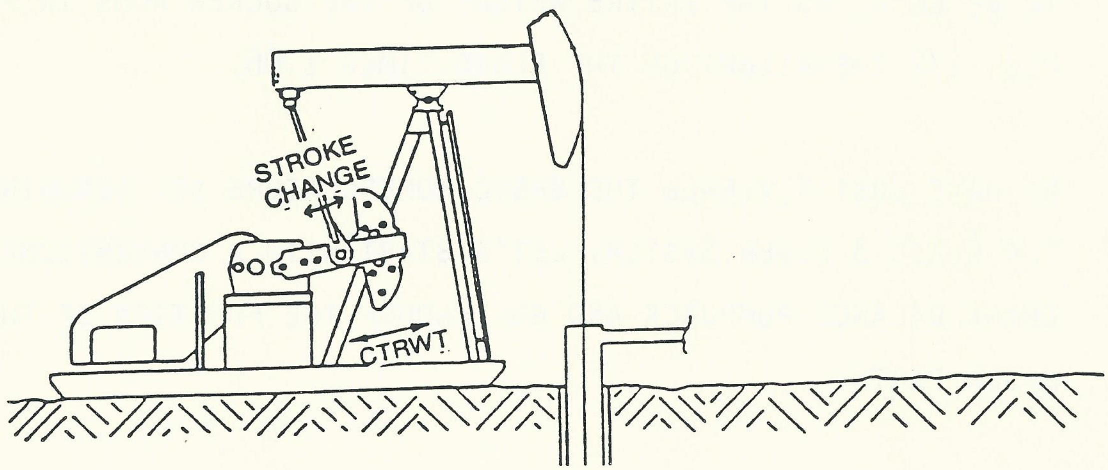
1. COUNTERWEIGHTS ARE ADJUSTED BY ADDING OR MOVING WEIGHTS ON THE WALKING BEAM.

2. STROKE CAN BE CHANGED BY MOVING TAIL BEARING LOCATION.

3. ADDITIONAL ENERGY IS REQUIRED TO STOP AND REVERSE THE OSCILLATING MOTION OF THE BEAM WHEN THE COUNTERWEIGHT IS ON THE BEAM.

4. PITMANS ARE IN TENSION ON THE UPSTROKE, AND IN COMPRESSION ON THE DOWN STROKE.

STANDARD PUMPING UNIT
CRANK BALANCE



1. ROTARY COUNTERBALANCE IS ADJUSTED BY MOVING WEIGHTS IN DIRECTION OF ARROWS.

2. NOTE: THERE ARE 3 CRANK PIN POSITIONS, GIVING 3 DIFFERENT STROKES PER UNIT SIZE.

3. ROTARY COUNTERWEIGHTS ARE APPLIED HARMONICALLY WITHOUT SHOCK, ROTARY MOTION GIVES UP ENERGY TO THE SYSTEM.

4. PITMANS ARE ALWAYS IN TENSION.

WEIGHTS, BUT THEY ARE TALKING ABOUT EFFECTIVE COUNTERBALANCE AT THE POLISHED ROD IN THE APPROXIMATE MIDDLE OF THE UPSTROKE.

THE COUNTERBALANCE EFFECT AT THE WELL IS USUALLY CALCULATED TO BE EQUAL TO THE ENTIRE WEIGHT OF THE SUCKER RODS IN FLUID PLUS 1/2 THE WEIGHT OF THE FLUID TIMES 1.06.

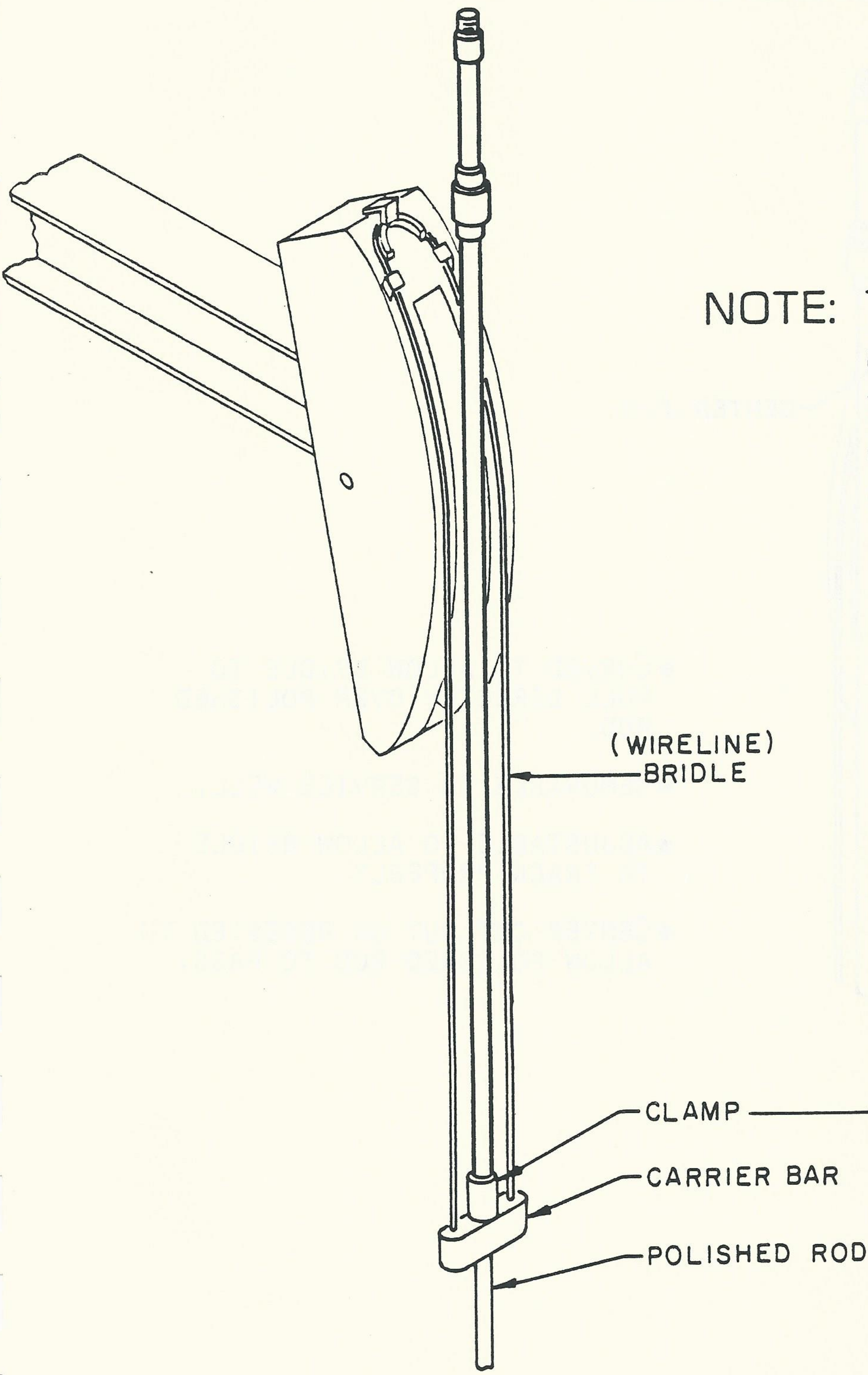
WE HAVE JUST REVIEWED THE BASIC NOMENCLATURE SO, IGNORING THE CLASS 3 LEVER SYSTEM, LET'S START WITH A CONVENTIONAL CRANK BALANCE PUMPJACK AND GO THROUGH THE FUNCTION OF THE MAJOR PARTS.

IF WE START AT THE WELL, THE PART OF THE SUCKER ROD STICKING OUT OF THE GROUND IS CALLED THE POLISHED ROD. IT IS CALLED THE POLISHED ROD DUE TO THE SURFACE FINISH OF THE ROD WHICH MUST BE SMOOTH AS IT SLIDES THROUGH THE PACKING GLAND THAT SEALS OFF THE OIL WELL TUBING.

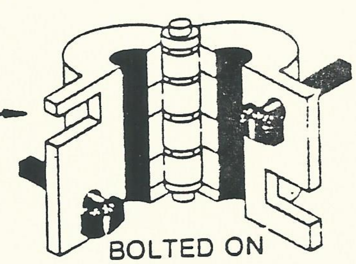
THIS SUCKER ROD IS ATTACHED TO THE WIRELINE HANGER BY MEANS OF A CLAMP WHICH BOLTS TO THE ROD AND SITS ON THE CARRIER BAR. THE WIRELINE HANGER CABLE LOOP FITS OVER AN ANCHOR ON THE HORSEHEAD. ADJUSTING THE LOCATION OF THE CLAMP SPACES THE DOWNHOLE PUMP PLUNGER.

IT IS IMPORTANT TO ALIGN THE HORSEHEAD DIRECTLY OVER THE CENTERLINE OF THE WELL. AS THE HORSEHEAD IS MOVED UP AND

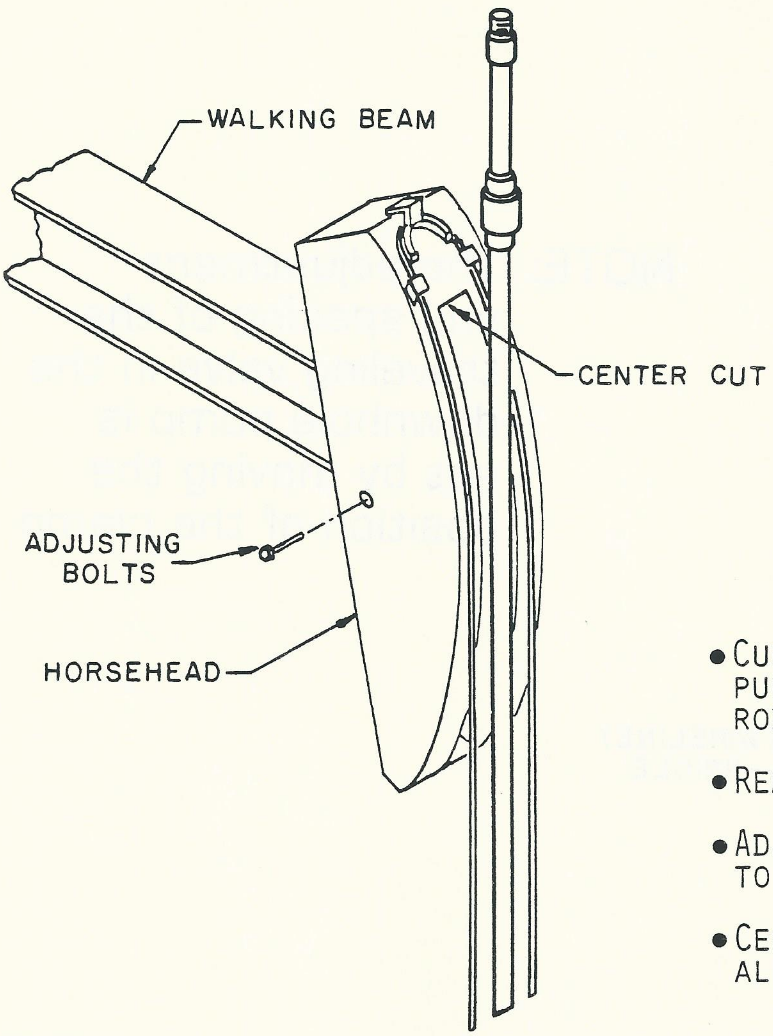
PART NOMENCLATURE
BRIDLE ASSEMBLY



NOTE: The adjustment and spacing of the traveling valve in the downhole pump is set by moving the position of the clamp.



HORSEHEAD



- CURVED TO ALLOW BRIDLE TO PULL DIRECTLY OVER POLISHED ROD.
- REMOVABLE TO SERVICE WELL.
- ADJUSTABLE TO ALLOW BRIDLE TO TRACK PROPERLY.
- CENTER CUT OUT OR RECESSED TO ALLOW POLISHED ROD TO PASS.

DOWN IN AN ARC ABOUT THE CENTER BEARING, THE WIRELINE HANGER FROM THE HORSEHEAD TO THE POLISHED ROD SHOULD REMAIN VERTICAL. IF THE HORSEHEAD DID NOT MOVE IN A TRUE ARC (WITHIN REASON) ABOVE THE POLISHED ROD, IT WOULD HAVE A TENDENCY TO PULL THE POLISHED ROD SIDEWAYS, EXERTING A SIDE FORCE ON THE PACKING GLAND AND WEAR OUT THE PACKING GLAND AND POLISHED ROD.

WHEN A PROBLEM DEVELOPS IN A DOWNHOLE SYSTEM, A WORK-OVER RIG PUMPING UNIT MUST BE MOVED DIRECTLY OVER THE WELL TO CORRECT THE PROBLEM. THEREFORE, THE HORSEHEAD MUST BE MOVED OUT OF THE WAY. IT CAN BE LIFTED OFF AND SET ON THE GROUND, TILTED SIDEWAYS OR VERTICALLY TO BE OUT OF THE WAY. THE HORSEHEAD SHOULD BE ADJUSTABLE TO ALLOW THE WIRELINE TO TRACK PROPERLY AND CENTER CUT TO CLEAR THE POLISHED ROD, WHICH CAN EXTEND HIGH IN THE AIR, PASSING THROUGH THE HORSEHEAD FRONT PLATE.

THE WALKING BEAM IS SIZED DIRECTLY FROM API SPECIFICATIONS. IT HAS 2 LIMITS OF STRESS: ONE IS BUCKLING LIMIT; ANOTHER IS FATIGUE LIFE LIMIT. IT PIVOTS ABOUT THE CENTER BEARING IN A SEE-SAW MOTION MOVING THE HORSEHEAD UP AND DOWN.

SADDLE OR CENTER BEARING IS SIMPLY A SUPPORT FOR THE WALKING BEAM, AND IS NORMALLY GREASE-LUBRICATED, ANTI-FRICTION BEARINGS; ALTHOUGH OTHER BEARINGS CAN BE USED. A SADDLE

BEARING SOMETIMES HAS SOME ADJUSTMENT TO ALIGN THE WALKING BEAM TO THE GEAR REDUCER AND THE WELL.

THE SADDLE BEARING IS SUPPORTED BY A STRUCTURAL ASSEMBLY CALLED A SAMSON POST. SOMETIMES, THE FRONT TWO LEGS ARE CALLED THE A-FRAME AND THE BACK FRAME IS CALLED THE REAR LEG.

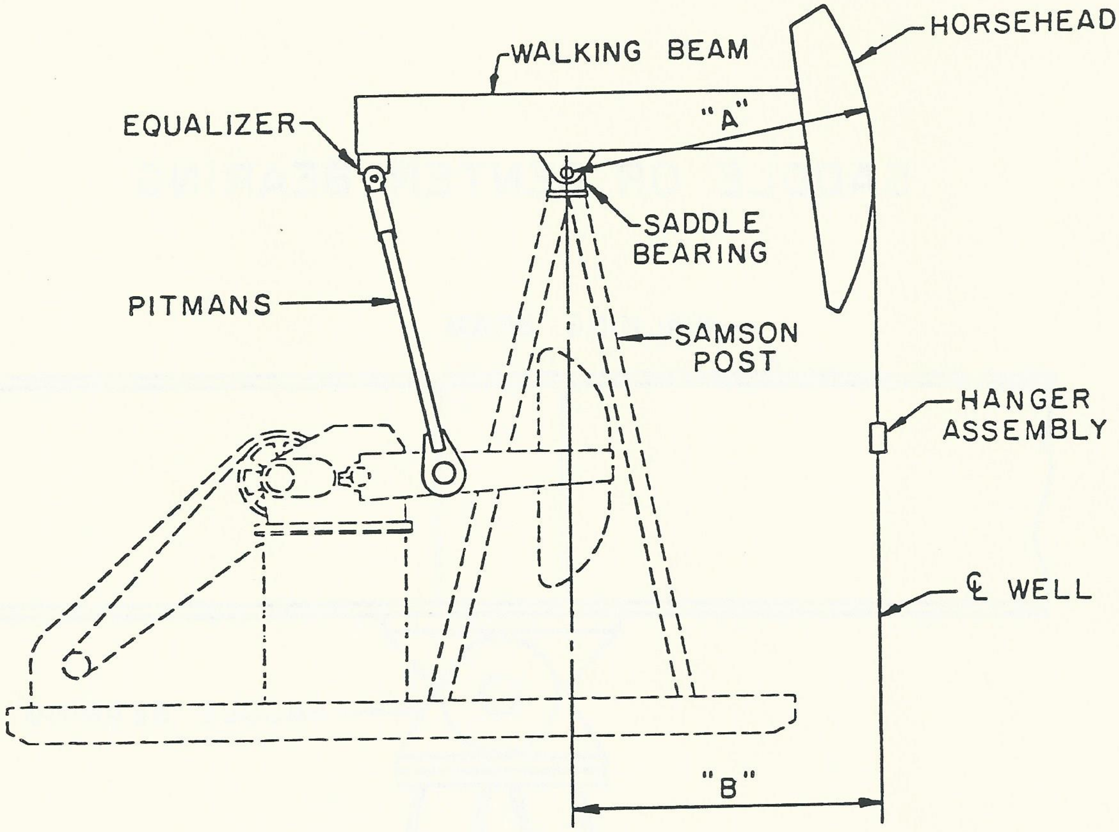
THE EQUALIZER ASSEMBLY IS THE CONNECTION AT THE BACK OF THE BEAM WHICH TRANSMITS THE FORCE TO THE BEAM TO PULL THE BACK END DOWN AND TO LIFT THE FRONT END IN EXACTLY THE SAME PRIN-PRINCIPLE AS A TEETER-TOTTER. IT EQUALIZES THE LOAD IN THE TWO PITMAN ARMS.

THE EQUALIZER ASSEMBLY CONSISTS OF AN EQUALIZER BEAM AT-TACHED TO THE EQUALIZER (OR TAIL) BEARING WHICH IN TURN IS ATTACHED TO THE WALKING BEAM.

THE EQUALIZER BEAM IS CONNECTED TO THE CRANK ARMS BY MEANS OF PITMAN ARMS AND WRIST PIN ASSEMBLIES.

AS THE GEAR BOXES HAVE A DOUBLE OUTPUT SHAFT AND THEY HAVE TWO CRANK ARMS (ONE ON EACH SHAFT EXTENSION), THERE MUST BE A CONNECTION BETWEEN THESE TWO CRANK ARMS AND THE EQUALIZER BEARING WHICH ALLOWS THE LOADS ON EACH CRANK ARM TO BE THE SAME.

WALKING BEAM



- THE WALKING BEAM ROCKS OR PIVOTS ABOUT THE SADDLE BEARING IN A SEE-SAW MOTION.

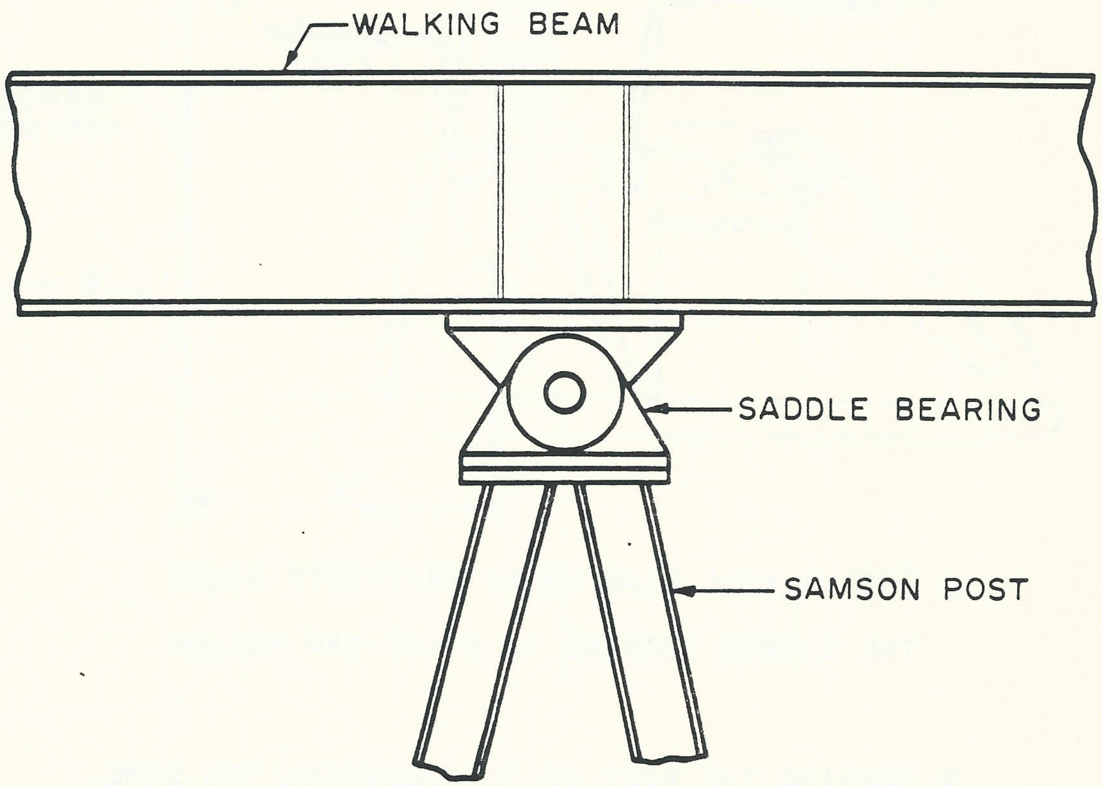
- KEEPING THE REST OF THE GEOMETRY THE SAME, INCREASING DIMENSION "A" INCREASES THE STROKE.

- KEEPING THE REST OF THE GEOMETRY THE SAME, DECREASING DIMENSION "A" DECREASES THE STROKE.

- WHEN DIMENSION "A" IS CHANGED, DISTANCE FROM WELL "B" AND HORSEHEAD CURVE ARE CHANGED ACCORDINGLY.

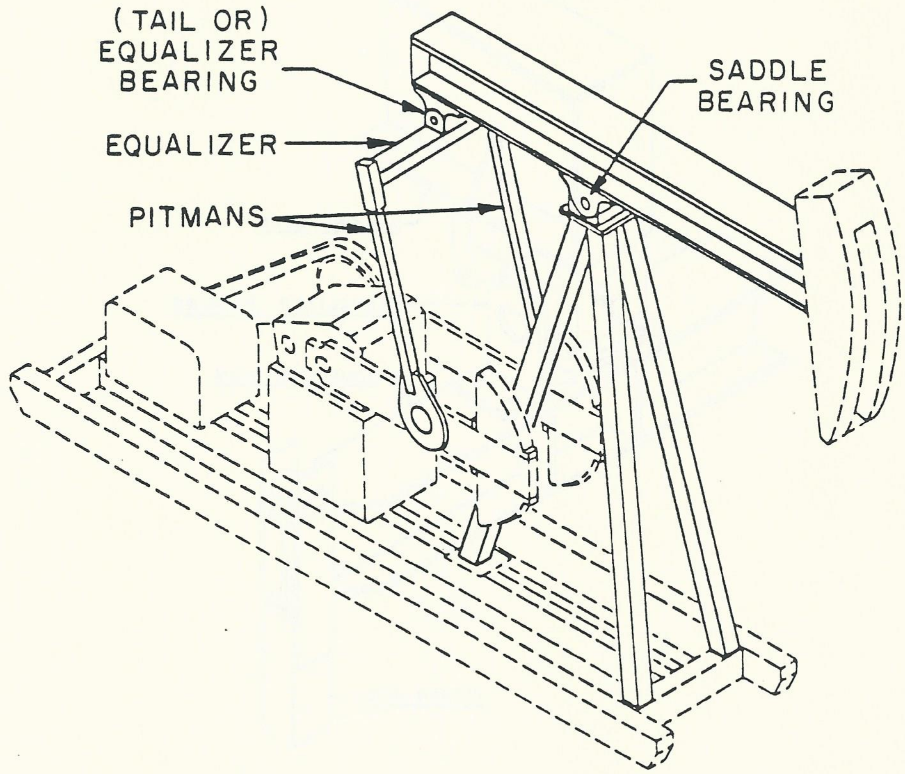
- "B" EQUALS "A" PLUS ½ WIRE LINE DIAMETER.

SADDLE OR CENTER BEARING



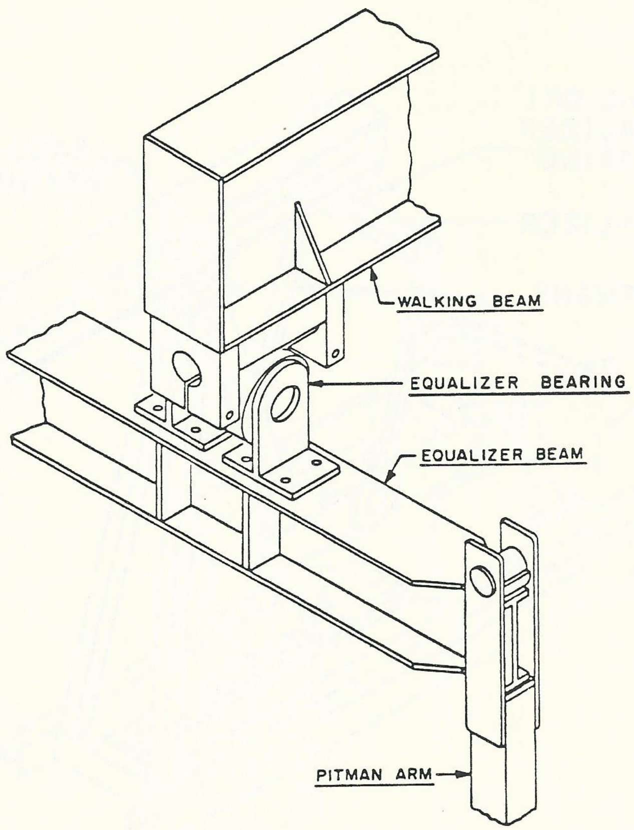
- The walking beam rocks or pivots about the saddle bearing.
- The saddle bearing is supported by the samson post.
- Normally, grease lubricated.
- Allows some lateral alignment & adjustment of the walking beam.

EQUALIZER ASSEMBLY



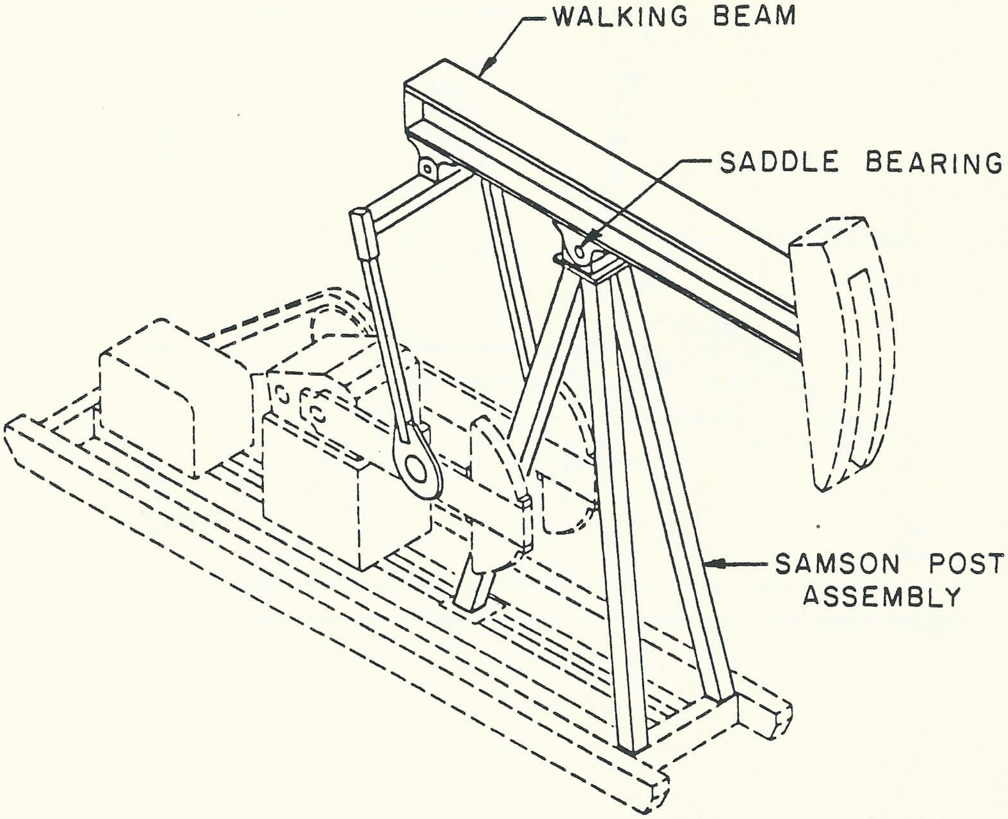
- The walking beam is actuated by the up and down motion of the two pitmans.
- The equalizer balances the load in the two pitmans to compensate for manufacturing tolerances.

EQUALIZER ASSEMBLY



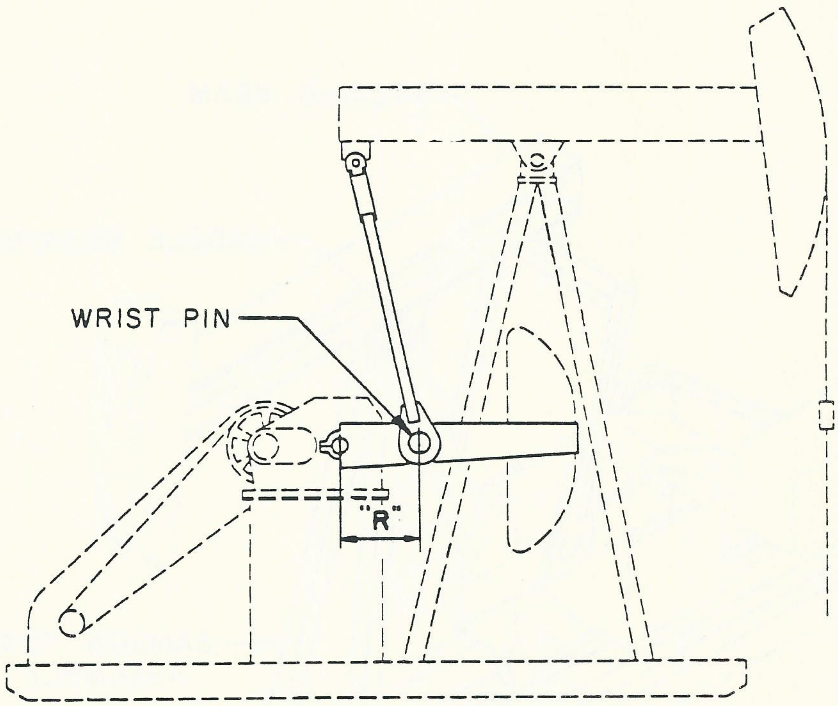
- The walking beam is actuated by the up and down motion of the two pitmans.
- The equalizer balances the load in the two pitmans to compensate for manufacturing tolerances.

SAMSON POST ASSEMBLY



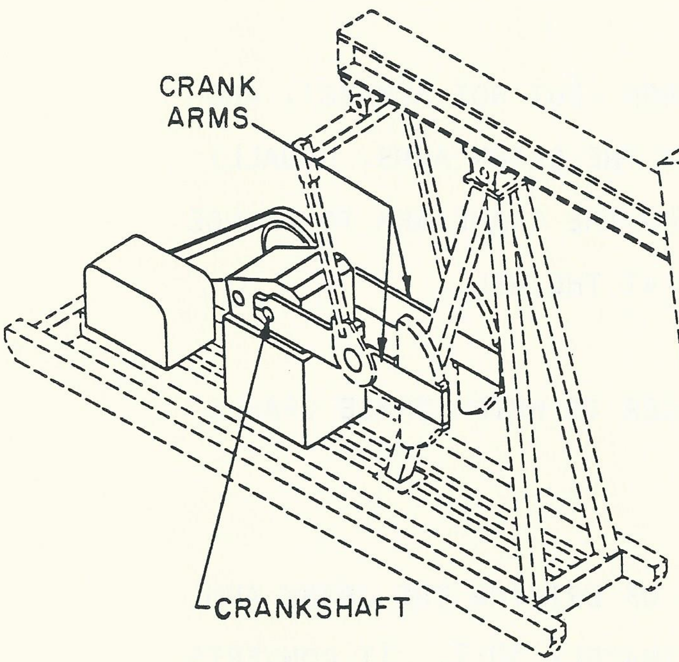
- The walking beam seesaws about the saddle bearing.
- The samson post supports the saddle bearing.

WRIST PIN ASSEMBLY



- THE WRIST PIN CONNECTS THE PITMANS TO THE CRANKS.
- KEEPING THE REST OF THE GEOMETRY THE SAME, INCREASING DIMENSION "R" INCREASES THE STROKE.
- KEEPING THE REST OF THE GEOMETRY THE SAME, DECREASING DIMENSION "R" DECREASES THE STROKE.
- NORMALLY, THERE ARE THREE (3) WRIST PIN LOCATION MOUNTING HOLES, WHICH CREATE THREE (3) DIFFERENT STROKE LENGTHS ON THE SAME UNIT.

CRANK ARMS



- The output shaft of the gear box is called the crank shaft.
- The crank shaft has double shaft extensions for balanced loading.
- There are two crank arms.
- Rotation of the crank shaft swings the crank arm which supplies the movement of the walking beam

THAT IS THE PURPOSE OF THE EQUALIZER BEAM ASSEMBLY. IT IS ABLE TO MOVE AND BALANCE THE LOADS BETWEEN THE TWO CRANK ARMS, THROUGH THE PITMANS.

THE WRIST PIN ASSEMBLIES CONNECT THE PITMANS TO THE CRANK ARMS AND ARE ONE OF THE PROBLEM AREAS OF THE PUMPING UNITS. IF THEY ARE NOT INSTALLED CORRECTLY OR DESIGNED CORRECTLY, THEY WILL HAVE A TENDENCY TO BECOME LOOSE AND WALLOW OUT THE BORE OF A CRANK ARM IN A VERY SHORT TIME AND CREATE A CATASTROPHIC FAILURE.

THE CRANK ARMS ARE USUALLY CAST IRON (BUT NOT ALWAYS), AND HAVE CAST IRON WEIGHTS ATTACHED TO THE CRANK ARMS, USUALLY BY BOLTS. THESE CAN BE MOVED ALONG THE CRANK ARM TO CHANGE THE EFFECTIVE COUNTERBALANCE LOAD AT THE WELL.

THE OUTPUT SHAFT OF THE GEAR REDUCER TO WHICH THESE CRANKS MOUNT IS CALLED THE CRANKSHAFT.

THE GEARBOX ASSEMBLY IS THE MEANS OF DRIVING THE CRANK ARM AND USUALLY HAS A RATIO OF APPROXIMATELY 30:1. IT CONVERTS HIGH SPEED LOW TORQUE INPUT TO HIGH TORQUE LOW SPEED OUTPUT.

THIS GEARBOX IN TURN IS CONNECTED TO ELECTRICAL MOTORS, GAS ENGINES OR OTHER PRIME MOVERS BY MEANS OF BELTS OR PULLIES.

THERE ARE TWO POSITIONS AT WHICH TO MOUNT THE PRIME MOVER:

1. LOW PRIME (OR LOW MOUNT)

2. HIGH PRIME (OR HIGH MOUNT)

LOW PRIME: THE MOTOR OR ENGINE IS SET DOWN ON TOP OF THE SKID OR IN THE CASE OF A T-BASE, A PONY BASE, MOUNTED ON THE FOUNDATION.

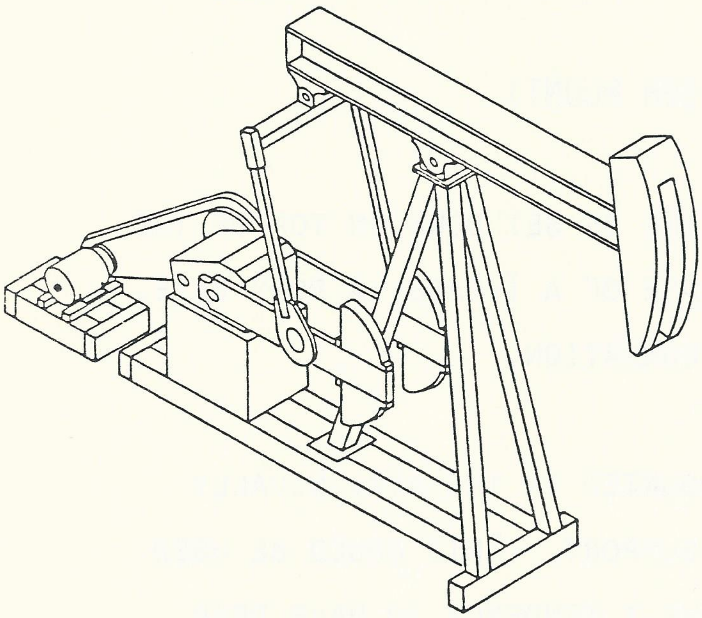
HIGH PRIME: PRIME MOVER IS MOUNTED IN THE AIR, USUALLY OFF THE GEARBOX SUPPORT. THIS COULD BE USED IN AREAS THAT HAVE A TENDENCY TO HAVE DEEP SNOW OR FLOOD WATERS. HIGH PRIMES ARE USUALLY ELECTRIC MOTORS, AND HAVE DECREASED LENGTH (AND COSTS) OF V-BELTS.

THE API RP11L SHOWS YOU HOW TO CALCULATE THE POLISHED ROD LOADS. ONCE YOU HAVE WORKED YOUR WAY THROUGH THE EXAMPLE AND HALF A DOZEN OTHER SOLUTIONS, YOU WILL FIND THAT CALCULATING BECOMES VERY EASY, BUT THAT IT IS TIME CONSUMING. (SEE PAGE 53 FOR API RP11L EXAMPLE.)

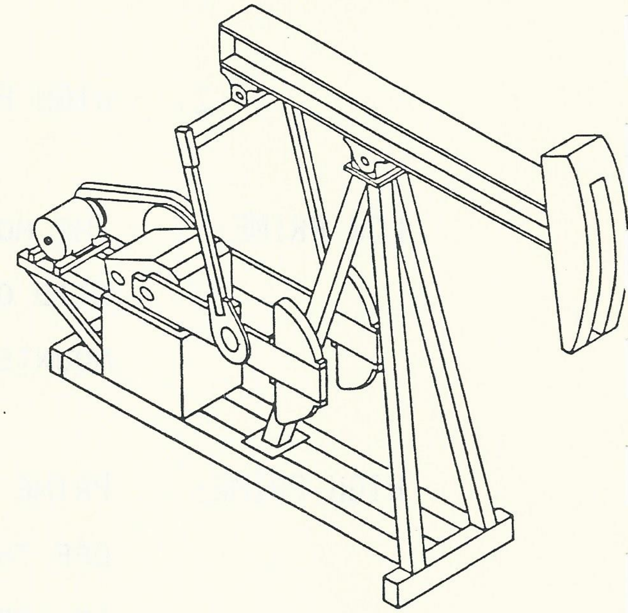
THERE ARE ALSO TWO OR THREE OTHER APPROXIMATE METHODS OF SIZING PUMPING UNITS. I WOULD ADVISE YOU TO USE THE API METHOD AS IT HAS BEEN MY EXPERIENCE THAT SOME OF THE APPROX-

DIFFERENT TYPE PUMPING UNITS

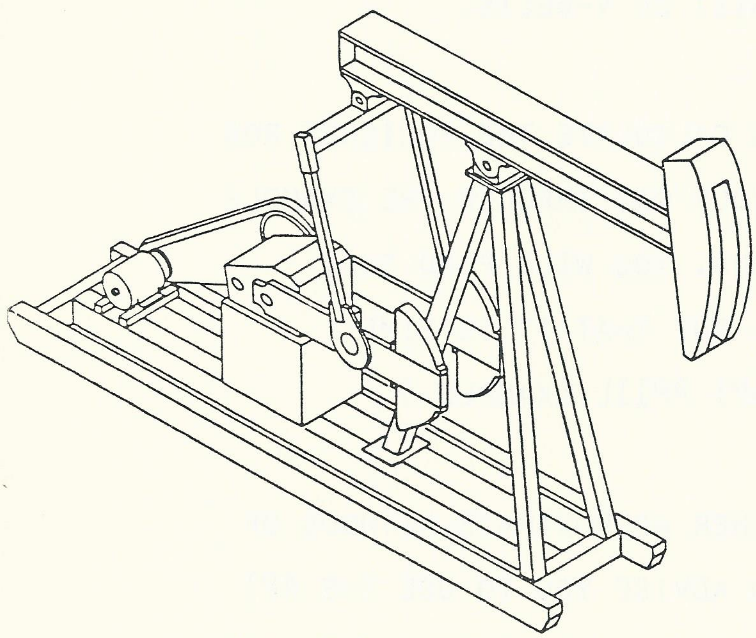
TEE BASE - LOW PRIME
(LOW MOTOR MOUNT)



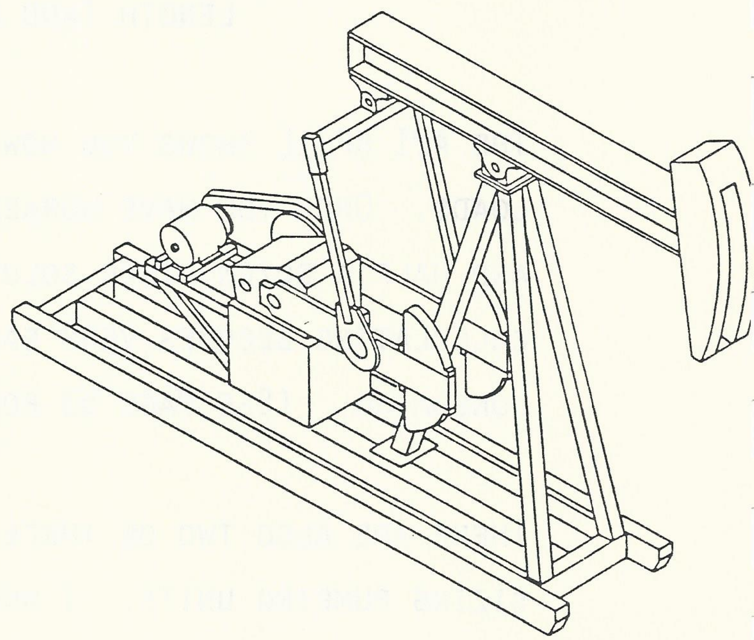
TEE BASE - HIGH PRIME
(HIGH MOTOR MOUNT)



WIDE BASE - LOW PRIME

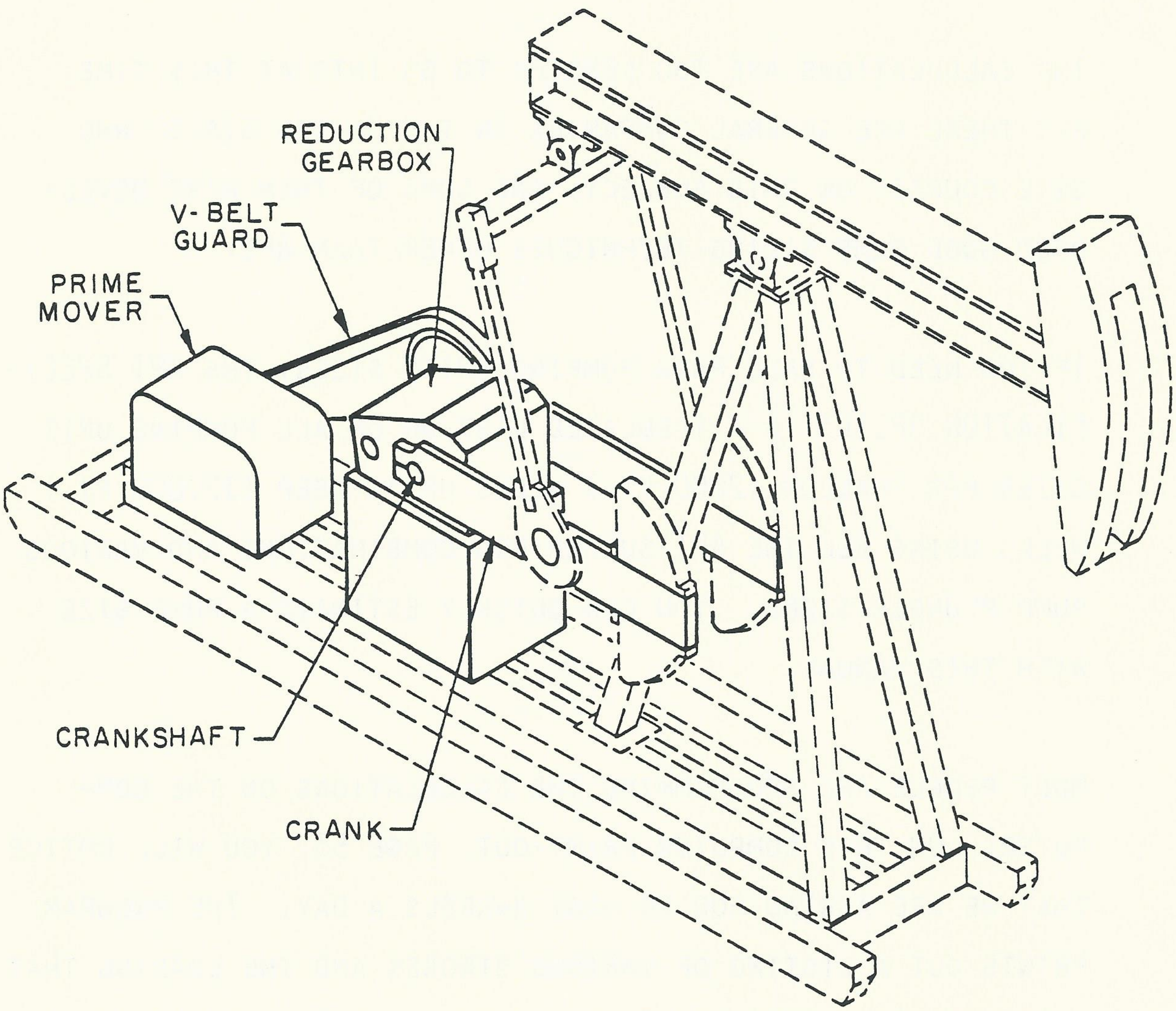


WIDE BASE - HIGH PRIME



(THE WORD "PRIME" REFERS TO THE MOTOR OR ENGINE. A MOTOR CAN BE CALLED A "PRIME MOVER" ON MACHINERY.)

POWER CONVERSION ASSEMBLY



Power for the pumping unit is supplied by the PRIME MOVER.

Rotary motion is changed to reciprocating motion at the pitmans and the crank on the crankshaft.

IMATION METHODS WORK WELL IN DEEP HOLES BUT NOT SHALLOW HOLES. ANOTHER APPROXIMATION METHOD WORKS WELL IN SHALLOW HOLES BUT NOT IN DEEP HOLES.

THE CALCULATIONS ARE TOO LENGTHY TO GO INTO AT THIS TIME, BUT THERE ARE SEVERAL COMPANIES IN THE UNITED STATED WHO GIVE COURSES ON THIS SUBJECT, AND SOME OF THEM HAVE DEVELOPED GOOD PUMP SIZING TECHNIQUES OTHER THAN API.

IF YOU NEED TO BALL PARK PUMPING UNITS SIZES, THE API SPECIFICATION RP11L3 IS A TABULATED LISTING OF ALL PUMPING UNIT SIZES FOR SHALLOW (2000 FT.) WELLS UP TO DEEP (12,000 FT.) WELLS USING ALL THE API SUCKER ROD COMBINATIONS AND VARIOUS PUMP PLUNGER SIZES. YOU CAN QUICKLY ESTIMATE A PUMP SIZE WITH THIS MANUAL.

MOST PEOPLE ARE PROGRAMMING THE CALCULATIONS ON THE COMPUTER. ON THIS COMPUTER PRINT-OUT, PAGE 53, YOU WILL NOTICE THAT WE ARE ASKING FOR SO MANY BARRELS A DAY. THE PROGRAM PRINTS OUT A LISTING OF VARIOUS STROKES AND THE LOADING THAT GOES WITH EACH STROKE. THIS ELIMINATES THE GUESSWORK FROM HOW MANY STROKES PER MINUTE TO RUN THE UNIT AND WHAT STROKE TO START WITH.

YOU WILL SEE THAT THE FIRST THREE STROKES FELL OUTSIDE OF THE API PARAMETERS. THIS DOES NOT MEAN THAT ONE OF THOSE STROKES CANNOT BE USED, BUT IT DOES MEAN THAT YOU SHOULD GO

MORGAN PUMP UNIT LOADING
AS PER API RP11L

JOB NUMBER API EXAMPLE   DATE 09/17/82   TIME 12.40

H = FLUID LEVEL	4500.00	FEET
L = PUMP DEPTH	5000.00	FEET
PD= PUMP DISPLACEMENT BARRELS PER DAY	175.00	
D = PLUNGER DIAMETER	1.50	INCHES
G = SPECIFIC GRAVITY OF FLUID	0.90	

TUBING IS NOT ANCHORED,TUBE SIZE =1.900
ROD NUMBER = 76.

ROD STRING, % OF EACH SIZE

1.25	1.125	1.00	.875	.750	.625	.500
0.0	0.0	0.0	33.8	66.2	0.0	0.0

NOTE: ALL LOADINGS CALCULATED ARE BASED ON THE ABOVE ROD STRING PERCENT. THE USE OF ANY OTHER STRING DESIGN VOIDS THE CALCULATIONS

***

STROKES	STROKES (MINUTE)	PD (BARRELS)	PPRL (POUNDS)	MPRL (POUNDS)	PT (LB-IN)	PRHP (HP)	CBE (POUNDS
16.NOTE:	THE NON-DIMENSIONAL VARIABLES EXCEED THE LIMITS OF THE PROGRAM						
20.NOTE:	THE NON-DIMENSIONAL VARIABLES EXCEED THE LIMITS OF THE PROGRAM						
24.NOTE:	THE NON-DIMENSIONAL VARIABLES EXCEED THE LIMITS OF THE PROGRAM						
30.	21.0	104.69	13579.1	5935.3	59374.3	6.06	10237.8
36.	21.0	142.62	14181.4	5504.6	77142.8	7.98	10237.8
42.	20.6	175.55	14693.7	5120.9	97152.9	9.72	10237.8
48.	18.7	176.60	14616.6	5018.0	113337.9	9.61	10237.8
54.	17.0	176.06	14613.7	5075.6	139525.4	9.58	10237.8
64.	13.0	176.04	14191.7	5518.8	162334.1	8.06	10237.8
74.	11.2	176.86	14021.3	5739.2	181640.7	7.90	10237.8
86.	9.5	176.12	13806.0	6043.2	199294.7	7.71	10237.8
100.	8.0	175.23	13586.5	6365.5	215150.4	7.50	10237.8
120.	6.5	175.18	13355.3	6705.4	234096.0	7.28	10237.8
144.NOTE:	THE NON-DIMENSIONAL VARIABLES EXCEED THE LIMITS OF THE PROGRAM						
168.NOTE:	THE NON-DIMENSIONAL VARIABLES EXCEED THE LIMITS OF THE PROGRAM						

DO YOU WISH TO CONTINUE ENTER YES OR NO
IF YES CLEAR SCREEN BEFORE ENTERING YES
READY

EXAMPLE DESIGN CALCULATIONS
CONVENTIONAL SUCKER ROD PUMPING SYSTEM

Object: To solve for—Sp. PD. PPRL. MPRL. PT. PRHP. and CBE

Known or Assumed Data:

Fluid Level, H = 4,500 ft. Pumping Speed, N = 16 SPM Plunger Diameter, D = 1.50 in.

Pump Depth, L = 5,000 ft. Length of Stroke, S = 54 in. Spec. Grav. of Fluid, G = 0.9

Tubing Size 2 in. Is it anchored? Yes, No Sucker Rods 33.8 % - 7/8" & 66.2 % - 3/4"

Record Factors from Tables 4.1 & 4.2:

1. W_r = 1.833 (Table 4.1. Column 3)
2. E_r = .804 x 10^{-6} (Table 4.1. Column 4)
3. F_c = 1.082 (Table 4.1. Column 5)
4. E_t = .307 x 10^{-6} (Table 4.2. Column 5)

Calculate Non-Dimensional Variables:

5. F_o = .340 x G x D² x H = .340 x 0.9 x 2.25 x 4,500 = 3,098 lbs.
6. $1/k_r = E_r \times L$ = .804 x 10^{-6} x 5,000 = 4.020 x 10^{-3} in/lb.
7. $Sk_r = S \div 1/k_r$ = 54 ÷ 4.020 x 10^{-3} = 13,433 lbs.
8. F_o/Sk_r = 3,098 ÷ 13,433 = .231
9. N/N_o = NL÷245,000 = 16 x 5,000 ÷245,000 = .326
10. $N/N_o' = N/N_o \div F_c$ = .326 ÷ 1.082 = .301
11. $1/k_t = E_t \times L$ = .307 x 10^{-6} x 5,000 = 1.535 x 10^{-3} in/lb.

Solve for S_p and PD:

12. S_p/S = .86 (Figure 4.1)
13. $S_p = [(S_p/S) \times S] - [F_o \times 1/k_t]$ = [.86 x 54] - [3,098 x 1.535 x 10^{-3}] = 41.7 in.
14. PD = 0.1166 x S_p x N x D² = 0.1166 x 41.7 x 16 x 2.25 = 175 barrels per day

If the calculated pump displacement fails to satisfy known or anticipated requirements, appropriate adjustments must be made in the assumed data and steps 1 through 14 repeated. When the calculated pump displacement is acceptable, proceed with the Design Calculation.

Determine Non-Dimensional Parameters:

15. $W = W_r \times L$ = 1.833 x 5,000 = 9,165 lbs.
16. $W_{rf} = W[1 - (.128G)]$ = 9,165 [1-(.128 x .9)] = 8,110 lbs.
17. W_{rf}/Sk_r = 8,110 ÷ 13,433 = .604

Record Non-Dimensional Factors from Figures 4.2 through 4.6:

18. F_1/Sk_r = .465 (Figure 4.2)
19. F_2/Sk_r = .213 (Figure 4.3)
20. $2T/S^2k_r$ = .37 (Figure 4.4)
21. F_3/Sk_r = .29 (Figure 4.5)
22. T_a = .997 (Figure 4.6)

Solve for Operating Characteristics:

23. PPRL = $W_{rf} + [(F_1/Sk_r) \times Sk_r]$ = 8,110 + [.465 x 13,433] = 14,356 lbs.
24. MPRL = $W_{rf} - [(F_2/Sk_r) \times Sk_r]$ = 8,110 - [.22 x 13,433] = 5,249 lbs.
25. PT = $(2T/S^2k_r) \times Sk_r \times S/2 \times T_a$ = .37 x 13,433 x 27 x .997 = 133,793 lb inches
26. PRHP = $(F_3/Sk_r) \times Sk_r \times S \times N \times 2.53 \times 10^{-6}$ = .29 x 13,433 x 54 x 16 x 2.53 x 10^{-6} = 8.5
27. CBE = $1.06(W_{rf} + 1/2\,F_o)$ = 1.06 x (8,110 + 1,549) = 10,239 lbs.

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

1	2	3	4
Pumping Unit Size	Reducer Rating, in.-lb	Structure Capacity, lb	Max. Stroke Length, in.
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

HOW TO READ SPECIFICATIONS

228 — 246 — 86

228	246	86
Peak Torque Rating In Thousands Of Inch Pounds.	Polished Rod Load Rating In Hundreds Of Pounds.	Stroke Length In Inches.

PUMPING UNIT GEAR REDUCED PEAK TORQUE AND STROKES PER MINUTE RATINGS TAKEN FROM API, STD, 11E.

Standard Sizes and Ratings. The pumping-unit reducer of a given size shall have a capacity, calculated as provided herein, as near as practical to, but not less than, the corresponding peak-torque rating in Table 3.1.

TABLE 3.1
PUMPING-UNIT REDUCER SIZES AND RATINGS

1	2
Size	Peak-Torque Rating, in.-lb.
6.4	6,400
10	10,000
16	16,000
25	25,000
40	40,000
57	57,000
80	80,000
114	114,000
160	160,000
228	228,000
320	320,000
456	456,000
640	640,000
912	912,000
1280	1,280,000
1824	1,824,000
2560	2,560,000
3648	3,648,000

Rating Formulas. Gear ratings shall be based on a nominal pumping speed of 20 strokes per minute up to and including the 320 API gear reducer size (peak torque rating — 320,000 inch-pounds). On gear reducers with ratings in excess of 320,000 the ratings shall be based on the following nominal pumping speeds:

Strokes Per Minute	Peak Torque Rating Inch-Pounds
16	456,000
16	640,000
15	912,000
14	1,280,000
13	1,824,000
11	2,560,000 and larger

STROKES PER MINUTE

The API maximum strokes per minute are as follows:

Size	SPM
114	20
160	20
228	20
320	20
456	16
640	16
912	15
1280	14
1824	13
2560 & larger	11

Additionally, the strokes per minute are also limited by the following equation on conventional units:

$$* \quad SPM = 0.70 \sqrt{\frac{60{,}000}{stroke}}$$

Example on a 320-256-120:

$$SPM = 0.70 \sqrt{\frac{60{,}000}{120}} = 15.6$$

The speeds for various strokes by this equation are: These strokes would additionally be limited by the API rating, which ever is lower.

Stroke length	*SPM
54	23.3
64	21.4
74	19.9
86	18.5
100	17.2
120	15.6
144	14.3
168	13.2
192	12.4
216	11.7
240	11.1
300	9.9

* The allowable stroke speed of a faster downstroke geometry would be less than this.

BACK TO THE CALCULATIONS BY HAND TO DETERMINE WHAT THE DIFFERENCE IS BETWEEN THE API CALCULATION AND THE CALCULATION USING THAT STROKE IF YOU WANT TO USE ONE OF THOSE STROKES.

LET US LOOK AT THE PARAMETERS THAT GO INTO DETERMINING THE STROKE OF A PUMPJACK. FIRST, LET US LOOK AT ANCHORED TUBING (SEE TOP OF PAGE 60). IHIS MEANS THE PRODUCTION TUBING IS ANCHORED OR LOCKED AT THE BOTTOM TO THE WELL CASING WHICH IS CEMENTED TO THE ROCK FORMATIONS. THIS ELIMINATES STRETCH IN THE PRODUCTION TUBING AS THE LOAD IS APPLIED OR REMOVED FROM THE STANDING VALVE.

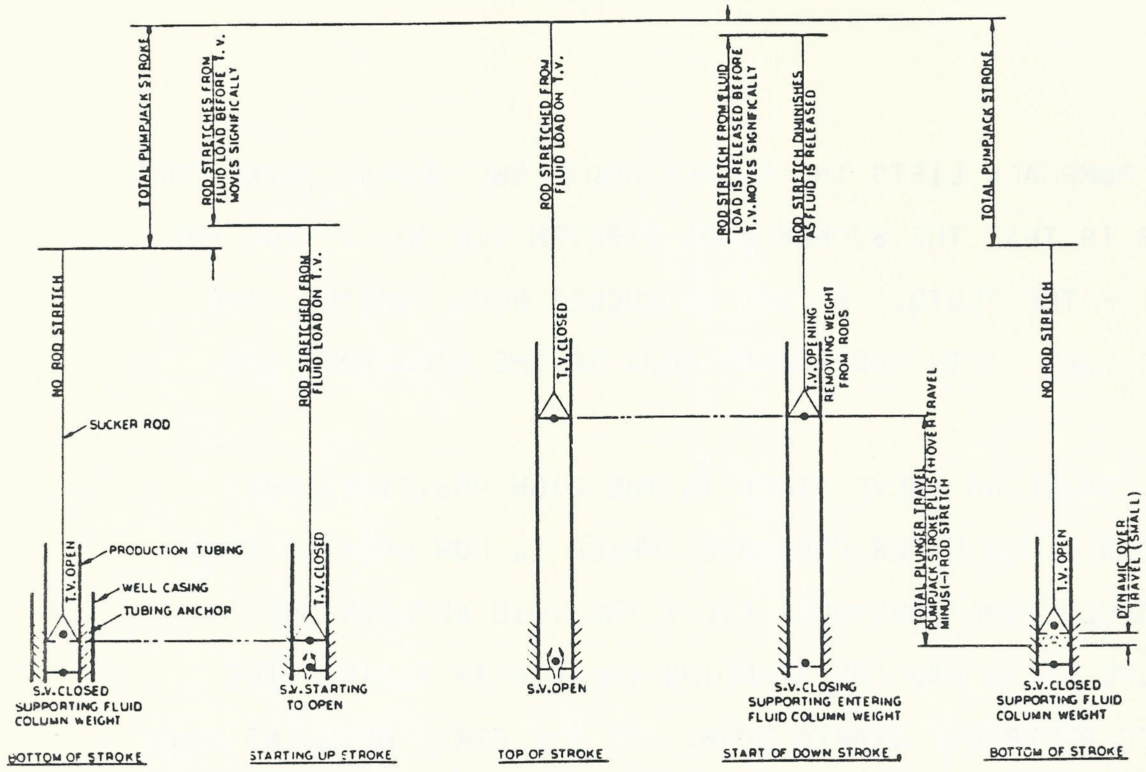
STARTING WITH THE PUMPJACK IN THE DOWN POSITION, STATIONARY, IN PREPARATION FOR THE UPSTROKE, THE STANDING VALVE IS CLOSED; THE TRAVELING VALVE IS OPEN. THE COLUMN OF OIL IS SITTING ON THE STANDING VALVE AND IS SUPPORTED FROM THERE TO THE SURFACE BY THE TUBING. THE TRAVELING VALVE IS OPEN, SUPPORTING NO LOAD.

AS WE START TO LIFT WITH THE PUMPJACK, AT THE FIRST MOVEMENT OF OIL THROUGH THE TRAVELLING VALVE, THE TRAVELING VALVE CLOSES; THE COLUMN OF OIL IS NOW SUPPORTED BY THE TRAVELING VALVE THROUGH THE SUCKER RODS. ITS WEIGHT IS REMOVED FROM THE STANDING VALVE, AND THE PRESSURE IS REDUCED IN THE PUMP.

AS THE PUMPJACK LIFTS THE SUCKER RODS, THE SECOND THING THAT HAPPENS IS THAT THE SUCKER RODS STRETCH DUE TO LIFTING THE WEIGHT OF THE FLUID. AFTER THE SUCKER RODS STRETCH, THE PUMPJACK NOW LIFTS THE SUCKER RODS TO THE HIGH POSITION.

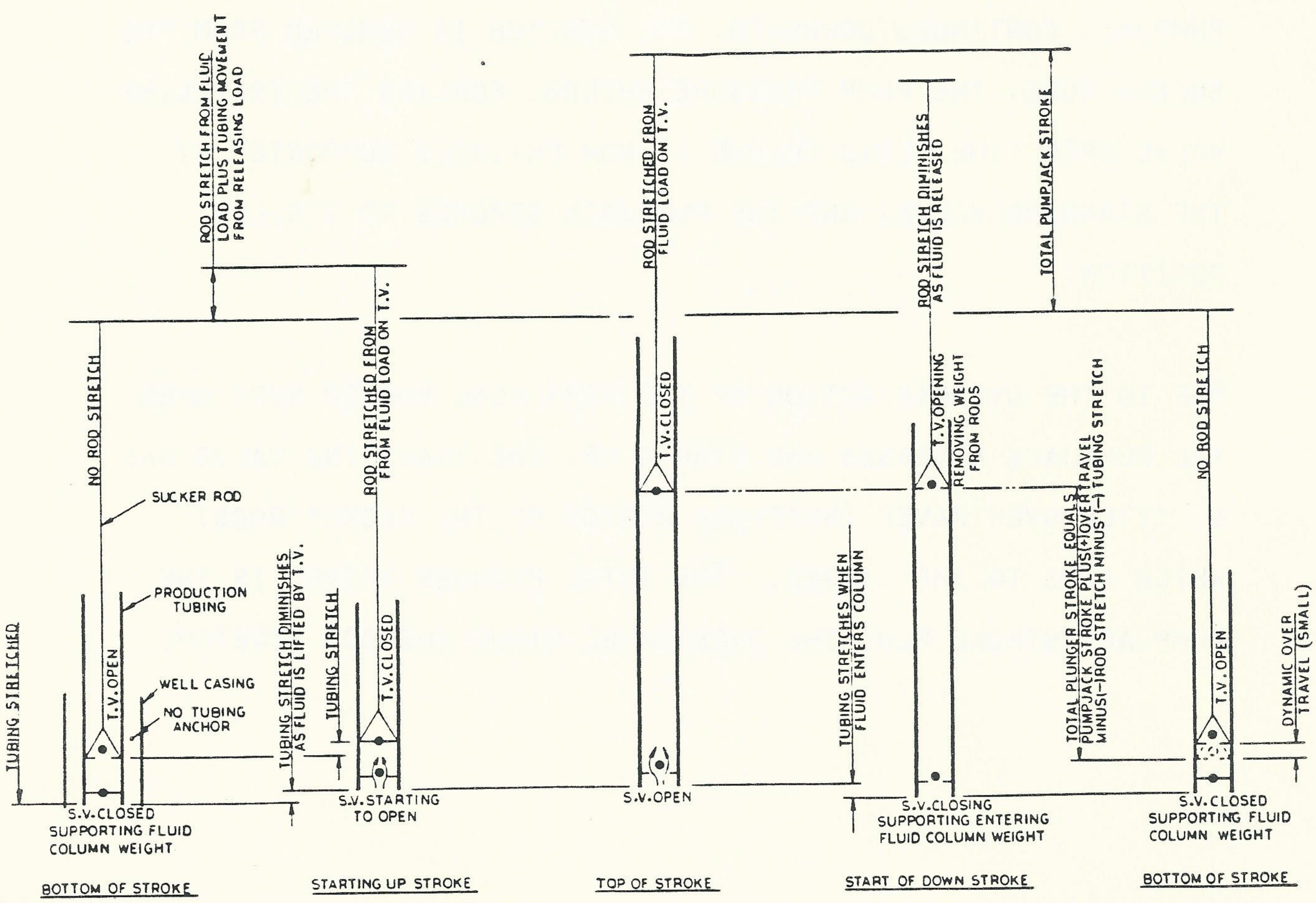
AS THE TRAVELING VALVE RISES TO THE HIGH POSITION, THE RESERVOIR FLUID UNDER PRESSURE, WHICH IS NOW GREATER THAN THE REDUCED PUMP PRESSURE, FILLS THE VOID BETWEEN THE TRAVELING VALVE AND THE STANDING VALVE WITH FLUID. THE PUMPJACK REVERSES, STARTS DOWN. AT THE FIRST MOVEMENT DOWN THE INCREASED PUMP PRESSURE FORCES OIL THROUGH THE STANDING VALVE TOWARD THE RESERVOIR, CAUSING THE STANDING VALVE TO CLOSE, WHICH NOW TRIES TO SUPPORT THE COLUMN OF OIL. AS THE PUMPJACK CONTINUES DOWNWARD, THE STRETCH IS REMOVED FROM THE SUCKER RODS, THE PUMP PRESSURE BUILDS, FORCING THE TRAVELING VALVE OPEN (THE FLUID COLUMN IS NOW ENTIRELY SUPPORTED BY THE STANDING VALVE) AND THE PUMPJACK STROKES TO ITS LOW POSITION.

DUE TO THE DYNAMIC ACTION OF THE VERY LONG SUCKER RODS WHEN THE PUMPJACK REVERSES AND STARTS UP, THE TRAVELING VALVE HAS A LITTLE OVERTRAVEL (WHIPLASH ACTION OF THE SUCKER RODS) WHICH ADDS TO THE STROKE. THE TOTAL PLUNGER TRAVEL IS THE PUMPJACK STROKE PLUS THE OVERTRAVEL MINUS THE ROD STRETCH.



NOTE
T.V. — TRAVELING VALVE
S.V. — STATIONARY VALVE

ANCHORED TUBING



NOTE
T.V. — TRAVELING VALVE
S.V. — STATIONARY VALVE

UNANCHORED TUBING

NOW LET US TAKE A LOOK AT THE EFFECT OF THE UNANCHORED TUBING (SEE BOTTOM OF PAGE 60). EVERYTHING REMAINS THE SAME, EXCEPT THAT AS THE SUCKER RODS AND THE TRAVELING VALVE PICK UP THE WEIGHT OF THE FLUID AND THIS WEIGHT IS REMOVED FROM THE STANDING VALVE, WHICH IS ATTACHED TO THE UNANCHORED TUBING, THE TUBING CONTRACTS AND FOLLOWS THE TRAVELING VALVE FOR SOME SHORT DISTANCE UP THE HOLE. THE REVERSE IS TRUE WHEN THE TRAVELING VALVE STARTS DOWN AND THE WEIGHT OF THE FLUID IS BEING TRANSFERRED FROM THE TRAVEL VALVE TO THE STANDING VALVE - THE TUBE STRETCHES.

THIS TUBING STRETCH SUBTRACTS FROM THE OVERALL PLUNGER STROKE. THE TOTAL PLUNGER STROKE EQUALS PUMPJACK STROKE PLUS OVERTRAVEL MINUS THE ROD STRETCH MINUS THE TUBING STRETCH.

THERE ARE A COUPLE OF THINGS THAT HAPPEN WHEN YOU USE UNANCHORED TUBING. THE FIRST IS THE MOVEMENT OF THE TUBING. THE FLUCTUATION OF THE TUBING CAN CAUSE BUCKLING OF THE TUBE, CREATING ADDITIONAL WEAR BETWEEN THE TUBING AND THE SUCKER RODS. ADDITIONALLY, THE FLUCTUATION OF STRESS REDUCES THE FATIGUE LIFE OF THE TUBE. HOWEVER, ON THE UP SIDE, THE MOVEMENT OF THE TUBING CAN REDUCE IMPACTS AND PEAK LOADINGS, THEREBY INCREASING THE LIFE OF THE OTHER COMPONENT PARTS.

AFTER WE HAVE DETERMINED THE POLISHED ROD LOADS FROM THE API CALCULATIONS, WHICH ARE BASICALLY COMPOSED OF THE WEIGHT OF

THE SUCKER RODS IN FLUID, THE WEIGHT OF THE FLUID PLUS ACCELERATION FORCES, WE MUST NOW DETERMINE HOW MUCH COUNTER-WEIGHT TORQUE IS REQUIRED TO PRODUCE THIS POLISHED ROD FORCE.

USING THE GEOMETRY OF THE UNIT (A RATIO OF THE LEVER ARMS), WE DEVISE A CONSTANT CALLED THE TORQUE FACTOR - MULTIPLYING THE TORQUE FACTOR TIMES THE POLISHED ROD LOAD GIVES US TORQUE ABOUT THE GEARBOX SHAFT.

SUBTRACTING THE COUNTERWEIGHT TORQUE WILL GIVE US THE REQUIRED GEARBOX TORQUE. THERE IS SOME ADDITIONAL EFFECT FROM THE WEIGHT UNBALANCE OF THE WALKING BEAM.

YOU NORMALLY COUNTERBALANCE A WELL BY THE WEIGHT OF THE RODS IN FLUID PLUS 1/2 THE FLUID WEIGHT TIME 1.06. CONSEQUENTLY, DURING THE UP STROKE, WHEN YOU ARE LIFTING THE WEIGHT OF THE RODS IN FLUID PLUS THE FLUID WEIGHT, THE PRIME MOVER MUST LIFT ROUGHLY 1/2 THE FLUID WEIGHT PLUS THE DYNAMIC FORCES. DURING THE DOWNSTROKE, WHEN THE WEIGHT OF THE FLUID HAS BEEN REMOVED FROM THE SUCKER RODS, THE PRIME MOVER MUST LIFT THAT PART OF THE COUNTERWEIGHT WHICH IS EQUAL ROUGHLY TO 1/2 THE FLUID WEIGHT. IDEAL COUNTERBALANCE IS FOR THE UPSTROKE GEAR BOX TORQUE TO EQUAL THE GEAR BOX TORQUE TO LIFT THE COUNTER-WEIGHT ON THE DOWNSTROKE.

TOTAL TORQUE ABOUT GEAR BOX
OUTPUT SHAFT ON CRANK BALANCED UNITS

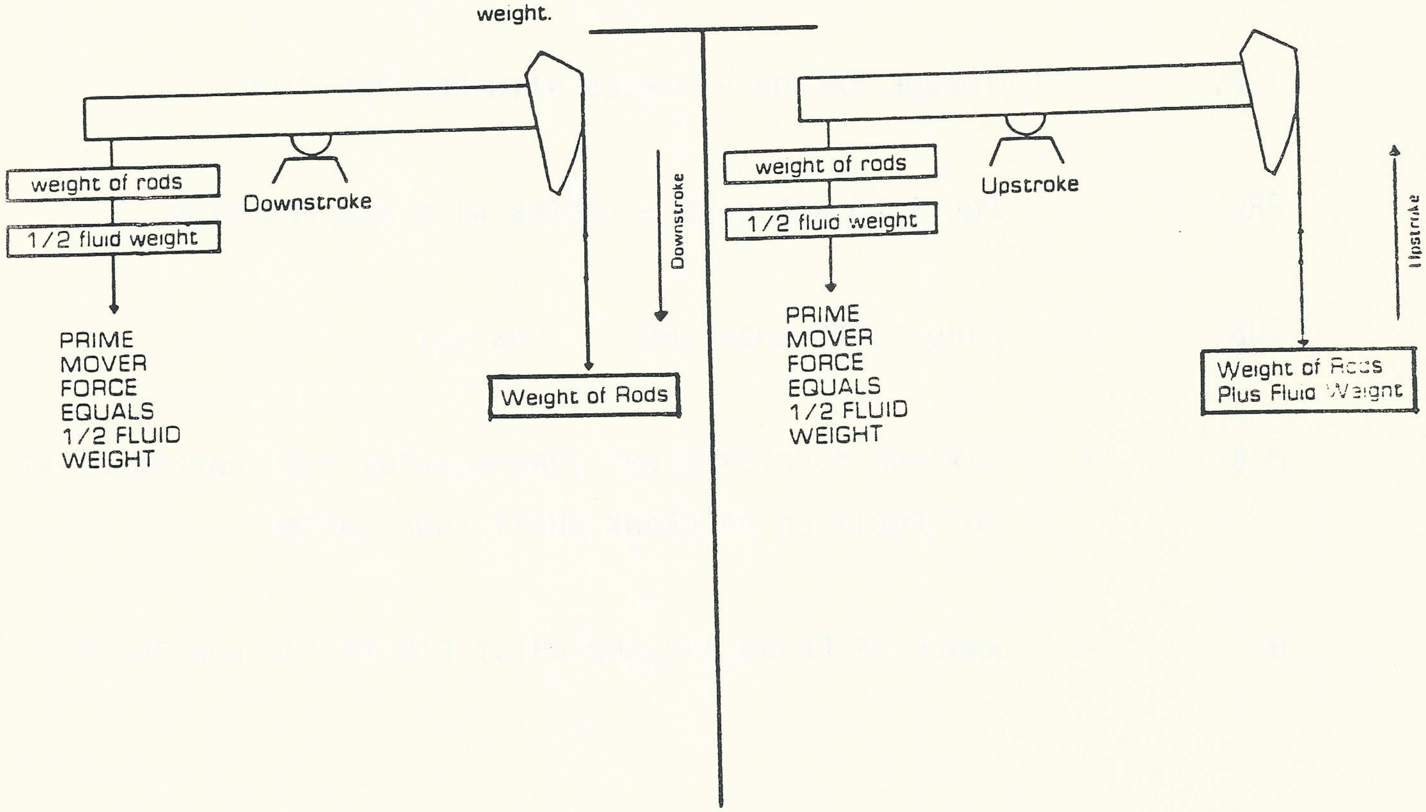
GEARBOX TORQUE + COUNTERWEIGHT TORQUE	= (TORQUE FACTOR) (PPRL - STRUCTURAL UNBALANCE)
GEARBOX TORQUE AT ANY ROD POSITION	= (T.F.) (PRL-SUB) - MCB (SINθ)

WHERE

T.F.	=	TORQUE FACTOR - VARIES WITH ROD POSITION
PRL	=	POLISHED ROD LOAD - VARIES WITH ROD POSITION
SUB	=	STRUCTURAL UNBALANCE - CONSTANT
MCB	=	MAXIMUM COUNTERWEIGHT TORQUE ABOUT THE CRANKSHAFT AT 90° POSITION OR CRANK SHAFT (HORIZONTAL)
θ	=	ANGLE OF CRANK ARM AND WRIST PIN TO THE HORIZONTAL

AMOUNT OF COUNTERBALANCE

To equalize the load on the prime mover and gear box, the units are counterbalanced for the weight of the rods plus 1/2 the fluid weight.



Starting at top dead center or at the bottom of the stroke, the counterbalance helps lift the load and lower reducer torque. When we reach the bottom of the stroke, the polished rod load, together with the reducer torque, is lifting the counterweight. Counterweight momentum would enter into the picture, but it is mostly ignored. Unless a high slip prime mover is used, see page 81 & 82, then we should consider rotating and articulating masses.

The torque curve created by the counterbalance is a smooth harmonic curve. Superimposing the torque curve of a well on top of the counterweight torque curve, which is derived by multiplying the polished rod load in the various positions times the torque factor of the geometry at that position, and subtracting these two curves, we end up with a net gear reduction torque curve which must remain below the rating of the gearbox. If the counterweights were not sized properly, for example: they were too small, the well could overload the gearbox on the upstroke. If the counterweights were too big, the counterweights would overload the gearbox on the downstroke.

Accordingly, it is very critical to have the proper setting of the counterweight. There are various ways to set the counterweight. One of the ways is to install a dynamometer between the polished rod clamp and the carrier bar. This is

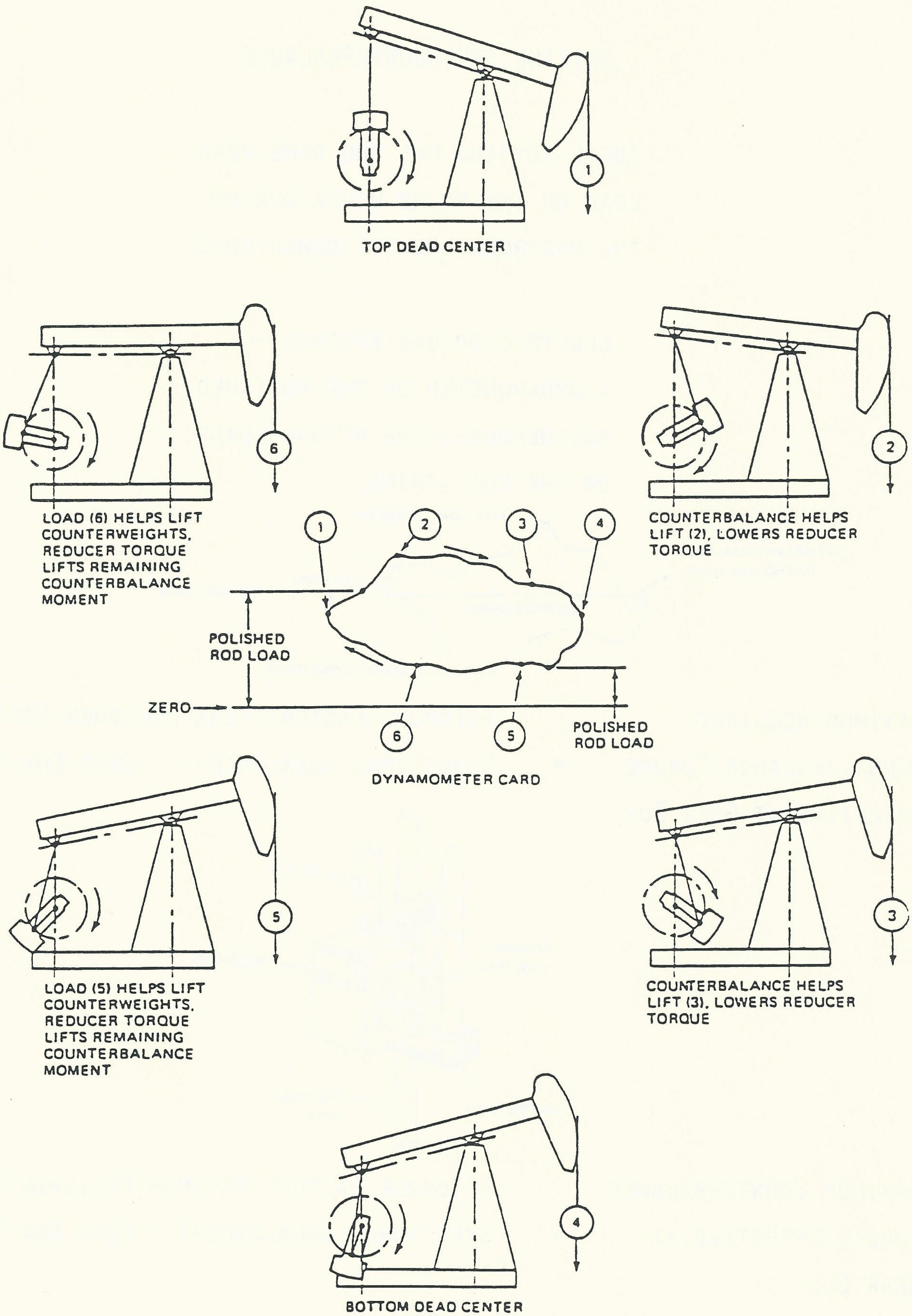
SIMPLY A LOAD CELL DEVICE THAT MEASURES THE VARIOUS POLISHED ROD LOADS. LATER ON IN THE DISCUSSION, WE WILL EXPAND A LITTLE BIT ABOUT THE DYNAMOMETER.

BY MEASURING THE MAXIMUM AND MINIMUM LOADS, WE CAN NOW CALCULATE HOW MUCH COUNTERWEIGHT IS NEEDED AND WHERE TO SET IT.

ANOTHER WAY, WITH ELECTRIC MOTORS, IS TO USE AN AMMETER AND MEASURE THE UPSTROKE AND THE DOWNSTROKE. ON MULTI-CYLINDER ENGINES, A VACUUM GAUGE OR A TACHOMETER CAN BE USED IN THE SAME MANNER AND IF NO EQUIPMENT IS AVAILABLE, LISTEN TO THE SOUND OF THE ENGINE EXHAUST.

SOMETIMES, AFTER A PERIOD OF TIME, YOU CAN DETERMINE IF THE COUNTERWEIGHT IS SET RIGHT BY THE WEAR ON THE BULL GEAR.

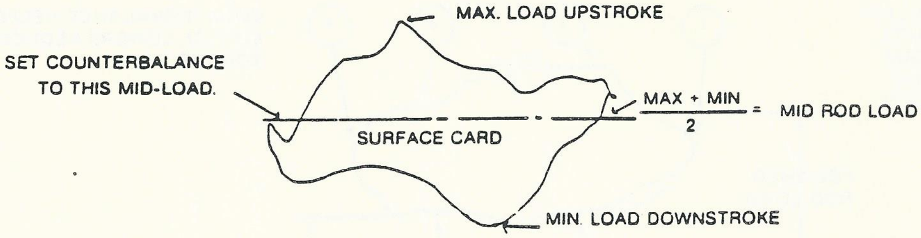
THE MAJOR WEAR OCCURS APPROXIMATELY 180° APART; WHEN THE COUNTERWEIGHT IS LIFTED AND WHEN THE FLUID IS LIFTED. THE AMOUNT OF WEAR SHOULD BE EQUAL. IF IT IS NOT, YOU KNOW THAT THE UNIT HAS NOT BEEN COUNTERBALANCED PROPERLY.
IN ORDER TO UNDERSTAND THE EFFECTS OF VARIOUS DOWNHOLE OPERATIONS AND PROBLEMS, LET US LOOK AT THE SUCKER RODS. THE RODS ARE NORMALLY IN TENSION AND IF WE CAN UNDERSTAND THE EFFECT OF LOADING ON THE ROD, WE THEN CAN UNDERSTAND THE EFFECT OF LOADING ON THE ENTIRE PUMPING SYSTEM.


TOP DEAD CENTER
LOAD (6) HELPS LIFT COUNTERWEIGHTS, REDUCER TORQUE LIFTS REMAINING COUNTERBALANCE MOMENT
COUNTERBALANCE HELPS LIFT (2), LOWERS REDUCER TORQUE
POLISHED ROD LOAD
ZERO
POLISHED ROD LOAD
DYNAMOMETER CARD
LOAD (5) HELPS LIFT COUNTERWEIGHTS, REDUCER TORQUE LIFTS REMAINING COUNTERBALANCE MOMENT
COUNTERBALANCE HELPS LIFT (3), LOWERS REDUCER TORQUE
BOTTOM DEAD CENTER

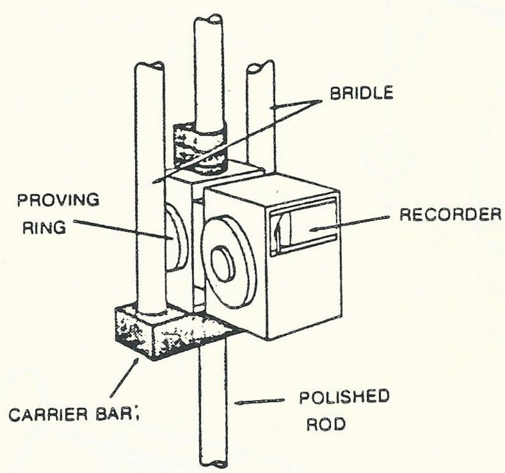
SETTING THE COUNTERBALANCE

IDEAL SETTING HAS THE SAME PEAK LOAD ON THE PRIME MOVER DURING THE UPSTROKE AND THE DOWNSTROKE

ELECTRIC OR GAS ENGINES - A DYNAMOMETER ON THE POLISHED ROD MEASURES THE ACTUAL LOADS ON THE ROD STRING.



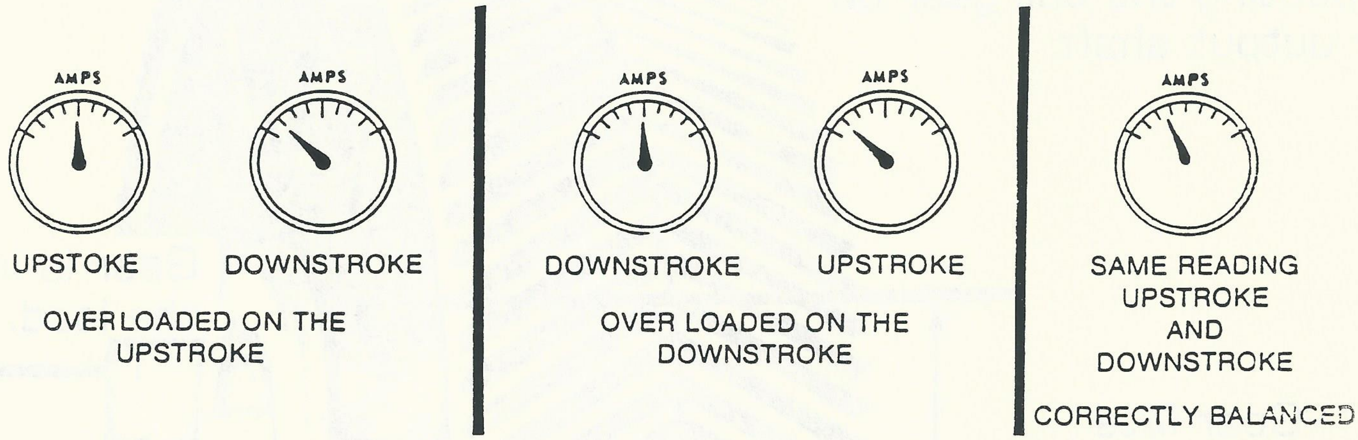
MINIMUM REQUIRED COUNTERBALANCE TORQUE EFFECTIVE AT GEAR BOX = [(TORQUE FACTOR)(PEAK POLISHED ROD LOAD - STRUCTURAL UNBALANCE)] - GEAR BOX TORQUE



MAXIMUM COUNTERBALANCE TORQUE EFFECTIVE AT GEAR BOX = [(TORQUE FACTOR)(MINIMUM POLISHED ROD LOAD- STRUCTURAL UNBALANCE)] + GEAR BOX TORQUE

SETTING THE COUNTERBALANCE

- ELECTRIC MOTORS.
  A AMMETER CAN TO BE USED TO MEASURE THE AMPS OR THE AMOUNT OF CURRENT USED. THEY SHOULD BE THE SAME ON THE UPSTROKE AND THE DOWNSTROKE.

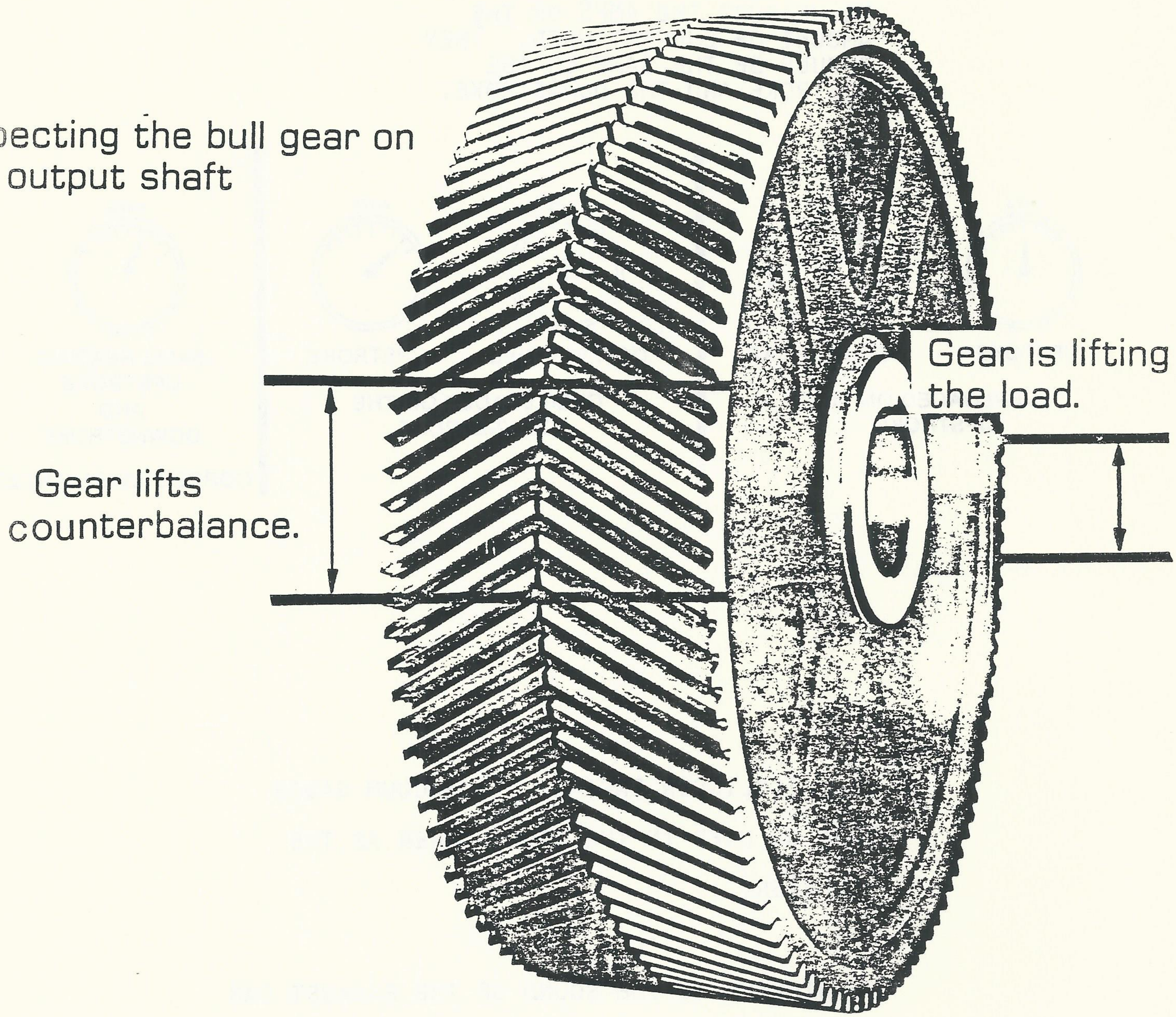


MULTICYLINDER ENGINES. A VACUUM GAUGE CAN BE USED IN THE SAME MANNER AS THE AMMETER.

LISTENING TO THE SOUND OF THE EXHAUST CAN HELP DETERMINE IF THE PEAKS ARE EQUAL.

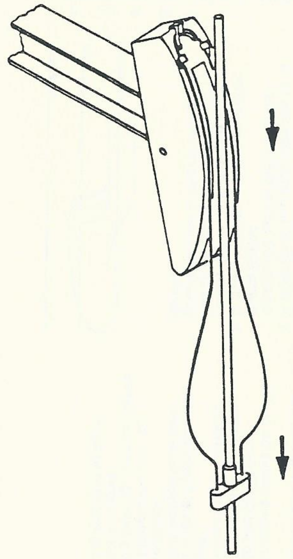
A TACHOMETER CAN BE USED TO MEASURE MOTOR RPMS.

Inspecting the bull gear on the output shaft



Two major points of wear occur ideally, the amount should be equal.

PROBLEM: Hanger Assembly Moves Down Faster Than The Rod String.



RESULTS

- The rod string does not help move walking beam.
- Counterbalance does not operate properly.
- Loss of contact of carrier bar shortens downhole pump stroke.
- Impact of carrier bar on upstroke can damage equipment.

CAUSES

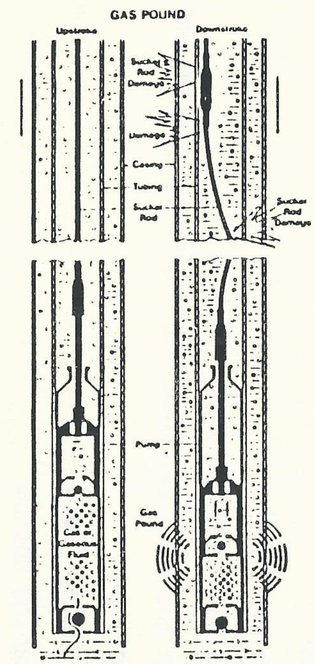
- Parrafin deposits.
- Tight packing in the stuffing box.
- Fluid pound.
- Stuck pump.
- Excessive downhole friction.
- Stroke speed is too fast.

PROBLEM: When watching the operation of a counterbalanced Pumping Unit, when erratic or jerky motion is observed:

Some possible causes are:

- Improper position of the counterweights causing an out of balance
- Parted Sucker Rods
- A stuck downhole pump
- Stuffing box packing too tight
- Polished rod surface not lubricated or is rough
- Paraffin build up in the well is hindering the sucker rod movement
- The foundation is failing
- Downhole pump is pounding fluid
- Downhole pump is tapping bottom

GAS POUND



Many wells produce gas along with the well fluids. The presence of free or break-out gas at the pump can interfere with the efficiency of the pump action, thereby reducing the amount of fluid produced. This interference can result in a gas pound and when extreme it results in completely gas locking the pump.

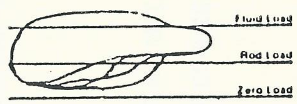
During the pumping unit upstroke, the pump barrel is filled with fluid and free gas, usually in a frothy condition. On the downstroke, the rod and fluid loads above the plunger must compress this gas until the pressure above and below the traveling valve are equalized to allow the traveling valve to open and discharge the gas and fluid into the tubing. At this time, a pound or shock wave similar to that produced by a fluid pound, only cushioned more, travels through the system.

Gas pounds cause the same problems with equipment and production as fluid pounds. Fluid pounds are controllable to a high degree, whereas gas pounds can usually only be partially controlled. Some control can be maintained if gas separation is possible before the gas enters the pump, but where gas breaks out of solution during the pressure drop within the pump, only partial control can be maintained. In a situation where gas completely fills the pump barrel the pump gas locks, which means that not enough pressure or vacuum can be built within the pump to open either the traveling or standing valve and production will cease.

A gas pound can cause the same severe damages as a fluid pound.

A gas pound may be identified as follows:

1. Fluid at the bleeder valve is frothy and foamy and has a tendency to blow out instead of flowing.
2. Production varies greatly from day to day.
3. Theoretical bottom hole pump efficiency below 70% with 20% and 10% not uncommon.
4. Fluid level readings in the annulus vary due to froth.
5. A gas pound, as reflected on a dynamometer card trace, constantly changes as the gas volume in the pump changes.



Handling gas interference in pumping wells is at best a trial and error problem. There is no best method for every well. Some suggestions to control gas pounds are:

Remedy

1. Reduce casing back pressure to only a few ounces but do not open to atmosphere.
2. Lower tubing perforations and pump intake as far below casing perforations or formation face as possible to prevent turbulence at the intake which starts vertical gas flow.
3. Install a downhole gas separation system to keep as much gas as possible out of the pump. A simple mud anchor—gas anchor system often works. Where the gas breaks out will determine the system to be tried.
4. Design the bottom hole pump to increase the compression ratio, such as close valve spacing, oversized standing valve and intake, etc. Work toward reducing turbulence inside the pump.
5. Try a backpressure valve on the flowline to help hold gas in solution.
6. Vary unit strokes per minute and stroke length.
7. Do not bump bottom to break gas locking. This will damage the pump as well as creating a secondary shock wave through the pumping system.
8. Change bottom hole pump design to better handle such gaseous fluids.

FLUID POUND

A great many pumping wells pound fluid, either intentionally or unintentionally, and any well is subject to pounding depending on many pumping conditions.

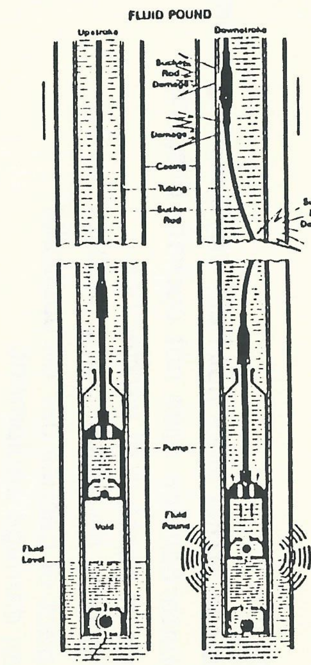
A "fluid pound," as experienced in oil well pumping, is caused by the pump not completely filling with fluid on the upstroke. As the downstroke begins, the entire fluid and sucker rod load moves down through a void until the plunger hits the fluid level in the pump barrel. The traveling valve opens, suddenly transferring the load to the tubing, causing a sharp decrease in load which transmits a shock wave through the pumping system. It is this shock wave that damages the parts of the pumping system.

A fluid pound is always undesirable and we recommend that the pumping system be monitored to detect a pound and changes be made in the system to eliminate or reduce the pound. The worst condition is when the pound occurs near the middle of the downstroke as the velocity of the falling loads are maximum and the resulting shock wave is greater.

In some wells the only way a pumper knows he is getting all of the fluid available is to pound the well. Under this situation, the system should be changed to keep the pound at the top of the downstroke where the resulting effect of the shock wave is less.

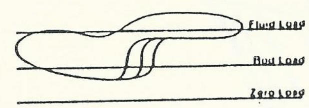
A fluid pound can cause severe damage as follows:

1. The sucker rod tends to bend in the same area on each downstroke due to the buckling effect. This stretches the skin on the convex side of the sucker rod bend. This repetitive bending eventually creates fatigue cracking at this point. Such fatigue cracking ultimately leads to failure.
2. The inhibitor film from the portion of the sucker rod that slaps against the tubing wall is removed by such rubbing action. It also causes wear of the rod at this point due to abrasion. Both conditions can result in failure.
3. Contributes to unscrewing of the rod and coupling caused by this bending motion and slapping the tubing.
4. Creates excessive stress on pump parts due to shock loading.
5. If tubing is not anchored, it also bends or buckles due to sudden load changes.
6. Pumping unit gear boxes and bearings can be damaged from the shock wave effect.
7. Increases lifting costs due to unnecessary pulling costs resulting from any of the above.
8. Production is lost due to such downtime.



A fluid pound may be identified as follows:

1. The pumping unit carrier bar turns on the downstroke.
2. The shock wave of the pound in shallow wells can often times be felt by holding the hand on the polished rod.
3. Production per day is less than the particular pumping cycle is capable of producing.
4. A fluid level test indicates the fluid level to be below the top of the pump barrel.
5. A dynamometer card trace will be nearly consistent, with the slope of the pound very steep when the pound occurs in the middle of the stroke.



Well costs can be reduced by keeping fluid pounds to a minimum consistent with maintaining maximum available production. The following are suggestions to make the pumping system more equal the amount of fluid available to it from the well, thus controlling the fluid pound.

Remedy

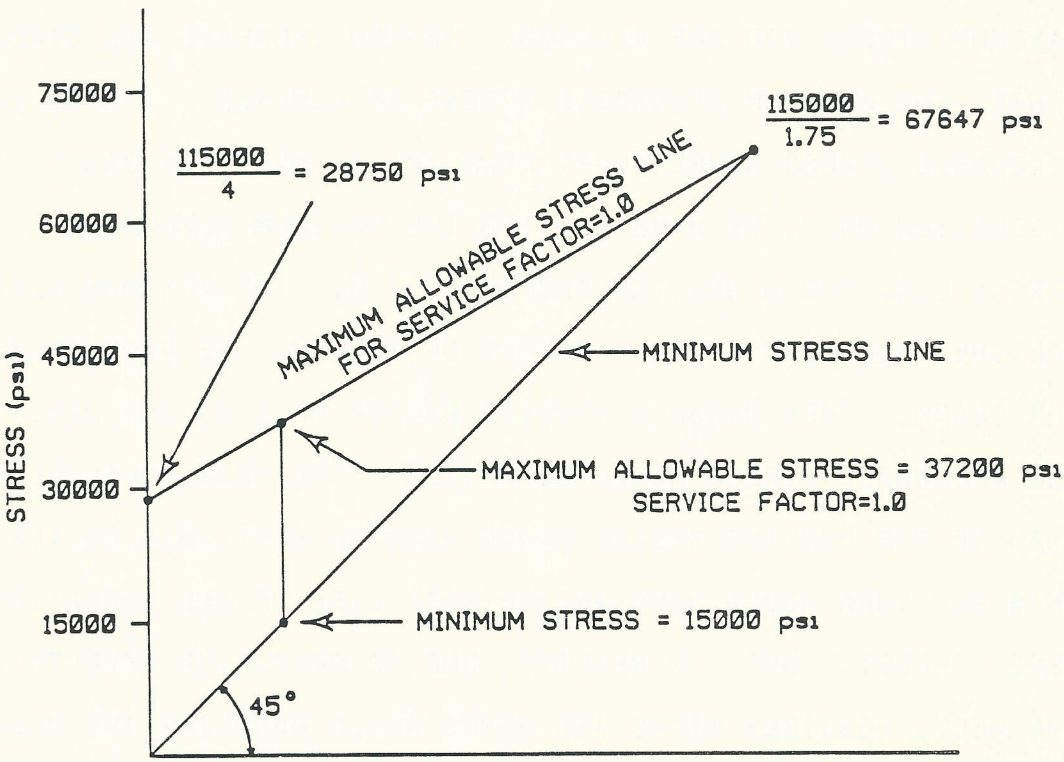
1. Change operational design by changing the stroke length, pumping speed, or pump size or design. The objective would be to change toward a longer stroke at a slower speed and/or reduce the plunger diameter.
2. Time cycle the pumping time to allow the pump barrel to fill.
3. Reduce casing back pressure, but do not open casing to atmosphere.
4. Lower pump as far as possible to increase submergence.
5. Install pump-off controller on pumping unit.
6. Keep casing and tubing perforations and pump intake open, clean, and treat against scale buildup.

A PUMPING UNIT CAN SEE MANY MILLIONS OF CYCLES IN A YEAR'S TIME. IN ORDER FOR IT TO LAST FOR A NUMBER OF YEARS, IT MUST BE DESIGNED WITH AN INFINITE LIFE FATIGUE ANALYSIS. WHEN WE TALK ABOUT FATIGUE DESIGN, IT IS IMPORTANT NOT ONLY TO TALK ABOUT THE MAXIMUM ALLOWABLE STRESS AND SAFETY FACTORS, BUT STRESS RANGE.

AS YOU DECREASE THE STRESS RANGE, THE HIGHER THE ALLOWABLE MAXIMUM STRESS YOU ARE ALLOWED. IF YOU INCREASE THE STRESS RANGE, THE MAXIMUM ALLOWABLE STRESS IS LOWERED. TAKING THIS ALLOWABLE STRESS RANGE CHART FROM API 11 BR, FOR BASIC DISCUSSION ONLY, WE HAVE PLOTTED THE MAXIMUM ALLOWABLE STRESS LINE FOR A SERVICE FACTOR OF 1.0. THE MINIMUM PEAK POLISHED ROD LOAD GIVES A STRESS OF 15,000. WE PLOT THAT ON OUR GRAPH. THEN UNDER A SERVICE FACTOR OF 1.0, OUR MAXIMUM ALLOWABLE STRESS WOULD BE 37,200. THE PEAK POLISHED ROD LOAD OF THE PUMPING SYSTEM WOULD HAVE TO MEET THIS CRITERIA. THIS ROD THEN WOULD HAVE AN INFINITE LIFE IF ALL OPERATIONS WERE PERFECT. BUT, IN REALITY, DUE TO CORROSIVE EFFECTS IN THE WELL, POSSIBLY NICKS AND BANGS WHILE HANDLING THE RODS, ANYTHING THAT CREATES STRESS CONCENTRATION OR OVERLOADS FROM DOWNHOLE PROBLEMS WOULD LIMIT THE LIFE OF THIS ROD. THE STRESS CONCENTRATIONS HAVE THE SAME EFFECT AS INCREASING THE MAXIMUM STRESS.

TO INCREASE THE LIFE YOU WOULD APPLY A SERVICE FACTOR TO THE MAXIMUM ALLOWABLE STRESS. REDUCING THE MAXIMUM ALLOWABLE

ALLOWABLE STRESS RANGE OF SUCKER RODS



API GRADE D
MINIMUM TENSILE STRENGTH = 115000 psi
SERVICE FACTOR = 1.0

STRESS AND STRESS RANGE, AND KEEPING THE MINIMUM STRESS THE SAME, WOULD INCREASE THE LIFE OF THE ROD. THIS SERVICE FACTOR WOULD HAVE TO BE SELECTED ON EXPERIENCE FOR THE TYPE OF WELL YOU ARE PUMPING.

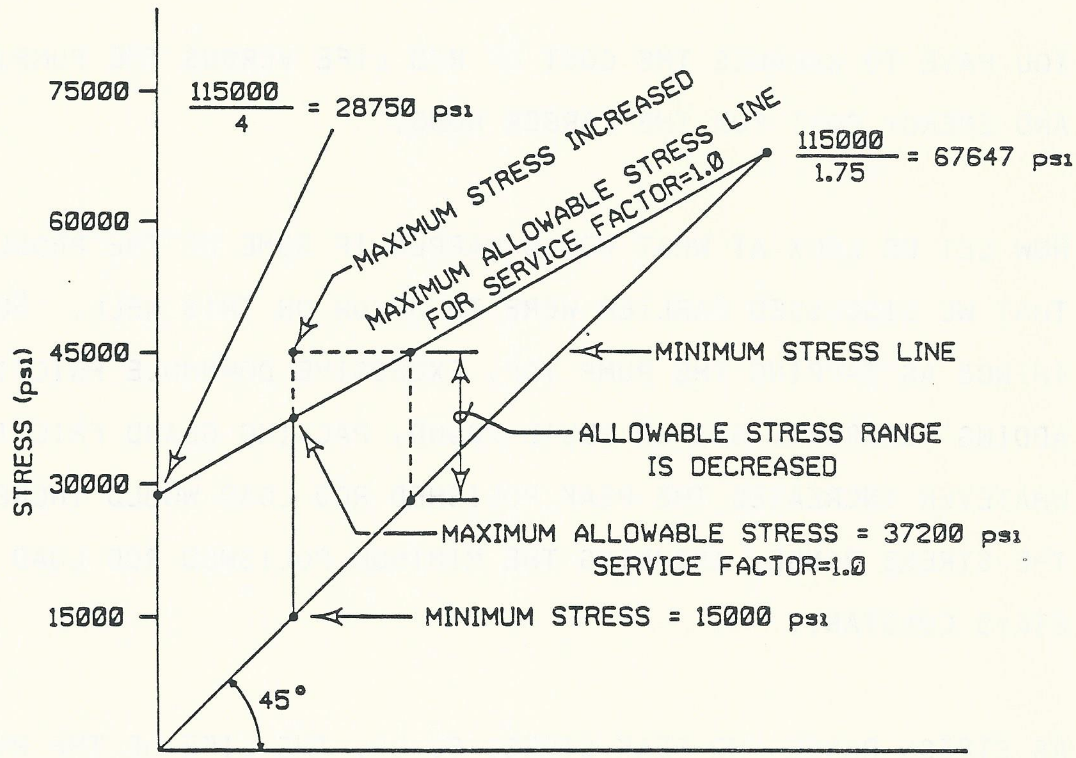
YOU HAVE TO BALANCE THE COST OF ROD LIFE VERSUS THE PUMPJACK AND ENERGY COST FOR THE LARGER RODS.

NOW LET US LOOK AT WHAT WOULD HAPPEN IF SOME OF THE PROBLEMS THAT WE DISCUSSED EARLIER WERE TO OCCUR ON THIS WELL. SUCH THINGS AS TAPPING THE PUMP TOP, EXCESSIVE DOWNHOLE FRICTION, ADDING SCRAPERS, GAS OR FLUID POUND, PACKING GLAND FRICTION; WHATEVER INCREASES THE PEAK POLISHED ROD LOAD WOULD INCREASE THE STRESS RANGE, ASSUMING THE MINIMUM POLISHED ROD LOAD STAYS CONSTANT.

AS STRESS RANGE AND PEAK STRESS GO UP, THE LIFE OF THE ROD WOULD GO DOWN. AND ADDITIONALLY, IF THE MINIMUM STRESS WOULD BE DECREASED FROM LIGHTLY TAPPING THE PUMP BOTTOM, FLUID POUND, DOWNHOLE PRESSURE, THE MAXIMUM ALLOWABLE STRESS WOULD BE DECREASED. THIS INCREASE OF STRESS RANGE BY REDUCING THE MINIMUM STRESS WOULD REDUCE THE LIFE OF THE RODS. TO MAINTAIN YOUR ROD LIFE, YOU MUST KEEP THE MINIMUM AND MAXIMUM STRESS WITHIN THE ALLOWABLE RANGE DESIGN.

LET'S NOW RETURN TO THE DYNAMOMETER WEIGHING OF THE WELL THAT I MENTIONED EARLIER.

ALLOWABLE STRESS RANGE OF SUCKER RODS



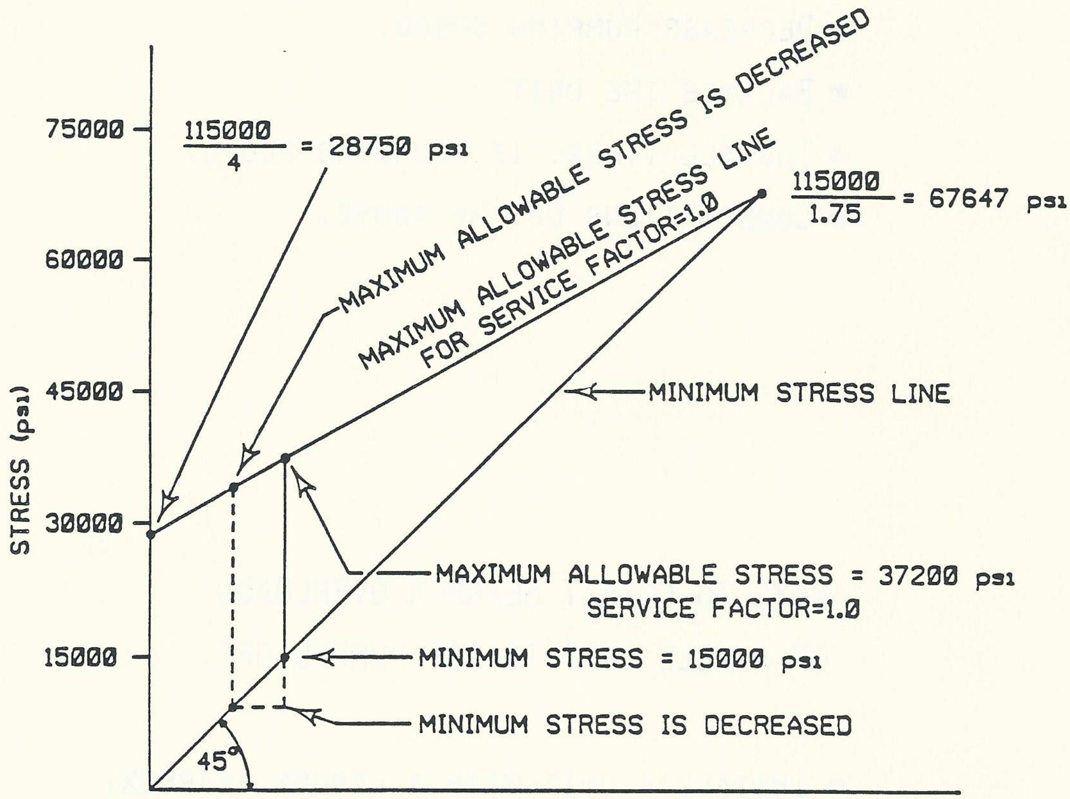
API GRADE D
MINIMUM TENSILE STRENGTH = 115000 psi
SERVICE FACTOR = 1.0

IF THE MAXIMUM STRESS IS INCREASED FROM SUCH THINGS AS:

- TAPPING THE PUMP TOP,
- EXCESSIVE DOWN HOLE FRICTION,
- ADDING SCRAPERS,
- FLUID POUND,
- PACKING GLAND FRICTION,
- ETC.;

THE LIFE OF THE SUCKER RODS WILL BE DECREASED AS THE STRESS RANGE INCREASES.

ALLOWABLE STRESS RANGE OF SUCKER RODS



API GRADE D
MINIMUM TENSILE STRENGTH = 115000 psi
SERVICE FACTOR = 1.0

IF MINIMUM STRESS IS DECREASED FROM SUCH THINGS AS:
- TAPPING THE PUMP BOTTOM,
- FLUID POUND,
- DOWN HOLE FRICTION,
- ETC.;

THE MAXIMUM ALLOWABLE STRESS WILL BE DECREASED.
IF NOT, AS THE SUCKER ROD STRESS RANGE IS INCREASED, THE LIFE OF THE RODS IS DECREASED.

SOME WAYS TO COMBAT GEARBOX OVERLOADS ON A WELL THAT IS PUMPED OFF.

- SHORTEN THE STROKE.
- INSTALL A SMALLER DOWNHOLE PUMP.
- DECREASE PUMPING SPEED.
- BALANCE THE UNIT.
- INSTALL TIMER, IF POUNDING FLUID.
- COMBINATIONS OF THE ABOVE.

WAYS TO COMBAT GEARBOX OVERLOADS ON A WELL THAT IS NOT PUMPED OFF.

- INSTALL A UNIT WITH A LARGER GEARBOX.
- BALANCE THE UNIT.
- REDUCE THE PRODUCTION BY:

  A. SHORTENING THE STROKE

  B. DECREASING THE PUMPING SPEED.

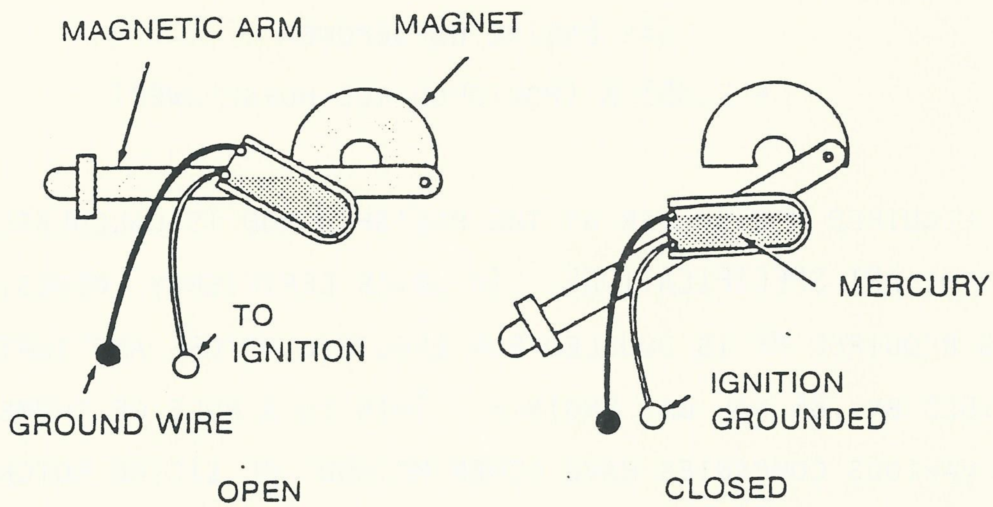
  C. INSTALLING A SMALLER DOWNHOLE PUMP.

  D. COMBINATIONS OF A, B, C.

- MAINTAIN PRODUCTION BY DECREASING STROKE AND INCREASING PUMPING SPEED WILL REDUCE GEARBOX TORQUE IN MANY CASES.

Several of the problems encountered in the pumping operation cause vibrations in the walking beam. This vibration can be sensed by various types of switches which can stop the prime mover.

SAFETY SWITCHES



Here is a switch that is held open by a magnetic arm.

When vibration jars the arm off the magnet, the mercury conducts the ground wire to the ignition, stopping an IC engine.

Or, liquid mercury may be used to conduct the current to an AC motor, when the switch lets the mercury fall out of the line of circuit, the motor stops.

PRIME MOVER HORSEPOWER
(APPROXIMATION)

ELECTRIC MOTOR HORSEPOWER
= 2 X (POLISHED ROD HORSEPOWER)

GAS ENGINE HORSEPOWER
= 2.353 X (POLISHED ROD HORSEPOWER)

THE REQUIRED HORSEPOWER AT THE POLISHED ROD IS CALCULATED AS PER THE API SPECIFICATIONS. TO COVER EFFICIENCY LOSSES, THIS REQUIRED HP IS DOUBLED FOR ELECTRIC MOTORS AND THAT IS DIVIDED BY .85 FOR GAS ENGINES. THIS IS A RULE OF THUMB, AND VARIOUS COMPANIES HAVE OTHER METHODS OF SIZING MOTORS.

PRIME MOVER

$$\text{SPEED VARIATION IN \%} = \frac{\text{MAX-MIN}}{\text{MAX}} \times 100$$

- SPEED VARIATIONS ON NEMA D ELECTRIC MOTOR SIMILAR TO LOW SPEED GAS ENGINE.

- SPEED VARIATIONS OF ULTRA HIGH SLIP ELECTRIC MOTOR SIMILAR TO HIGH SPEED GAS ENGINE.

PRIME MOVERS

ELECTRIC MOTORS -

STANDARD NEMA D MOTOR

SPEED VARIATION IS SMALL

5 - 12% OF FULL LOAD

ULTRA HIGH SLIP PRIME MOVER

SPEED VARIATION IS LARGE

35 - 50% OF FULL LOAD

GAS ENGINES -

HIGH SPEED

USUALLY MULTI-CYLINDERS

OPERATE BETWEEN 800 TO 1400 RPM

SMALL FLY WHEEL EFFECT

LOW SPEED

NORMAL SINGLE CYLINDER

OPERATE BETWEEN 200 - 600 RPM

LARGE FLY WHEEL EFFECT

ADVANTAGE OF GAS ENGINE OVER ELECTRIC MOTOR

LESS ENERGY COST

DISADVANTAGE OF GAS ENGINE OVER ELECTRIC MOTOR

INITIAL COST HIGHER

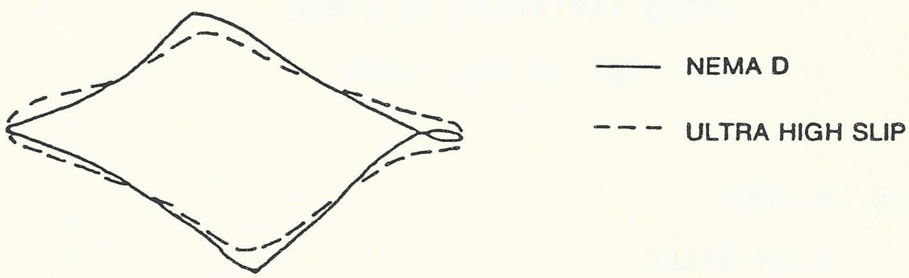
MAINTENANCE HIGHER

LESS AUTOMATION

PRIME MOVERS

Large Speed Variation Advantages.

1. Improved rod load pattern.
   - Higher minimum load - lower peak load.
   - Illustrated by these dynameter cards.



2. Rotary inertia effects of
   cranks and weights.
   - Slipping (slowing down) of motor at high torque reduces peak load.
   - Slowing down of drive allows slowing down of cranks and weights giving of their energy to the pump.
   - Energy is put back (motor speed up) when the rod load is small.
   - Reducing the fly wheel effect ahead of the reducer allowing cranks and weights to slow down increases the gear life.
   - Counter weight position can aid the inertia effect.

BETWEEN THE POLISHED ROD CLAMP AND THE CARRIER BAR, A LOAD CELL CAN BE LOCATED. THIS LOAD CELL MEASURES THE LOADS IMPOSED ON THE PUMPJACK FROM THE SUCKER RODS. IF THE SPEED OF THE SUCKER ROD STROKE WAS JUST ABOUT "0", IN OTHER WORDS, HARDLY MOVING, THE MEASURED MAXIMUM LOAD OF THE SUCKER RODS DURING THE DOWNSTROKE WOULD BE THE WEIGHT OF THE RODS FLOATING IN THE FLUID, AND THE WEIGHT OF THE RODS FLOATING IN FLUID ALONG WITH THE FLUID WEIGHT ON THE UPSTROKE. HOWEVER, ONCE WE APPLY DYNAMICS TO THE SUCKER ROD AND MOVE IT UP AND DOWN AT A GIVEN RATE, SOME OF THESE LOADS ARE ACCELERATING FORCES IN THE UPSTROKE WHICH WOULD INCREASE THE PEAK LOAD. THE RODS FLOATING ON THE FLUID IN THE DOWNSTROKE WOULD BE RESISTED BY THE FLUID AND HAVE A LESSER MINIMUM LOAD. ADDITIONALLY, FRICTION FORCES OF THE RODS RUBBING THE TUBING, PARAFFIN BUILD-UP, PACKING GLAND FRICTION - THESE THINGS WOULD CHANGE THE LOAD PATTERN TO A VERY IRREGULAR SHAPE.

AS THE LOAD CELL MEASURES THE LOADS AT THE VARIOUS POSITIONS OF THE STROKE, THIS INFORMATION IS FED INTO A COMPUTER. ALREADY IN THE COMPUTER YOU HAVE INSERTED ALL THE PHYSICAL WELL DATA; DOWNHOLE PUMP AND PUMPJACK PARAMETERS - WITH THESE KNOWN PHYSICAL PROPERTIES OF THE EQUIPMENT AND WITH THE MEASURED LOAD, THE COMPUTER AUTOMATICALLY DRAWS TO SCALE A GRAPH WHICH SHOWS THE DOWNHOLE PUMP LOADING CONDITION AND THE TOTAL ROD LOAD PATTERN.

BEARING SELECTION

DUE TO THE LARGE NUMBER OF OPERATING CYCLES, THE BEARINGS USED IN THE WRIST PIN, EQUALIZER AND SADDLE BEARING ASSEMBLIES SHOULD BE SELECTED ON B-10 BEARING LIFE. IT IS IMPORTANT THAT WE UNDERSTAND THE B-10 BEARING LIFE DEFINITION.

B-10 BEARING LIFE

HOURS OF LIFE FOR 20 YEARS =

24 HRS DAY X 300 WORK DAYS/YR X 20 YRS = 144,000 HRS.

B-10 LIFE MEANS THAT 90% OF ALL BEARINGS WILL EXCEED THE CALCULATED LIFE; 50% OF ALL BEARINGS WILL HAVE FIVE (5) TIMES THIS LIFE.

THE LIFE IS DIRECTLY PROPORTIONAL TO THE SPM AND IS INVERSELY PROPORTIONAL TO THE 10/3 POWER FOR LOADS. ON ROLLER BEARINGS, OR CUBED (3) POWER ON BALL BEARINGS.

IF A UNIT IS DESIGNED FOR 20 SPM AND ACTUALLY OPERATES AT 10 SPM; THEN:

B-10 BEARING LIFE = 20/10 TIMES 20 YEARS, OR 40 YEARS LIFE.

IF THE BEARING LOAD IS REDUCED BY 1/2, THE NEW LIFE WILL BE:

B-10 BEARING LIFE = $(2)^{3.33}$ TIMES 20 YEARS, OR OR 40 YEARS LIFE.

CONVERSELY, IF THE UNIT IS OVERLOADED BY DOUBLE, THE NEW LIFE WILL BE:

B-10 BEARING LIFE = $(1/2)^{3.33}$ TIMES 20 YEARS, OR 2 YEARS LIFE.

EXAMPLE - SIZE #456 OPERATING AT 10 SPM, DESIGNED FOR 16 SPM WITH A 20% OVERLOAD.

B-10 LIFE = DESIGN LIFE X 16/10 X $(1/1.2)^{3.33}$ = .87 X DESIGN LIFE

THE LARGER THE NUMBER OF BEARINGS USED IN A GROUP, THE HIGHER THE ODDS ARE OF A PREMATURE FAILURE. IT IS IMPORTANT TO SELECT THE CORRECT B-10 LIFE.

WITH THESE TWO PICTURES YOU CAN NOW VISUALLY SEE THAT YOUR DOWNHOLE PUMP IS OPERATING PROPERLY AND CHECK THE LOADING ON YOUR PUMPING UNIT.

THERE ARE VARIOUS COMPANIES WHICH CAN BE EMPLOYED TO WEIGH YOUR WELL AND TELL YOU IF THE DOWNHOLE PUMP IS PERFORMING PROPERLY, IF THE UNIT IS OVERLOADED, IF IT IS COUNTERBALANCED PROPERLY AND HOW MUCH POWER IT IS CONSUMING SO YOU CAN MAKE RECOMMENDATIONS TO SOLVE PROBLEMS.

TYPES OF GEARS

THE SIMPLEST OF ALL GEARS TO DESIGN IS STRAIGHT SPUR GEAR. AS IT CREATES NO SIDE FORCES, THE BEARINGS AND HOUSING ARE EASY TO DESIGN. AS THERE IS NO APEX IN THE GEAR TO WOBBLE, ALL THE SHAFTS CAN BE HELD FIRMLY IN PLACE. NO SHAFT IS REQUIRED TO FLOAT. THE DRAWBACK TO THE STRAIGHT SPUR GEAR IS LESS HP PER INCH OF FACE WIDTH AND HAS SLIGHTLY HIGHER NOISE LEVELS DURING OPERATION. THESE TWO PROBLEMS, THOUGH, CAN BE CORRECTED BY SHAPING THE TEETH SLIGHTLY DIFFERENT AND INCREASING THE GEAR SIZES A LITTLE BIT.

FROM THE SPUR GEARING EVOLVED THE HELICAL GEAR. THIS GEAR HAS MORE TEETH IN CONTACT AS IT OPERATES DUE TO THE TOOTH HELIX ANGLE WHICH CAUSES OVERLAPPING, THEREBY INCREASING THE TOTAL TOOTH SURFACE AREA IN CONTACT WHICH INCREASES THE HP

CARRYING CAPACITY PER INCH OF FACE WIDTH. BECAUSE IT HAS MORE TEETH IN CONTACT IT RUNS WITH LESS TOOTH CONTACT NOISE.

ADDITIONALLY, ALL SHAFTS CAN BE LOCKED FIRMLY IN PLACE; NO SHAFT IS REQUIRED TO FLOAT. THE DISADVANTAGE IS THE HORIZONTAL THRUST LOAD CREATED BY THE SLOPE OF THE TOOTH. THIS MUST BE RESTRAINED BY THE GEARBOX AND THE BEARINGS.

IN THE INTERMEDIATE SHAFT OF A SINGLE HELICAL GEARBOX, THE THRUST FORCES FROM THE INPUT GEAR AND THE THRUST FORCES FROM THE OUTPUT PINION ARE OPPOSING EACH OTHER AND ONLY THE REMAINING FORCE IS REQUIRED TO BE TAKEN BY THE BEARINGS AND GEARBOX.

IN A HERRINGBONE GEAR, THE CALCULATION IS EXACTLY THE SAME AS A HELICAL GEAR. THE ONLY DIFFERENCE IS THAT THERE IS NO THRUST TO BE TAKEN BY THE BEARINGS AND GEARBOX.

THE THRUST FORCES ARE TAKEN INTERNALLY WITHIN THE GEAR BUT BECAUSE OF THIS ACTION, ONLY ONE SHAFT IN THE GEAR TRAIN CAN BE ANCHORED. THE OTHER SHAFTS HAVE TO FLOAT BEHIND THE ANCHORED SHAFT.

IF YOU HAVE AN APEX WOBBLE DUE TO INCORRECT MACHINING, THE OTHER SHAFTS WILL NOW MOVE CONSTANTLY AND HORIZONTALLY TO LINE UP WITH THE ANCHORED SHAFT. NORMALLY, THE ANCHORED SHAFT IS THE OUTPUT SHAFT.

THE DOUBLE HELICAL GEAR IS THE SAME AS THE HERRINGBONE. THE END RESULT IS EXACTLY THE SAME AND YOU HAVE A HIGHER QUALITY LEVEL OF GEAR. A TRUE HERRINGBONE IS CUT ON A SYKES MACHINE WITHOUT A GROOVE IN THE MIDDLE. IF THERE IS A SMALL GROOVE IN THE MIDDLE, IT COULD STILL BE CUT ON A SYKES MACHINE AND BE A HERRINGBONE OR IT COULD BE CUT ON ANOTHER MACHINE AND BE A DOUBLE HELICAL.

A TRUE DOUBLE HELICAL AS WE THINK OF IT HAS A LARGE ENOUGH GAP IN THE MIDDLE TO ALLOW THE HOB CUTTER TO RUN OUT OF GEAR TEETH WITHOUT HITTING TEETH ON THE OTHER HALF OF THE GEAR. THIS GAP WOULD BE ANYWHERE FROM 1-1/2 IN. AND UP DEPENDING ON THE SIZE OF THE TEETH.

SINGLE HELICAL GEARBOXES ARE USUALLY USED IN THE PUMPING INDUSTRY ON BEAM BALANCED UNITS, IN SIZES UP TO 80. THERE ARE EXCEPTIONS, BUT THIS IS GENERALLY THE CASE.

THE HELICAL GEAR IS NORMALLY CUT TO AN AGMA QUALITY 8 TO 10 LEVEL AND THE HERRINGBONE IS NORMALLY CUT 5 TO 6. THE HELICAL GEARS WOULD BE A BETTER DRIVE AT HIGH SPEEDS DUE TO ITS HIGHER QUALITY.

TODAY, IN SIZES OVER 80, THE INDUSTRY IS GENERALLY USING HERRINGBONE BOXES, COMBINED HERRINGBONE AND DOUBLE HELICAL, OR FULL DOUBLE HELICAL BOXES.

THE API METHOD OF CALCULATIONING GEARBOXES DOES NOT DISTINGUISH BETWEEN HELICAL OR HERRINGBONE GEARS. IT IS COMPARABLE TO THE AMERICAN GEAR MANUFACTURERS ASSOCIATION CALCULATION WITH A BUILT-IN SERVICE FACTOR OF APPROXIMATELY 1.2 TO 1.3 AND IS A DURABILITY CALCULATION. THE SELECTION OF THE STRENGTH SERVICE FACTOR, WHICH IS NOT COVERED BY API, BECOMES EXTREMELY CRITICAL WHEN YOU VIEW THE POSSIBLE IMPACTS AND OVERLOADS WHICH CAN BE TRANSMITTED THROUGH THE GEARBOX FROM POSSIBLE WELL PROBLEMS. FOR EXAMPLE, SUCKER ROD PARTING CAN THROW AN OVERLOAD ON THE GEARBOX EQUAL TO THE TORQUE REQUIRED TO LIFT THE COUNTERWEIGHTS. THIS IS A POTENTIAL OVERLOAD OF 300 TO 400 PERCENT. EVEN SAFETY SWITCHES CANNOT ACT QUICKLY ENOUGH TO PREVENT MAJOR OVERLOADS ON SOME OF THE TEETH. ALL COMPONENTS IN THE GEARBOX MUST BE DESIGNED ON THIS SAME CRITERIA.

SOME OF THE BASIC CAUSES OF GEAR TOOTH WEAR AND FAILURE VARY FROM OVERLOADS TO MATERIAL OR DEBRIS IN THE LUBRICATION WHICH RUNS THROUGH THE GEARING. GEAR EXPERTS CAN REVIEW THE WEAR PATTERN ON THE GEAR AND DETERMINE THE TYPE OF FAILURE. IF THERE IS A FAILURE OF THE GEAR, YOU WOULD FIRST CHECK THE GEAR FOR THE PROPER MATERIAL AND PROPER ALIGNMENTS. IF YOU HAVE PROPER GEAR MATERIAL AND PROPER ALIGNMENT AND THE GEAR STILL FAILED, THIS FAILURE COULD BE CAUSED BY OVERLOADS FROM THE WELL, THE COUNTERBALANCE OR LUBRICATION PROBLEMS.

SOMETIMES WHEN THE GEAR FIRST GOES INTO OPERATION, THERE IS A LITTLE PITTING ALONG THE PITCH LINE. THIS IS THE MACHINING IRREGULARITIES FLAKING OFF. IF THAT IS ALL IT IS, IN A SHORT PERIOD OF TIME THE ACTION WILL STOP AND THE TOOTH WILL POLISH OFF.

LUBRICATION

MOST OF THE GEARBOXES HAVE AN OIL BACKUP SYSTEM WHICH SPLASH LUBRICATES ALL THE GEARS AND ALL THE BEARINGS. YOU SHOULD FOLLOW THE MANUFACTURERS RECOMMENDATIONS AS TO WHEN TO CHANGE OIL BOTH FOR CONTAMINATION AND FOR TEMPERATURE CHANGES.

THERE ARE THREE OTHER PLACES TO LUBRICATE. THOSE ARE THE SADDLE OR CENTER BEARING, THE EQUALIZER OR TAIL BEARING, AND THE WRIST PIN OR CRANK PIN BEARING. ADDITIONALLY, IT DOES NOT HURT TO PUT A LITTLE OIL ON THE WIRE ROPE.

NAME PLATES

API REQUIRES THAT YOU ADD A NAME PLATE TO THE GEAR REDUCER INDICATING THE TORQUE RATING, THE RATIO, THE SERIAL NUMBER AND THE NAME AND ADDRESS OF THE MANUFACTURER.

TO THE PUMPING UNIT STRUCTURE YOU MUST ADD A NAME PLATE WITH THE STRUCTURAL LOAD RATING IN HUNDREDS OF POUNDS, STROKE IN

INCHES, STRUCTURAL UNBALANCE IN POUNDS, SERIAL NUMBER, NAME AND ADDRESS OF MANUFACTURER.

GUARDS

API RP11ER IS A RECOMMENDED PRACTICE FOR GUARDING THE PUMPING UNIT - YOU MUST MAKE AVAILABLE TO YOUR CUSTOMERS THESE GUARDS.

- BELT GUARDS
- CRANK GUARDS
- FLYWHEEL GUARDS
- HORSEHEAD GUARDS

FENCES THAT GO COMPLETELY AROUND THE UNIT AND CAN COME EQUIPPED WITH GATES.

VARIOUS WARNING SIGNS SHOULD BE PLACED ON THE UNIT. UNITS WHICH ARE OPERATING ON TIMERS CAN BE VERY DANGEROUS TO AN ONLOOKER IF HE DOES NOT REALIZE THAT IT MIGHT START AT ANY TIME.

YOU SHOULD HAVE SIGNS WHICH WARN HIM OF THE DANGER AND GUARDS TO KEEP HIM AWAY.

IN SUMMARY, IT IS A VERY SIMPLE MACHINE IF YOU UNDERSTAND THE LOADING CONDITIONS.

IF YOU DO NOT, IT CAN BECOME A VERY COMPLICATED DEVICE.

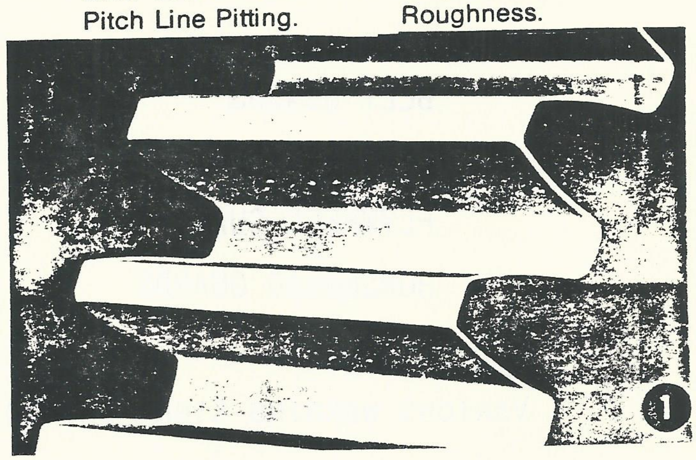
Basic Causes of Gear Tooth Wear and Failure

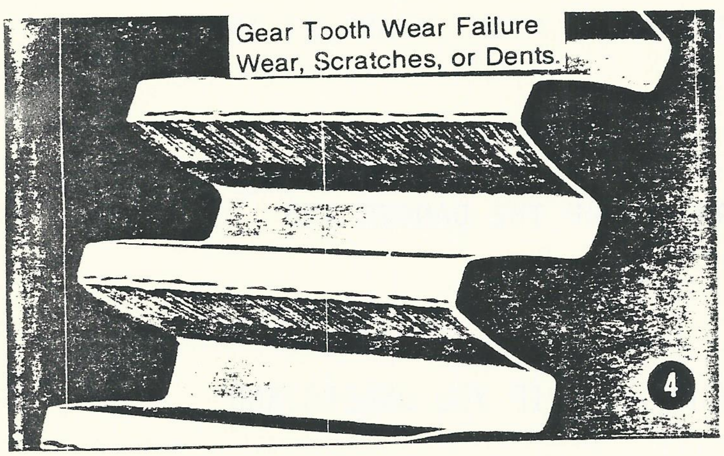
Shown on these two pages are typical illustrations of the five basic causes of gear tooth failures; although this is a much over-simplification of gear tooth failures, it might be of some value in solving problems in the field. The five basic causes are:

1. *Original surface roughness*
2. *Foreign matter in the lubricant*
3. *Mechanical damage*
4. *Metal failure*
5. *Lubrication failure*

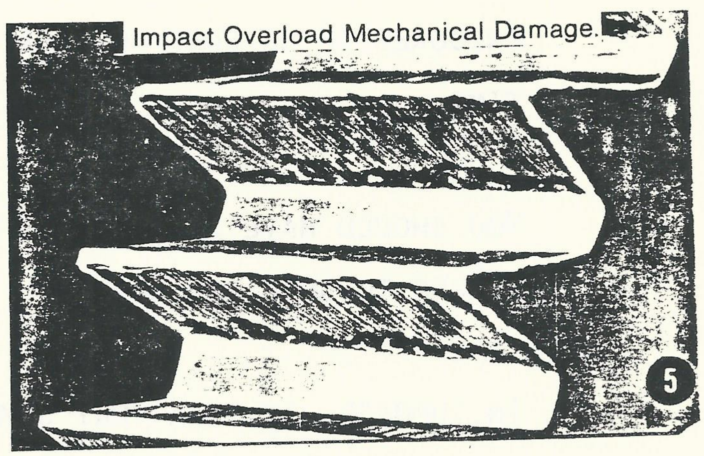
Gear Tooth Wear Pitch Line Pitting.

Original Surface Roughness.



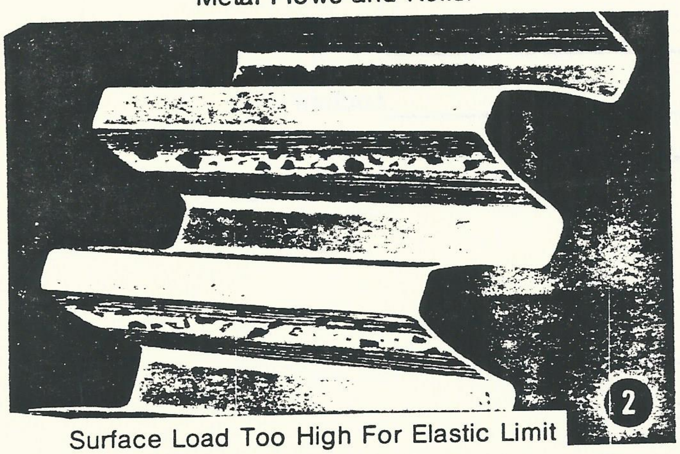


Foreign Material In Lubricant



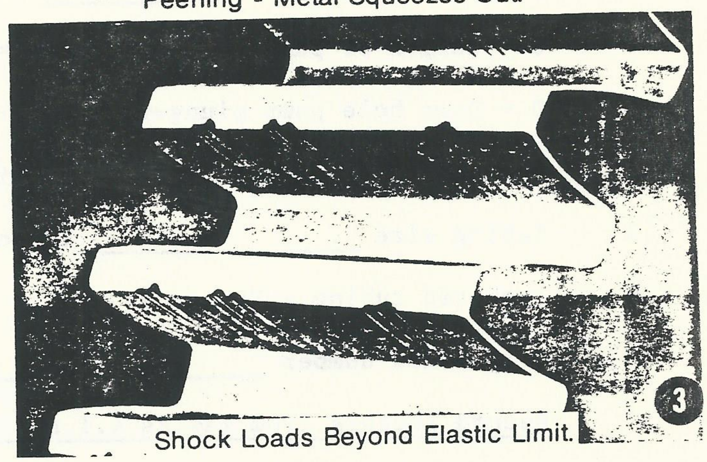
Gear Tooth Damage
Bent, Broken or Scarred Teeth.

Gear Tooth Failure
Metal Flows and Rolls.



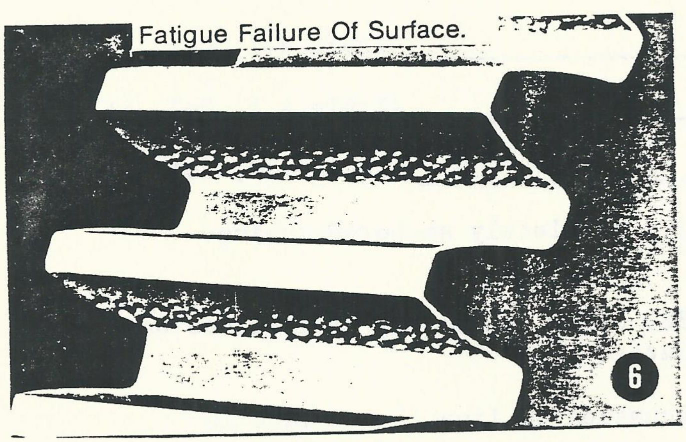
Surface Load Too High For Elastic Limit

Gear Tooth Failure
Peening - Metal Squeezes Out.



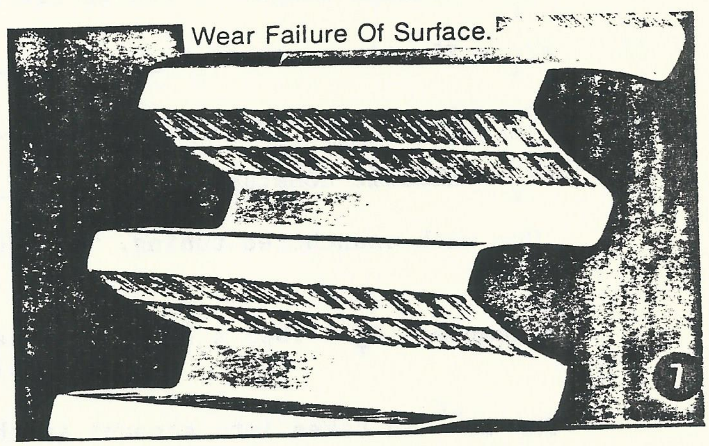
Shock Loads Beyond Elastic Limit.

Fatigue Failure Of Surface.



Gear Tooth Failure
Spalling.

Wear Failure Of Surface.



Gear Tooth Failure Both
Meshing Gear Surfaces Are Scarred.

DESIGN CALCULATION STANDARD

CONVENTIONAL SUCKER ROD PUMPING SYSTEM

(SAME AS EXAMPLE IN API RP11L CLARIFIED FOR EASIER USE)

OBJECT: To solve for - S_p, PD, PPRL, MPRL, PT, PRHP, AND CBE

Known or Assumed Data:

1 H = Fluid level = ____________________ ft.

2 L = Pump depth = ____________________ ft.

3 N = Pumping speed = ____________ SPM (Strokes Per Minute)

4 S = Length of pumping unit stroke = ____________________ inches

5 D = Down hole pump plunger diameter ____________________ inches

6 G = Specific gravity of pumped fluid = ____________________

7 Tubing size ________________ inches

8 Anchored tubing - Yes or No

9 Sucker Rod number ____________________

Record Factors from Tables 4.1 and 4.2:

10 W_r = Average weight of rod in air/Foot = ____________ (Table 4.1, Col. 3)

11 *E_r = Elastic constant = ____________________________ (Table 4.1, Col. 4)

12 F_c = Frequency factor = ____________________________ (Table 4.1, Col. 5)

13 *E_t = Elastic constant = ____________________________ (Table 4.2, Col. 5)

Use with unanchored tubing. Use "0" for completely anchored tubing.

*Example: $E_r = .883 \times 10^{-6}$ enter as .883 only, due to the limitations on hand calculators

The 10^{-6} is taken into account in the equation on line 15 and line 20.

Calculate Non-Dimensional Variables:

14 $Fo = 0.340 \times G \times D^2 \times H = 0.340 \times \text{Line \#6} \times (\text{Line \#5})^2 \times \text{Line \#1} =$

$0.340 \times$ ______ $\times$ (______)^2 $\times$ ______ = ______ lbs

(fluid load on plunger area)

15 $1/Kr = Er \times L = \dfrac{\text{Line \#11} \times \text{Line \#2}}{*(1{,}000{,}000)} = \dfrac{\_\_\_\_ \times \_\_\_\_}{*(1{,}000{,}000)} =$ ______ in per lb.

(Elastic constant = load required to stretch total rod string 1 inch)

$* \dfrac{1}{1{,}000{,}000} = 10^{-6}$ Because of limitations of hand calculators, division is the last operation.

16 $SKr = S \div 1/Kr = \dfrac{\text{Line \#4}}{\text{Line \#15}} = \dfrac{\_\_\_\_}{\_\_\_\_} =$ ______ (pounds necessary to stretch the rod string an amount equal to S)

17 $Fo/Skr = \dfrac{\text{Line \#14}}{\text{Line \#16}} = \dfrac{\_\_\_\_}{\_\_\_\_} =$ ______

18 $N/No = \dfrac{N \times L}{245000} = \dfrac{\text{Line \#3} \times \text{Line \#2}}{245000} = \dfrac{\_\_\_\_ \times \_\_\_\_}{245000} =$ ______ Constant

19 $N/No^1 = N/No - Fc = \dfrac{\text{Line \#18}}{\text{Line \#12}} = \dfrac{\_\_\_\_}{\_\_\_\_} =$ ______ Constant

20 $1/Kt = Et \times L = \dfrac{\text{Line \#13} \times \text{Line \#2}}{*1{,}000{,}000} = \dfrac{\_\_\_\_ \times \_\_\_\_}{*1{,}000{,}000} =$ ______ in/lb

(elastic constant for unanchored portion of tubing string inches per lb)

$* \dfrac{1}{1{,}000{,}000} = 10^{-6}$ Because of limitations of hand calculations, division is the last operation.

Solve for S_p and PD:

21 Sp/S = ______________(Constant - Fig. 4.1 = $\frac{\text{downhole pump stroke}}{\text{pumping unit stroke}}$)

22 Sp (down hole pump stroke) = (Sp/S x S) - (Fo x 1/Kt) =

(Line #21 x Line #4) - (Line #14 x Line #20 =

(________x________) - (__________x__________) = ___________________inches

23 PD = 0.1166 x Sp x N x D^2 = 0.1166 x Line #22 x Line # 3 x $(\text{Line \#5})^2$

0.1166 x ___________ x _____________ x $(\_\_\_\_\_\_\_\_\_\_\_\_\_)^2$ = $\frac{\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_}{\text{barrels per day}}$

Determine Non-Dimensional Parameters:

24 W = Wr x L = Total weight of rods in air = Line #10 x Line #2 =

_________________ x __________________ = ______________________#

25 Wrf = Total weight of rods in fluid = W x [1 - (.128 G)] =

Line #24 x [1 - (.128 x Line #6)] = ________ x [1 - (0.128 x __________)]

= ______________________lbs.

26 $\frac{\text{Wrf}}{\text{Skr}} = \frac{\text{Line \#25}}{\text{Line \#16}} = \frac{\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_}{\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_}$ = ________________________

<u>Record Non-Dimensional Factors from Figures 4.2 through 4.6:</u>

27 F1/Skr = ________________(Constant - Figure 4.2)

28 F2/Skr = ________________(Constant - Figure 4.3)

29 $2T/S^2Kr$ = ________________(Constant - Figure 4.4)

30 F3/Skr = ________________(Constant - Figure 4.5)

31 *Ta = ________________(Constant - Figure 4.6 - See example calc. below)

$$*Ta = \left[\frac{(Wrf/Skr - .3)}{.1} \times \frac{\_\_\_\_\_\_\_\_\%}{100} \text{ (from chart fig. 4.6)}\right] + 1.00$$

$$*Ta = \left[\frac{(\text{Line 26} - .3)}{.1} \times \frac{\_\_\_\%}{100}\right] + 1.00 = \left[\frac{(\_\_\_\_ - .3)}{.1} \times \frac{\_\_\_\%}{100}\right] + 1.00 = \_\_\_\_\_\_$$

Example: Line 25 = .600, from chart = 3%.

Given: Fo/Skr = .188

N/No1 = .200

$$Ta = \left[\frac{.600 - .3}{.1} - \frac{3}{100}\right] + 1.00 = 1.09$$

<u>Solve for Operating Characteristics:</u>

32 PPRL = Wrf + (Fl/Skr x Skr) = Line #25 + (Line #27 x Line #16) =

_______ + (_______ x _______) = __________ pounds peak polished rod load

33 MPRL = Wrf - (F2/Skr x Skr) = Line #25 - (Line #28 x Line #16) =

_______ - (_______ x _______) = __________ pounds minimum polished rod load

34 PT = $2T/S^2kr$ x Skr x S/2 x Ta = Line #29 x Line #16 x .5 x Line #4 x Line #31 = _______ x _______ x <u>0.5</u> x ______ x _______ = ____________ inch pound peak gear box torque.

35 *PRHP = F3/Skr x Skr x S x N x 2.53 x 10^{-6} = Line #30 x Line #16 x Line #4 x Line #3 x .00000253 =

_______ x ________ x ________ x _________ x .00000253 = ______ Polished Rod Horsepower

36 CBE = 1.06 (Wrf + ½ Fo) = 1.06 x $\left[\text{Line \#25} + \frac{\text{Line \#14}}{2}\right]$ =

$1.06 \left[\_\_\_\_\_\_\_ + \frac{\_\_\_\_\_\_\_}{2}\right]$ = ___________ pounds - required effective counterbalance at well.

*Prime mover HP:

Electric motor HP = PRHP x 2 = ____________x2 = ___________________

Gasoline engine HP = $\frac{\text{PRHP x 2}}{.85}$ = $\frac{\_\_\_\_\_\_\_\_\_\_\_\_ \text{x2}}{.85}$ = ___________________

PUBLICATIONS LIST

The following publications are under the jurisdiction of the API Committee on Production Equipment and are available from the American Petroleum Institute, 211 N. Ervay, Suite 1700, Dallas TX 75201.

SPECIFICATIONS

Spec 1B, Specification for Oil-Field V-Belting.

Covers standard and premium quality V-belts, dimensional and marking requirements on V-belt sheaves, recommended practices for power application of V-belts, and recommendations on care and use of V-belts.

Std 7B-11C, Specification for Internal-Combustion Reciprocating Engines for Oil-Field Service.

Covers methods for determining maximum brake horsepower and fuel-consumption rates of internal-combustion bare engines and power units and provides for the manufacturer's maximum-horsepower rating of such equipment for specific service applications.

Spec 11AX, Specification for Subsurface Sucker Rod Pumps and Fittings.

Covers a number of basic subsurface rod-type and tubing-type pumps in commonly used bore sizes.

Spec 11B, Specification for Sucker Rods.

Covers dimensional requirements on sucker rods, couplings, sub-couplings, polished rods and liners, stuffing boxes, pumping tees, stipulations on gages, gaging practice.

Std 11E, Specification for Pumping Units.

Covers designs and ratings of beam-type pumping-unit components.

Spec 11N, Specification for Lease Automatic Custody Transfer (LACT) Equipment.

Covers requirements for assemblies for the unattended automatic custody transfer (ACT) of liquid hydrocarbons, at rates below 11,000 U. S. barrels per 24-hour day, in field applications at less than 500 psig operating pressure.

Spec 11P, Specification for Packaged High Speed Separable Engine-Driven Reciprocating Gas Compressors.

Covers requirements for packaged high speed separable or belt driven oil-field type engine driven compressors in natural gas service. Includes specifications for auxiliary equipment and contains data sheets that may be used to set out service conditions, material requirements, acceptable vendors list, and fabricator's design sheets.

Spec 12B, Specification for Bolted Tanks for Storage of Production Liquids.

Covers material, design, and erection requirements for vertical, cylindrical, above-ground, bolted steel tanks in nominal capacities of 100 to 10,000 bbl. (in standard sizes) for production service.

Spec 12D, Specification for Field Welded Tanks for Storage of Production Liquids.

Covers material, design, fabrication, and erection requirements for vertical, cylindrical, above-ground, welded steel tanks in nominal capacities of 500 to 10,000 bbl. (in standard sizes) for production service.

Spec 12F, Specification for Shop Welded Tanks for Storage of Production Liquids.

Covers material, design, and construction requirements for vertical, cylindrical, above-ground, shop-welded, steel production tanks in nominal capacities of 90 to 500 bbl. for production service.

Spec 12J, Specification for Oil and Gas Separators.

Covers minimum requirements for the design, fabrication, and plant testing of oil-field type oil and gas separators and/or oil-gas-water separators used in the production of oil and/or gas.

Spec 12K, Specification for Indirect Type Oil-field Heaters.

Covers minimum requirements for the design, fabrication, and plant testing of oil-field type indirect heaters.

Spec 12L, Specification for Vertical Emulsion Treaters.

Covers minimum requirements for material, design, fabrication, and testing of vertical emulsion treaters.

RECOMMENDED PRACTICES

RP 11AR, Recommended Practice for Care and Use of Subsurface Pumps.

Covers recommendations for handling and using sub-surface pumps; the general characteristics of the pump types standardized in Std 11AX; inspection of pumps for compliance with API standards and assurance of good operating conditions; precautions in transporting and handling; operation; and proper procedures for assembly and disassembly of pumps.

RP 11BR, Recommended Practice for Care and Handling of Sucker Rods.

Covers recommendations on storage, transportation and running and pulling sucker rods.

RP 11ER, Recommended Practice for Guarding of Pumping Units.

Provides a reference or guide for the design, manufacturing, and installation of guards for oil well pumping units.

RP 11G, Recommended Practice for Installation and Lubrication of Pumping Units.

Covers installation of beam-type pumping units and lubrication of pumping-unit reducers.

RP 11L, Recommended Practice for Design Calculations for Sucker Rod Pumping Systems (Conventional Units).

Covers recommendations for design calculations for conventional unit sucker rod pumping systems based on test data submitted to API by Sucker Rod Pumping Research, Inc.

Form No. 11L-1, Design Calculations for Conventional Sucker Rod Pumping Systems.

A 100-sheet pad of a form to make design calculations in accordance with procedures set forth in RP 11L.

RP 11M, Recommended Practice for Grounded 830-Volt, Three-Phase Electrical System for Oilfield Service.

Provides a reference or guide for the design and installation of electrical equipment for grounded 830-volt systems at production facilities.

RP 500B, Recommended Practice for Classification of Areas for Electrical Installations at Drilling Rigs and Production Facilities on Land and on Marine Fixed and Mobile Platforms.

Classifies areas surrounding drilling rigs and production facilities on land and on marine fixed and mobile platforms for safe installation of electrical equipment.

RP 7C-11F, Recommended Practice for Installation, Maintenance, and Operation of Internal-Combustion Engines.

Covers field practices designed to promote efficient use of internal-combustion engines in oil-field service.

RP 12R1, Recommended Practice for Setting, Connecting, Maintenance, and Operation of Lease Tanks.

A guide for new tank battery installations and a guide for revamping existing batteries if this is necessary for any reason.

BULLETINS

Bul 11K, Data Sheet for Design of Air Exchange Coolers for Packaged Compressor Units.

Covers engineering data required for the design, rating, and purchase of air exchange coolers for packaged compressor units. A standard form for specifying the data is provided.

Bul 11L2, Catalog of Analog Computer Dynamometer Cards.

Contains over 1100 polished rod dynamometer cards taken with the electronic analog simulator arranged in convenient form for comparison with field tests.

Bul 11L3, Sucker Rod Pumping System Design Book.

Contains print-out tables of computer calculated values for selecting sucker rod systems. Values are included for depths of 2000 feet to 12,000 feet in increments of 500 feet and production rates of 100 barrels per day to over 1500 barrels per day in varying increments. Various rod string, pump stroke, pump size and pumping speed combinations that will do the job within the limiting parameters are listed.

Bul 11L4, Curves for Selecting Beam Pumping Units.

Contains 160 master curves for selecting beam pumping units, derived from the application of a computer program to portions of RP 11L. Included are curves for torque ratings of 57,000 through 912,000 in.-lb. from Table 3, Std 11E and for various stroke and rod designs.

TERMS AND DEFINITIONS FOR PUMPING UNITS

1. A-FRAME: CONSISTS OF FRONT TWO LEGS OF SAMSON POST ASSEMBLY, WHICH SUPPORT WALKING BEAM.

2. BEARING CARRIERS: BEARINGS IN GEAR REDUCERS ARE MOUNTED IN THEIR OWN HOUSING, WHICH IN TURN ARE MOUNTED IN GEAR REDUCER. THIS PROTECTS GEAR REDUCER.

3. BELTS: DRIVES GEAR REDUCER BY MEANS OF BELTING RUNNING ON SHEAVES OR PULLEYS MOUNTED ON PRIME MOVER AND ON GEAR REDUCER.

4. BELT GUARD: METAL COVER OVER THE BELTS BETWEEN PRIME MOVER AND GEAR REDUCER TO PROTECT PERSONNEL AND ANIMALS FROM INJURY.

5. BRAKE ASSEMBLY: USED TO STOP WELL FOR MAINTENANCE ON COMPONENT PARTS.

6. BULL GEARS: LARGE GEARS ON OUTPUT SHAFT.

7. CARRIER BAR: ATTACHMENT BETWEEN THE POLISHED ROD AND WIRE LINE HANGER.

8. COMPUTER SIZING: API CALCULATION HAS BEEN PROGRAMMED INTO A COMPUTER FOR SIZING PUMPING UNITS.

9. <u>COUNTERWEIGHTS</u>: ATTACHED TO CRANK ARMS BY BOLTS; ARE ADJUSTABLE ALONG LENGTH OF CRANK ARM, REDUCING LOAD ON THE GEAR REDUCER.

10. <u>COUNTER LOCKS</u>: COUNTERWEIGHTS CAN FALL IF IMPROPERLY INSTALLED. LOCKS CAN BE ADDED TO THE COUNTERWEIGHT TO PREVENT MOVEMENT OF COUNTERWEIGHT ALONG CRANK ARM.

11. <u>CRANK ARMS</u>: MOUNTED AND KEYED TO OUTPUT SHAFTS OF GEAR REDUCER, PROVIDING MEANS OF SWINGING WRIST PIN ASSEMBLY IN A CIRCLE ALLOWING PITMAN ARMS TO GO UP AND DOWN, OSCILLATING THE WALKING BEAM.

12. <u>CRANK ARM GUARDS</u>: USED TO PREVENT PEOPLE FROM WALKING INTO SWINGING CRANK ARMS.

13. <u>DOUBLE HELICAL</u>: SAME AS HERRINGBONE WITH A GROOVE IN THE MIDDLE, OR OPPOSED SINGLE HELICAL GEARS TO FORM ONE GEAR.

14. <u>EFFECTIVE COUNTERBALANCE</u>: MAJOR FORCE AT THE WELL CAUSED BY LOCATION OF WEIGHTS.

15. <u>EFFECTIVE WELL STROKE</u>: STROKE OF THE SURFACE PUMP, MINUS THE STRETCH OF SUCKER RODS AND/OR TUBING, WHICH BECOMES THE STROKE ON DOWN HOLE PUMP.

16. EQUALIZER BEAM: Connects TAIL BEARING ASSEMBLY to two PITMAN ARMS coming up from CRANK ARMS. Purpose is to equalize the load between PITMAN ARMS.

17. FENCES: Some units are completely enclosed in fence with locked gate.

18. FLUID: Crude oil consisting of oil, gas and water.

19. FLUID POUND: A drop in the well producing pressure and fluid volume which causes downhole pump to fill with gas and not with FLUID; can cause an obvious pound or shock on pumping system components.

20. GEAR REDUCER: Consists of a double reduction, approximately 30:1 gearbox. Purpose of GEAR REDUCER is to reduce input RPM of drive to 20 RPMS or less of the output. WALKING BEAM Oscillates at same SPM as the output shaft RPM.

21. HIGH PRIME: Configuration in which the PRIME MOVER which drives the GEAR REDUCER is mounted about 4 feet from ground off GEAR REDUCER support and is used where animals, flooding, or deep snow conditions exist.

22. HORSEHEAD: Pulley mounted on GEAR REDUCER and PRIME MOVER to control input speed to the GEAR REDUCER.

23. HORSEHEAD GUARDS: GUARD AROUND FRONT OF MACHINE WHICH PROTECTS PEOPLE OR ANIMALS FROM MOVING SUCKER ROD, WIRE LINE HANGER, AND HORSEHEAD.

24. LOW PRIME: CONFIGURATION IN WHICH PRIME MOVER IS SET DIRECTLY ON WIDE BASE OR, IN THE CASE OF THE T-BASE, IS SET ON SEPARATE PONY BASE FASTENED TO SAME FOUNDATION WHICH SUPPORTS THE T-BASE.

25. MARK II: THE MARK II IS A DIFFERENT GEOMETRY SYSTEM WHICH DOES NOT OSCILLATE WALKING BEAM ABOUT SADDLE BEARING ASSEMBLY AS A CONVENTIONAL UNIT DOES, BUT LIFTS WALKING BEAM BY EXERTING FORCE IN MIDDLE OF BEAM APPROXIMATELY WHERE SADDLE BEARING ASSEMBLY WAS, CAUSING BEAM TO LIFT AND OSCILLATE ABOUT REAR BEARING ASSEMBLY.

26. MOTOR RAILS: USED TO CONNECT PRIME MOVER TO WIDE BASE OR TO PONY BASE AND TO ADJUST PRIME MOVER LONGITUDINALLY IN ORDER TO TIGHTEN BELTS.

27. OIL DAMS: PLATES AT EACH GEAR REDUCER BEARING LOCATION WHICH HOLD THE OIL AT A LEVEL AT LEAST TO MIDDLE OF BOTTOM ROLLERS, TO PREVENT DRY START-UP.

28. OIL IN GEAR REDUCER: CONSISTS OF:

AN SAE 90 EP FOR 90° TO 140° F

-AND- AN SAE 80 EP FOR -30° TO -110° F

29. <u>OIL SCRAPER</u>: PLATE WHICH RUBS AGAINST GEARS IN GEAR REDUCER TO SCRAPE OIL FROM GEAR AND CONDUCT IT TO THE BEARINGS.

30. <u>PEAK POLISHED ROD LOAD</u>: COMBINED EFFECT OF THE WEIGHT OF SUCKER RODS, WEIGHT OF FLUID, AND ACCELERATION FORCES REQUIRED TO LIFT LOAD.

31. <u>PITMAN ARMS</u>: CONNECT CRANK ARMS BY MEANS OF WRIST PIN ASSEMBLIES TO EQUALIZER BEAM AND ARE USED TO PULL DOWN ON BACK OF WALKING BEAM THROUGH EQUALIZER BEAM AND TAIL BEARING ASSEMBLIES, CAUSING WALKING BEAM TO OSCILLATE AND LIFT WIRE LINE HANGER.

32. <u>POLISHED ROD</u>: A SPECIAL ROD OF THE SUCKER ROD ASSEMBLY WHICH SLIDES THROUGH PACKING GLAND AT TOP OF WELL, AND HAS BEEN GROUND AND POLISHED TO AVOID DAMAGE AT THE SEALS.

33. <u>PONY BASE</u>: LOOSE PRIME MOVER SUPPORT BASE MOUNTED ON THE SAME FOUNDATION AND USED WITH T-BASE.

34. <u>PRIME MOVER</u>: ELECTRIC, GAS OR DIESEL ENGINE WHICH PROVIDES ROTARY POWER TO DRIVE GEAR REDUCER.

35. <u>REAR LEG</u>: 3RD LEG OF SAMSON POST ASSEMBLY WHICH EXTENDS BACK TOWARD THE GEAR REDUCERS.

36. RPM: Revolutions per minute of the PRIME MOVER and of GEAR REDUCER input or output shafts.

37. SADDLE BEARING ASSEMBLY (or CENTER BEARING ASSEMBLY): Anti-friction bearing mounted on top of SAMSON POST ASSEMBLY, supporting WALKING BEAM. Function is to allow WALKING BEAM to oscillate.

38. SAMSON POST ASSEMBLY: Tripod which supports SADDLE BEARING ASSEMBLY which holds WALKING BEAM, consisting of REAR LEG and A-FRAME.

39. SHEAVES: Pulley mounted on GEAR REDUCER and PRIME MOVER to control input speed to the GEAR REDUCER.

40. STROKE: Travel of WIRE LINE HANGER caused by oscillations of WALKING BEAM.

41. SUCKER ROD: String of rods, usually steel, fastened together which connect down hole pump to surface pumping unit.

42. TAIL BEARING ASSEMBLY (EQUALIZER or EVENER BEARING ASSEMBLY): Mounted between EQUALIZER BEAM and tail or rear end of WALKING BEAM and is pulled down by PITMANS which exert a force on back of WALKING BEAM, causing WALKING BEAM to oscillate about SADDLE BEARING ASSEMBLY.

43. T-BASE: STRUCTURAL SUPPORT FOR GEAR REDUCER AND SAMSON POST ASSEMBLY. NORMALLY MOUNTED ON CONCRETE FOUNDATION.

44. THREE POINT GROUND LUBE: CENTER BEARING ASSEMBLY AND TAIL BEARING ASSEMBLY ARE LUBRICATED FROM GROUND BY MEANS OF HOSES AND FITTINGS.

45. TORQUE: TURNING FORCE OF GEAR REDUCER, OR TURNING FORCE OF COUNTERWEIGHT ON CRANK ARMS, ABOUT GEAR RE-DUCER OUTPUT SHAFT.

46. TORQUE FACTOR: CALCULATED NUMBER IN INCHES WHICH WHEN MULTIPLIED BY THE POLISHED ROD LOAD GIVES WELL LOAD TORQUE AT THE GEAR REDUCER AND INDICATES REQUIRED COUNTERWEIGHT TORQUE.

47. TUBING: DOWN HOLE PIPE THROUGH WHICH THE FLUID IS PUMPED TO THE SURFACE.

48. WALKING BEAM: WIDE FLANGE STRUCTURAL STEEL BEAM WHICH HAS HORSEHEAD CONNECTED ON ONE END; SETS ON SADDLE BEARING ASSEMBLY AND HAS A TAIL BEARING ASSEMBLY CON-NECTED TO EQUALIZER BEAM. ROTATION OF GEAR REDUCER CAUSES WALKING BEAM TO BE PULLED DOWN BY TAIL BEARING ASSEMBLY AND TO OSCILLATE ABOUT SADDLE BEARING ASSEMBLY.

49. <u>WELL CASING</u>: DOWN HOLE PIPE THROUGH WHICH THE FLUID IS PUMPED TO THE SURFACE.

50. <u>WIDE BASE</u>: STRUCTURAL SUPPORT, LARGER THAN THE T-BASE, FOR GEAR REDUCER, SAMSOM POST ASSEMBLY, PRIME MOVER AND BELT GUARDS, USUALLY SET ON TIMBERS AND GRAVEL.

51. <u>WIRE LINE HANGER</u>: CONSISTS OF A CASTING WHICH ATTACHES TO THE POLISHED ROD AND FLEXIBLE WIRE ROPE ATTACHED TO THE CASTING AND IS CONNECTED TO HORSEHEAD.

52. <u>WRISI PIN ASSEMBLY</u>: BEARING CONNECTIONS BETWEEN PITMAN ARMS AND CRANK ARMS WHICH ALLOW THE CRANK ARMS TO MOVE IN A CIRCULAR MOTION WHILE PULLING PITMAN ARMS DOWN.

BIBLIOGRAPHY

AGMA - AMERICAN GEAR MANUFACTURERS ASSOCIATION, 1901 NORTH FORT MEYER DRIVE, ARLINGTON, VIRGINIA 22209

API SPEC. 11AX, "SPECIFICATION FOR SUBSURFACE SUCKER ROD PUMPS AND FITTINGS".

API SPEC. 11B, "SPECIFICATION FOR SUCKER RODS".

API STANDARD 11E, "SPECIFICATION FOR PUMPING UNITS".

API RP11AR, "RECOMMENDED PRACTICE FOR CARE AND USE OF SUBSURFACE PUMPS."

API RP11BR, "RECOMMENDED PRACTICE FOR CARE AND HANDLING OF SUCKER RODS".

API RP11E, "RECOMMENDED PRACTICE FOR GUARDING OF PUMPING UNITS".

API RP11G, "RECOMMENDED PRACTICE FOR INSTALLATION AND LUBRICATION OF PUMPING UNITS".

API RP11L, "RECOMMENDED PRACTICE FOR DESIGN CALCULATIONS FOR SUCKER ROD PUMPING SYSTEMS (CONVENTIONAL UNITS)".

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TECHNOLOGY OF ARTIFICIAL LIFT METHODS, DR. KERMIT E. BROWN

THE UNIVERSITY OF TEXAS AT AUSTIN, TEXAS, PETROLEUM EXTENSION SERVICE, 10100 BURNETT ROAD, AUSTIN, TEXAS 78758.

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