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STATEMENT OF WORK

The project’s aim is to complete development of the Radial Cutting Torch, a pyrotechnic cutter, for use in all downhole tubular cutting operations in the petroleum industry. Project objectives are to redesign and pressure test nozzle seals to increase product quality, reliability, and manufacturability; improve the mechanical anchor to increase its temperature tolerance and its ability to function in a wider variety of wellbore fluids; and redesign and pressure test the RCT nozzle for operation at pressures from 10 to 20 ksi. The proposal work statement is included in the statement of work for the grant via this reference.

Task 1: Redesign nozzle seals.

A. Research alternative seal materials, designs, and manufacturers.
B. Make selection based on research findings.
C. Make drawing changes per the proper Engineering Change Notice (ECN).
D. Machine new parts per the new ECN and Q.C. parts.
E. Assemble RCT with new parts, leaving out fuel load.
F. Pressure test the RCT per Pressure Test Procedure.
G. Load the tested RCT with fuel mixture.
H. Retest loaded RCT and cut.
I. Repeat steps A through H three more times to test for repeatability.
J. Proceed to Task 2 if all tests are successful, otherwise, repeat testing.

Task 2: Redesign torch for operation at 20 ksi.

A. Optimize sleeve design to hold pressure at 20 ksi without deformation.
B. Make new drawings for the sleeve design.
C. Build parts per new specifications.
D. Design and build a test fixture to test the sleeve to 20 ksi then to failure to determine its upper limits.
E. Optimize sleeve design to shift at 20 ksi.
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F. Make drawings and build parts.

G. Test design until tool dynamics are operational at 20 ksi.  
   (Note: this is a proprietary procedure, which cannot be dis-  
   cussed in detail.)

Task 3: Upgrade mechanical anchor.

A. Investigate spring manufacturers for coil springs for high  
   temperature, corrosive environments, and greater force con-  
   stant.

B. Correct the drawings with an ECN.

C. Purchase/manufacture springs according to specification.

D. Investigate the bow springs for optimum thickness vs. force  
   requirement and use of carbide coated vs. non-carbide coated.

E. Correct drawings with an ECN.

F. Purchase bow springs per new specifications.

G. Redesign slip section to optimize the number, size, and angle  
   of teeth as well as the angle of diverter at contact point.

H. Correct drawings with an ECN.

I. Build new slip sections and diverters per new specifications.

J. Investigate the J-latch travel mechanism design:

   1. Select new materials for the pin to include higher strength,  
      ability to be surface hardened, and high Izod characteris-  
      tics;

   2. Use of high-strength ball and locking stud vs. pin; and

   3. Correct drawings/make new drawings to reflect the new speci-  
      fications.

K. Build new J-latch mechanism per new specifications.

L. Test completed anchor assembly in 2-3/8" and 2-7/8" tubing for:

   1. Force to lower anchor into the tubing/force to shift steeve  
      with the slips;

   2. Ease of shifting from no-set to set;
STATEMENT OF WORK

3. Ease of release from the set position; and

4. Force required to set and hold during torch operation.

Task 4: Reporting.

A. Prepare quarterly and final reports outlining the tasks performed and their respective results -- with selective, proprietary details omitted. Submit pictures of test results.
TASK 1: REDESIGN NOZZLE SEALS

Purpose: To redesign the seals and nozzle section to obtain a more effective sealing area for the improvement of operation of the RCT.

Objective: Task 1 was undertaken upon issuance of the D.O.E. Grant on September 2, 1994. Completion of this task will allow for the present torch design to reliably function as a standard manufactured product for all down hole conditions at pressures to 10,000 psi.
Procedure:

A. Research of Alternative elastomeric compounds

The first part of this task was to investigate the market for an appropriate material which could reliably withstand the following criteria:

1. Withstand temperatures to 500 degrees F
2. Be of sufficient durometer so as not to extrude at pressures of 10,000 psi
3. Must be compatible with aqueous salt solutions at temperatures of 500 degrees F
4. Must be compatible with a wide variety of hydrocarbons at temperatures of 500 degrees F

Should a suitable compound not be found, then I would have to resort to qualifying each torch based upon the application of the torch. This would be a very undesirable situation which would require detailed tracking measures for special type applications. This problem could be further compounded if a torch was shipped to a wireline company for a special application, then not used for that application due to a change in plans for that well. This torch would then be subject to a possible misapplication due to poor record keeping or tracking methods of the wireline company. The above example is a very real world problem facing the RCT in the present market place.

My original selection of a Viton fluorocarbon compound showed a great deal of promise. It has the broadest range of chemical resistance than any other compound. Viton is also rated to 450 degrees F for continuous duty service making it my first choice. I quickly learned, however, that Viton performs poorly in aqueous environments. Since, the large majority of oil wells have aqueous salt solutions in the well, the Viton compound quickly looses appeal. A 1-3/8" RCT was damaged beyond repair due to o'ring failure caused by hot aqueous environment at 15,300 ft. well depth.

Further investigation of an appropriate compound was made by inquiry of a wireline company in Louisiana which faces these type of field problems on a daily basis. Mr. Bill Boelte who for many years managed the New Orleans district for N.L. McCullough, Inc., the premier provider of chemical cutters to the petroleum industry was
queried for information. He was charged with the task of investigating elastomeric compounds for use on their Chemical Cutters. His research led to a Nitrile compound known as C-67. This compound has been used in general application for many years by manufacturers of chemical cutters. However, for my purpose this compound lacks due to temperature limitations of the compound which are further aggravated by high pressures which will be experienced down hole. Since the chemical cutter itself is also limited by these same factors, further research of acceptable elastomeric compounds was unnecessary.

Not being satisfied with the "industry standard" compound, further investigation of an acceptable elastomeric compound was continued. My research led to a fairly new compound called Aflas. This is a new fluorocarbon compound with a greater range of chemical compatibility and a higher temperature range than other fluorocarbon compounds. Its published resistance to steam and hot water make it very attractive for use down hole. The compound also has a temperature rating of 500 degrees F working and 600 degrees intermittent. This compound exhibits all the necessary and desirable qualities for a universal down hole o'ring for use as the standard on all RCT's (see Results section).
B. Machining, Assembly, and Testing

The selection of a suitable o’ring compound paved the way to solving the sealing problem associated with the RCT nozzle. Torch bodies which were built during the SIE Geosource contract were selected for testing of this new compound. Concentricity checks were made on the RCT. The o’ring groove, the body O.D., and the internal surface of the nozzle area were checked for concentricity. From this data, it was learned that the machining process that was followed to manufacture these torch bodies would produce a torch that would not be concentric enough to allow the sliding sleeve to seal. This lack of concentricity would also cause the sleeve to drag on the side of the o’ring surfaces of the body and the cap. This drag force also increases the forces necessary to move the sleeve during operation of the torch.

The drawings for all RCT designs were changed to reflect concentricity requirements necessary to maintain acceptable clearances for final assemblies. The drawings were also changed to add a second o’ring on the body and a second on the cap; this reflects the desire of oil company engineers for adequate down hole protection.

Parts were manufactured per the above changes, assembled and checked for fit, clearances, ease of assembly, and for drag forces. Once the checks were made, the torches were loaded and tested (see Results section).
Results

Section I.:

The final decision for an acceptable elastomeric compound is to use the 80 durometer Aflas fluorocarbon compound manufactured by Utex Corp. of Houston, Texas.

Section II.:

The assembly of the sleeve on all RCT designs was greatly improved. The addition of the second o’ring seemed to further stabilize the sleeve and aid during assembly. Not once were the o’rings found to be cut or damaged during the assembly process no matter how many times the sleeves were actuated. It appears that the lack of concentricity between the body and the cap played a very important part in obtaining a proper fit and a proper seal. Not one torch has leaked under pressure.

Once the mechanical torch assemblies were loaded, testing was begun. Trouble was experienced with the torches under pressures above 2000 psi. When fired the sleeve would either not slide clear or would slide only slightly enough to clear only one o’ring. This would cause a back flow in the torch which resulted in the bodies burning through or the nozzles would wash out due to an improper flow path. Many factors were weighed in this resulting phenomena before the answer was finally pinpointed. In order to add the second o’rings to the body and the cap, both o’ring sections had to be increased in length. Two problems were created with the addition of the second o’rings:

1. The increase in length of the o’ring section meant that the sleeve had to slide an additional 3/8" farther when fired; a 100% increase in travel distance.

2. The addition of two o’rings meant that the drag forces exerted on the sleeve by the o’rings were also doubled.

To overcome the adverse effects created by the addition of double o’rings on the sleeve, I had to resort to a redesign of either the chemical mix or the ignition system. I determined that the torch was not building internal pressure fast enough to overcome the external forces created by the well bore pressure, thus causing the sleeve to malfunction per the above described actions. I chose to
address the problem in the simplest fashion by increasing the output capacity of the thermal generator. This increase in the thermal generator would yield a much higher initial pressure pulse and produce a much faster initiation rate. Several torches were built and tested with the new design. All torches functioned properly and cut the tubing as designed. Not one failure of the torches due to improper initiation or sleeve malfunction has been reported.

Conclusion

From the testing performed, the field data gathered, and the ease of assembly, it appears that a very functional and practical torch design has been achieved. This design is also cost effective to manufacture. Although some work is still required to refine the operational design configuration, it appears that the stage has been set to achieve acceptable results on a torch to operate at pressures to 20 ksi. Much of this work must be achieved through extensive field use and obtained at actual down hole conditions.
III. Task 2: Redesign Torch for Operation at 20 ksi

No work has been started on this task as yet.
IV. Task 3: Upgrade Mechanical Anchor

Purpose: To redesign the Mechanical Anchor for improved shifting, to increase available drag forces, and to improve slip design for increased holding power during operation of the torch.

Objective: To produce a much improved version of a proven anchoring mechanism with greater reliability and functional ability and enhanced holding power.
Procedure:

To upgrade the mechanical anchor to achieve a universal field worthiness is a truly a long and experience oriented process. Many factors affect use of this type of tool for the full range field needs; some of which are:

1. The ability of the tool to set in coated tubing including plastic lined, cement lined, fiberglass lined, corroded, etc.

2. The ability of the tool to pass through the same pipe restrictions such as nipples, valves, collapses, etc. and still set and hold the cutter in place.

3. The ability of the tool to remain in down hole conditions for extended periods of time and still function.

The initial design has undergone many improvements and has successfully functioned in some limited down hole jobs. The tool has also failed to function in other applications. It is these failures which point to the most notable areas of the tool which require design improvement and change. The areas of the anchor which require the most immediate attention are the following:

A. The compression spring,

B. The shifting pin,

C. The slip section.

A. The compression spring:

This spring applies an adjustable compression force on the drag springs. The present spring is a standard spring steel material with a maximum load output of 100 lbs. This spring is insufficient in load output and most definitely loses much of this output as down hole temperatures rise. A decrease in this springs output will adversely affect the ability of the anchor to shift. A decrease in the shifting forces available will prevent the tool from setting and anchoring. The anchor must have available a sufficient force to set and release the tool. This coil spring is the source of this force.
The source for high temperature coil springs for operation in corrosive environments has been found. The same source can also supply the appropriate bow springs.

The original coil springs were an off the shelf item from a local supplier. The supplier did not have a spring of appropriate diametrical dimensions, so I used the closest size spring available which was a 1-1/2" diameter with a .162" diameter wire and 4" in length. The spring had a spring rate of 52 lb/in and a load of 73 lb. at the max. compression length of 2.6". The internal diameter of the spring was 1.176". The spring was housed on a shaft of 1" diameter which allowed for nearly 3/16" of play and distortion during compression of the spring. This particular spring was initially chosen because it was:

1. Of acceptable dimensional size,
2. Of tolerable force output,
3. A shelf item and readily available,
4. Very inexpensive compared to a custom manufactured spring.

A number of suppliers were investigated to find a standard size spring with the following specifications:

1. Outside diameter 1.688 max.
2. Inside diameter 1.06 governing dimension
3. Length 4.5" ground ends
4. Effective Length, L1 2.6" to 2.9"
5. Load 175 - 250 lbs.
6. Temperature rating 80% of load @ 350 deg F

A standard spring could not be located, neither could a close substitute be found. The majority of spring sizes were confined to wire diameters of approximately 3/16" or less and with Load P capacities of only 100 lbs. Heavy duty springs are available only in standard incremental sizes with controlled O.D./I.D. dimensions none of which are suitable for this application and Load P outputs far in excess of that desired.
A local manufacturer of compression springs was consulted. The above criteria was presented for evaluation and a spring with the following characteristics was recommended:

1. O.D. 1-11/16"
2. I.D. 1.250 "
3. Length 5"
4. Wire dia. .218"
5. Effective length, L1 2.83"
6. Temp. rating 500 deg. F
7. Load Rate 80.96 #/in
8. Material 302 Stainless Steel
9. Load 175 #
10. Cost $ 7.29 ea. min. of 35 units

A smaller I.D. was requested, but the load would diminish to 120# with a travel length of less than 1.9 inches. An I.D. of 1.25 inches would allow the spring to oscillate on the 1" shaft; a very undesirable characteristic. Yet to obtain an appropriate yield from the spring, the I.D. of the spring would have to remain at 1.25 inches. This would require the top nut P/N MAS-1687-014 and the adjusting screw P/N MAS-1687-021 to be modified with a shoulder and inner sleeve to house the 1.25 I.D. spring. This could be accomplished by redesigning both parts or by designing a single spacer which would accommodate both ends of the spring. I opted to design and build a single spacer to fix this problem. This would be the cheapest and easiest solution to the problem.

B. The shifting pin:

The original shifting pin was crudely made from a standard 1/4"-20 x 1/2" long socket set screw modified by machining the end opposite the socket. Performing this type of machining process on a set screw will work satisfactorily, but is expensive and requires specialized tooling and fixtures to maintain tolerances.

An investigation of a stock manufactured part to fill this requirement, uncovered a set screw known as a dog-nose socket set screw. These screws can be purchased with both coarse and fine threads and in standard lengths appropriate for my needs. The 5/16" size had a dog-nose diameter of .225 inches. This would work satisfactorily in the 1/4" wide J-latch, and would allow for the outer sleeve to rotate slightly on the main shaft.
A closer tolerance fit could not be achieved, so the 5/16" dog nose set screw was selected for use. The outer shifting sleeve P/N MAS-1678-020 (ref. dwg. #1 Appendix A) was revised to accommodate the 5/16" set screw.

C. The slip section:

The original slip section (ref. dwg #2 Appendix A) used a Slip Holder to attach the Slip Arms to the outer sliding sleeve components. The Slip Holder held the Slip Arms in position with an internal grooved cavity which functioned as a "socket" for the Slip Arms and which allowed a minimum of travel of the slip arms. The slips were made of tubing which was lathe machined to specification, then slit into six equal pieces. This design worked well for small tubulars such as 2-1/16" and 2-3/8" where radial travel was held to a minimum. Although the anchor design did function in 2-7/8" tubing, the anchor did possess the characteristic of sticking when run in the lighter weight sizes. Another problem with this design of Slip Arm, occurred in heavy mud and scaled wells. The debris from the well would coagulate beneath the extended arms, thus preventing them from retracting back into position for easy removal from the well.

Another problem associated with the original Slip Arm design was the small tooth configuration and the fixed profile of the tooth design which rendered the slip unable to fully engage the inside diameter of the various tubing sizes when extended. The result was anchoring ended up depending on the lower one or two slip teeth of each arm.

The first and foremost problem was to design a slip mechanism that could fully engage various tubing diameters when actuated. To accomplish this a lever arm with two pin joints was used (ref. dwg #3 Appendix A). The Slip Holder MAS-1688-015 Rev. A (ref. dwg #4 Appendix A) was redesigned to accommodate three pin joints for lever arms.

The Slips MAS-1688-017 (ref. dwg. #5 Appendix A) were redesigned to also accept a pin joint. To accomplish this design the number of slips were changed from six sections to three. This did not change the potential overall diametrical gripping area. Another important change in the slip design was made in the size of the slip teeth. The original design used a .062 deep x 60 deg. profile which yielded a .125 gap between slip teeth. The new design uses a .100 deep x 60 deg. profile which yields a .200 gap between the slip teeth. This larger
slip tooth design is far more effective in gripping the pipe and is much less likely to "plug up" with well debris. The larger tooth also provides a deeper cavity for the o’rings which are used as a stabilizing and return mechanism for the slips.

Results:

When all changes were completed and the parts were built and assembled on the anchor body, the new anchor design proved to be a marked improvement over the original design. Drag Spring adjustment was much improved with much higher drag forces available for setting and releasing the slips. The larger toothed slips with the lever arm attachments offered a great deal more holding force without compromising releasing ability.

Conclusion:

At this point, the anchor improvements seem to have provided a much improved anchoring tool which should aid in a greater success ration of cutting due to improved holding and operating forces.
APPENDIX A
DRAWING #2
2.425

2.000 TYP

1.750 TYP

0.800 TYP

0.100 TYP 3 PLS.

0.563 TYP

2.50 TYP

0.19 TYP

0.750 TYP

0.500 TYP

0.780 φ

1.688 φ

0.1"-12UNF-2 B
M.D. = .010/.028
.19 x 1.05 DIA. RELIEF

SECTION A-A

DRILL & TAP 2 HOLES
1/4"-20 UNC 3 PLS.
@ 120°

DRILL THRU 1/8" DIA. @ 120°
3 PLS. AS SHOWN SPOT
FACE AS NEEDED

MCR ENTERPRISES
P.O. Box 1465 Alvarado, Texas 76009

DRAWN BY

REVISED

DRAWING NUMBER

SLIP HOLDER 1-11/16"

ALLOY STEEL RC 45+

MCR-1688-D15
DRAWING #5
SECTION A-A

DRILL THRU 3 HOLES .125" DIA. SPOT FACE ENTRY AS NEEDED

.240 MILL SLOT 3 PLCS. @ 120°
SAW CUT 3 PLCS. @ 120° AS SHOWN
MAX. WIDTH OF CUT .125
DRILL THRU #29 (0.136)