BETTER UNDERSTANDING THE MAXIMUM TORQUE POTENTIAL OF PUMPING UNIT GEAR REDUCERS

N. David Ballard

Mobil Exploration and Producing U.S. Inc (MEPUS)

Summary

This paper discusses several studies undertaken by Mobil's Bakersfield, California, operating unit to better understand the operating limits of oilfield pumping unit gear reducers. Theoretic information was secured from several manufactures, both domestic and foreign, about gear reducer, gear and bearing life expectations. Limited testing was conducted in an effort to validate four manufactures' claims. Inspections were also conducted on about 200 pumping units to gain a wider information base on gear reducer operating life. From these data it was determined that some manufacturers' information could be proven while others could not. A compilation of the theoretical information was developed and reports were distributed that provide Mobil with the ability to more accurately determine the maximum allowable operating limits for several brands of pumping units.

Introduction

Mobil's Bakersfield operating unit embarked on a strategy to double the company's heavy oil production in California. Several hundred wells were planned to be drilled over about a seven year period. The wells are quite shallow with some having total depths of only 300 ft. The deepest wells have total depths of 2400 ft. with the majority of wells being between 1000 ft. and 1600 ft. deep. Anticipated fluid volumes ranged from 100 BFPD to 2200 BFPD. Pumping units in this service typically require long stroke lengths and low torque gear reducers. Anticipated well lives were between 7 and 15 years. The cost to equip new wells with pumping units was considerable. Several questions arose about how long could existing units be used and could lower rated gear reducers be purchased and expected to last for the expected project life.

The American Petroleum Institute (API) "Specification for Pumping Units 11E (SPEC 11E)" Sixteenth Edition, October 1, 1989 and American Gear Manufactures Association (AGMA) "AGMA Standard Practice For Helical and Herring Bone Speed Reducers For Oilfield Pumping Units" 422.03 May 1984 bulletins were reviewed. Various domestic and foreign pumping unit manufactures supplied data on torque limits for several sizes of gear reducers. Testing was planned and conducted on four different brands of pumping units over about an 18 month period of time.

The need for units to supply the new wells led to the purchase of new and used pumping units. All used pumping units were inspected and rated against other units that had operated in similar conditions.

Information from the tests and inspections has been utilized to determine what the operating load on gear reducers could be expected to be without experiencing failure. Also, this information has been used as a consideration in the purchase of additional pumping units.

Theory

API recommends operating limits for various size pumping units. API also essentially recommends an operating point for this rotating mechanism, this can be found on page 11 API Specification 11E. See Figure 1. Units with gear reducer torque ratings of 320,000 inch pounds and less are rated at 20 rpm for the output shaft. Equations that define operating limits for gear torque resistance, found in the API and AGMA bulletins mentioned above, indicate that an operating curve can be constructed.

The equations are as follows:

$$Tac = \frac{Np * d^2 * C_5}{2N_o} * \frac{F}{C_M} * I * \left(\frac{S_{ac}}{C_p}\right)^2$$

$$T_{at} = \frac{N_p * d * K_5}{2 * N_o} * \frac{F}{K_M} * S_{at} * \frac{J}{P_d}$$

$$C_5 = \frac{78}{78 + (v_t)^2}$$

$$v_t = d * N_p * .262$$

$$K_5 = \sqrt{\frac{78}{78 + (v_t)^2}}$$

Where:
T_{ac} = Allowable toque - pitting (in-lb)
T_{at} = Allowable torque - bending (in-lb)
N_p = Pinion speed - rpm
N_o = Speed of output shaft - rpm
d = Pitch diameter of pinion - inches

This curve is a function of output shaft speed. Speed related factors in the above equations are N_p, N_o, v_t, C_5 and K_5. A representative operating curve can been seen in Figure 2. Allowable torque for any given gear reducer increases as speed decreases. Each gear reducer from each manufacturer has its own unique curve for both pitting and bending resistance. Knowledge of this allows the operator to load a gear reducer somewhat above the API name plate torque rating at low speeds and not overload the gear train.

To apply these data properly one must know whether the weakest member of the gear train is bending or pitting limited. API manufactures gear data sheets give information on a number of facts associated with a particular gear reducer, such as torque resistances, material of construction, material hardness, bearing sizes, bearing L_{10} life, etc. Every manufacturer studied designs its gear reducers to have more torque resistance than the API name plate rating at rated output shaft speed. Some consider this a safety factor, others claim that this is an experience factor to achieve the API rating. This resistance could be called the design limit. See Figure 3. The difference between the API and design limit operating curves begs the question "can the gear reducer be safely operated above the API curve, but below the design limit?".

If gear reducers could be safely operated above the API curve it could mean that pumping units might not have to be moved as early as is now done after loads increase. Also, perhaps lower rated units could be purchased to preform at higher loads. The potential of this could save large sums of operating expenses and up front capital outlay.

An additional consideration with loading gear reducers is the reduction of bearing life. As load and rpm increase bearing life is reduced. These relationships can be demonstrated by equations found on page 406 of the SKF Industries, Inc. Product Service Guide 190-710 dated December 1988.

Bearings are rated at a L_{10} life, which represents the operating time at which 10% of the bearings will have failed at those specific load and speed conditions. Sets of bearings will have less life due to the statistics of having more than one bearing being analyzed. A family of curves can be generated for a set of operating rpms at various load conditions. Figure 4 represents a typical bearing set life plot. It assumes that the bearings are continuously loaded at constant conditions for various operating rpms. These plots will differ from manufacturer to manufacturer and from gear reducer size to gear reducer size. These plots can be derived using Miner's rule as set out in Appendix D, page 62, of AGMA 218.01, dated December 1982.

It also may be possible to estimate the remaining life of the bearing set at any other loading condition if prior loading conditions are known. This is done by using the known load and speed to determine bearing set life. Next determine the bearing set life under the new load conditions. Subtract the used life from the new condition life projection to estimate remaining life. Although this is possible, Mobil has not yet attempted to validate this.

Testing

Two separate limited tests were conducted to determine if gear reducers could in fact operate safely at torque loads in excess of the API name plate rating.

Test One:

The first test was conducted on a single pumping unit for sixteen weeks. Mobil Bakersfield was considering the purchase of a foreign pumping unit, but was unsure of the operability of the unit. A test procedure was agreed upon with the supplier and a unit was set on a producing well. The gear reducer's API operating and design curves were known. See Figure 5. The test was scheduled to be in three phases. Phase one was a three week break-in period with the unit operating below the API line at about 50% of design. The second phase was to be a three week period operating at 75% of design. The last phase was to be a ten week period operating the unit at ± 90% of design. All test phases were to be run at 8 spm. Test phases two and three were to have torque loads above the API operating curve. Torque loads were to be increased by increasing the counter balance effect, the manufacture was to provide the proper placement of the weights to accomplish the loadings. Torque analysis were conducted after start up and before and after each change in counter weight.

Phases one and two went according to plan and the unit was operated with torque loads near those anticipated. However, the torque analysis after moving into the third phase showed that the unit was operating in excess of the design curve. See Figure 5 week 6 and 8. After the unit operated for two weeks with this loading, the unit was shut down and the reducer was inspected. Some slight shallow pitting was noted on the second reduction pinion. This component was the weakest in the reducer and should have seen pitting at these loads. The rest of the gears showed no evidence of pitting. The manufacturer recalculated the counterweight placement, the weights were moved and the unit operated for fourteen weeks at torque loads near 90% of design and 167% of API name plate. Upon completion of the test the reducer was removed from the unit and disassembled in the supplier's yard. No additional damage was seen after the loads were reduced to a point below the design curve.

Test Two:

The second test included six pumping units of three different sizes and three different manufacturers. The test lasted for 14 months and included torque analysis and oil contaminant tests at the start of the test and every three months there after. The unit gear reducers were inspected once during the test and at the end of the test. The units were selected at random from wells that were known to be "heavy pumpers". During the test period pump size, strokes per minute, stroke length and counterweight placement were held constant for all the wells. Torque loads and iron contamination in the lube oil were plotted over time. See Figures 6, 7 and 8. These figures show a unit of each manufacturer. API and design lines are indicated for the operating spm of the units.

At the start of the test it was determined that the existing lube oil in each unit should have its lube oil filtered. Samples of the oil before and after filtering were sent to Mobil's St. Louis Lab. Filtration showed about a 50% reduction in contaminant levels which was still above desired limits to start the test. It was determined that all reducers be drained filled with a new charge of oil. Some of the subsequent oil samples indicated initially high rates of contaminants. This was attributed to improper flushing of the reducers by the contractor. The testing was continued to monitor the rate of contaminant increase with time. All units showed an increase in iron contamination throughout the test, indicating the removal of metal from the gears.

Inspections validated the lube oil contamination. Pitting was seen during the first inspection on the gear teeth of five reducers. The reducer that had no pitting, however, was the one that had the highest proportionate loading. Pitting increased dramatically from the first to the second inspection in the five reducers that had previously experienced pitting. The two reducers experiencing the most severe pitting were the newest units and of the same manufacturer. The remaining three pitted reducers were of the same manufacturer. The reducer having the largest proportional load still experienced no pitting. This reducer, however, demonstrated the highest gear face polishing.

These tests pointed out that not all manufactures use the same safety factors in designing their gear reducers. Recommendations have been made to not use two brands of the units above their respective API curves while two other brands can be used to loads between their API and design curves.

Inspections

Over a two year period, inspections were carried out on used pumping units that Mobil intended to transfer to or purchase for its heavy oil operations. Only two brands of units were inspected. A comparison was made of units that had most likely experienced similar loading in the same fields. One brand consistently showed less wear and pitting then the other. The brand that showed the least pitting had an 84% acceptance rate while the other brand had a 61% acceptance rate for the Bakersfield operations. It became evident that Mobil should focus its efforts on securing the brand that demonstrated the best inspection record.

Conclusions

1. Pumping unit gear reducer torque loading limits may be defined by an operating curve based on pitting and bending limit equations found in API and AGMA literature.
2. The gear reducer operating curve that is used should be the lowest value of either pitting or bending.
3. Each manufacturer will have a distinct curve for each size gear reducer.
4. Each manufacturer uses different design safety factors.
5. Some manufacturers gear reducers may be operated between their API and theoretic design curves.
6. Some manufacturers gear reducers may not be safely operated above their API curve.
7. Heavily loaded gear reducers should have periodic lube oil contaminant tests preformed. If the iron contamination level continues to increase over time the unit should be unloaded until the iron contaminant level stabilizes.
8. Inspections tended to validate the limited testing.

Acknowledgements

Mobil (MEPUS)
D. E. Coppersmith - Production Foreman
C. E. Lakey - Operations Foreman
J. A. Gonsalves - Production Technician
R. L. Palmer - Operations Engineer
M. Sipes - Engineer (Separated)

Mobil (Marketing & Refining)
C. D. Tanner - Sales Engineer

American
R. L. Webb - Sales Representative

Churchill
B. Van Matre - District Sales Manager (Deceased)

DARCO International
G. A. Darnell - President
M. W. Hale - General Manager

Jensen International
J. B. Jensen - President

Lufkin
D. Freeze - District Manager
C. Hunt - Chief Engineer

NASA
D. P. Townsend - Manager, Gear Research, Lewis Research Center

Representatives of AMPSCOT and Legrand

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

STROKES PER MINUTE, No	PEAK TORQUE RATING POUND INCHES
16	456,000
16	640,000
15	912,000
14	1,280,000
13	1,824,000
11	2,560,000 and larger

Figure 1 - From page 11, API RP 11e, October 1, 1989

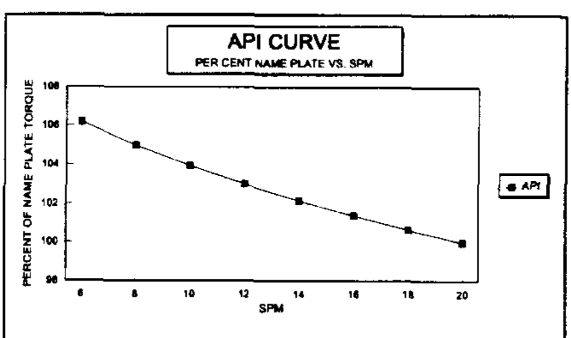

Figure 2

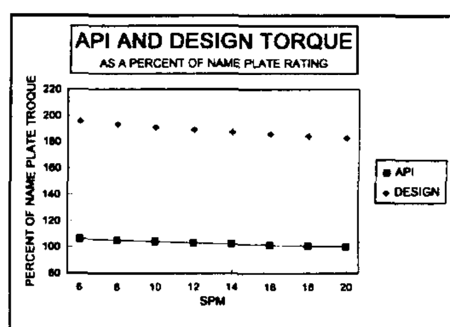

Figure 3

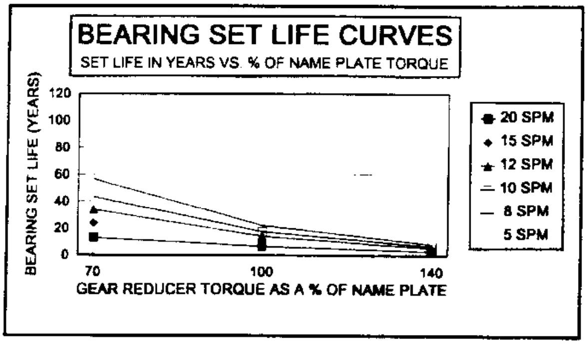

Figure 4

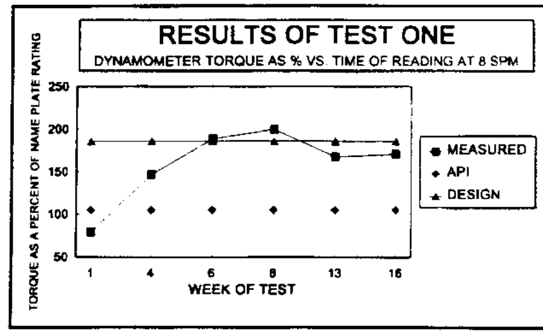

Figure 5

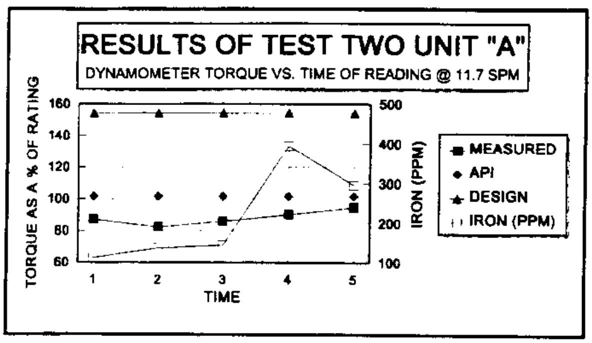

Figure 6

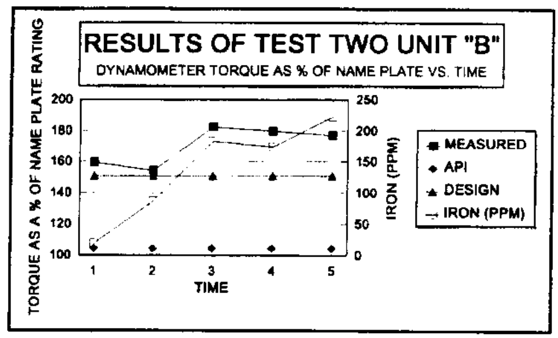

Figure 7

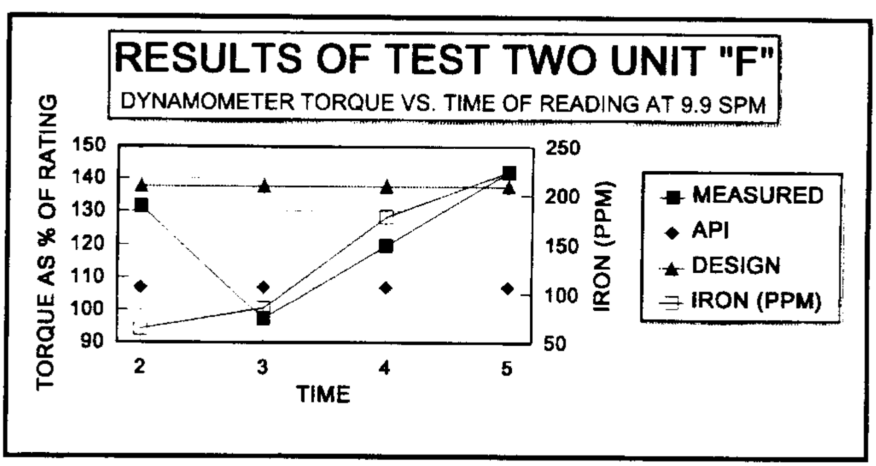

Figure 8