BUILDING AND TESTING A NEW TYPE OF VACUUM PUMP FOR CASING HEAD PRESSURE REDUCTION IN STRIPPER WELLS

JANUARY 17, 2007

FINAL TECHNICAL REPORT

JUNE 1, 2005
DECEMBER 31, 2006

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2936-WWVC-DOE-2098

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ABSTRACT

Note: This report is written assuming that the reader has previously become familiar with the Weatherbee pump design and concept of operation, which has been detailed in previous reports to the SWC.

This report summarizes results of our efforts for this Grant in seven categories:

- Category 1: design and build a new 8.5-inch pump
- Category 2: build a new test stand
- Category 3: lab test new pump
- Category 4: pump re-design and bearing load calculations
- Category 5: seal design for re-designed pump
- Category 6: build one prototype new pump with carrier ring to the outside
- Category 7: bench test newly designed pump with carrier ring to the outside

<u>**TABLE OF CONTENTS**</u>

EXECUTIVE SUMMARY

W&W Vacuum & Compressors, Inc. is developing the Weatherbee Wedge Pump, using new, patented technology, to enhance the production and economics of stripper wells for the independent operator.

There are two ways that vacuum pumps can help increase production in a stripper well. The first is to reduce the weight that wet flashgas exerts on the formation. The second is to reduce backpressure on the formation itself. By reducing the amount of gas down hole, pump gas locking problems are dramatically reduced. Although each formation responds differently, many mature basins have shown phenomenal increases in production with the use of vacuum assist. The Weatherbee Wedge Vacuum Pump is a ported device that can handle extremely high BTU gas without requiring the additional expense of multi-component systems, and has the potential for reducing equipments costs required to increase production, thereby improving the economics and productivity of stripper well production.

The objectives of the Proposal and research project were: to building a prototype model to determine the feasibility of using the Weatherbee Wedge Pump as a pressure reducing device on stripper wells in order to increase oil and gas production; design and build a test stand to simulate various wellhead conditions most common in stripper wells, and; bench test the prototype thus allowing automated real-time measurement of pressures and flow rates during operation of prototype pumps. Based on testing, we decided to apply design improvements and are currently building two new pumps with the improved design for testing in late February 2007.

EXPERIMENTAL RESULTS AND DISCUSSIONS

Category 1: Design and build of 8.5-inch pump. During the 1st Quarter, we designed and built one 8.5-inch pump at the facilities of Rogers Mfg., Inc., with the pump having the carrier ring to the inside. Molds were built for the 8.5-inch pump.

Task 2: Design and Build Test Stand:

An extra heavy-duty, low-center-of-gravity adjustable-height test stand was designed and built to simulate various wellhead conditions most common in stripper wells which allowed us to evaluate the vacuum pump's performance and energy requirements under various simulated conditions.

The test stand built included the following:

* 25 hp motor - built by US Electrical Motors
* Variable Frequency Drive - (EMS/CIMR - P5UL2018
* F22 flow controller (Omega/FC22
* Y temperature sensors - Omega/DP78
* X pressure sensors - Omega/DP18)
* Back pressure valve and vacuum valve

Category 2: Lab Test of 8.5-inch pump. The 8.5-inch pump built as described about was tested in vacuum service. Our original intent was to run this vacuum pump in the field. However, during testing we had problems with trade-off between throughput and overheating. Without using seals, the pump ran well, but we weren't getting enough throughput (at 500 rpm, we were getting about 20 SCFM), and the pump had moderate success as a vacuum pump (12"-15" vacuum). With the seals in place, the pumps could attain 27-28 inches of vacuum, but the pump would run too hot. The overheating was due to friction. This is different than the case for compression, where the heat is due to both friction and adiabatic compression. Materials tested for seals included Viton, Rulon, Teflon, Peek, and oil-impregnated polyurethane. We also tried other proprietary materials from Parker and Seely-Cook. Also, the geometries of the seals were fairly complicated. As a result of this complexity, the overheating in vacuum mode, and due to previous similar problems with the pump in compressor mode, we decided to make a major design change, and move the carrier ring to the outside. After the design change was implemented, we built two new pumps for testing and evaluation; a total of 3 pumps were built and tested even though the contract only required the building and testing of one pump.

Category 3: Pump Re-design and Bearing Load Calculations. An initial design concept of the pump with the carrier ring to the outside (with a hinge joint in the center of the pump in place of the interior ball with tracking ring) was provided to Glenn Wendell of SRI, who performed bearing load calculations. The results were used to send specification packages to five bearing vendors. Their feedback led to the conclusion that they could all handle loads with the carrier ring to the outside. Focus was then made on this new design, with load calculations performed on the hinge joint, pin joints that connect the secondary vane to the outside carrier ring, and main shaft that drives the primary vane. The design was also reviewed from the standpoint of manufacturability, tolerance accumulation and ease of incorporating seals in needed areas (e.g., hinge joint, etc.). Glenn also performed initial stress calculations to insure that components affected by the design changes were not subject to stresses beyond their capacity. The affected components and loadings included the input shaft, wedges, loading between secondary vanes and hubs, and loadings between primary vanes and hubs. An example output of these calculations is presented in Figure 1 below.

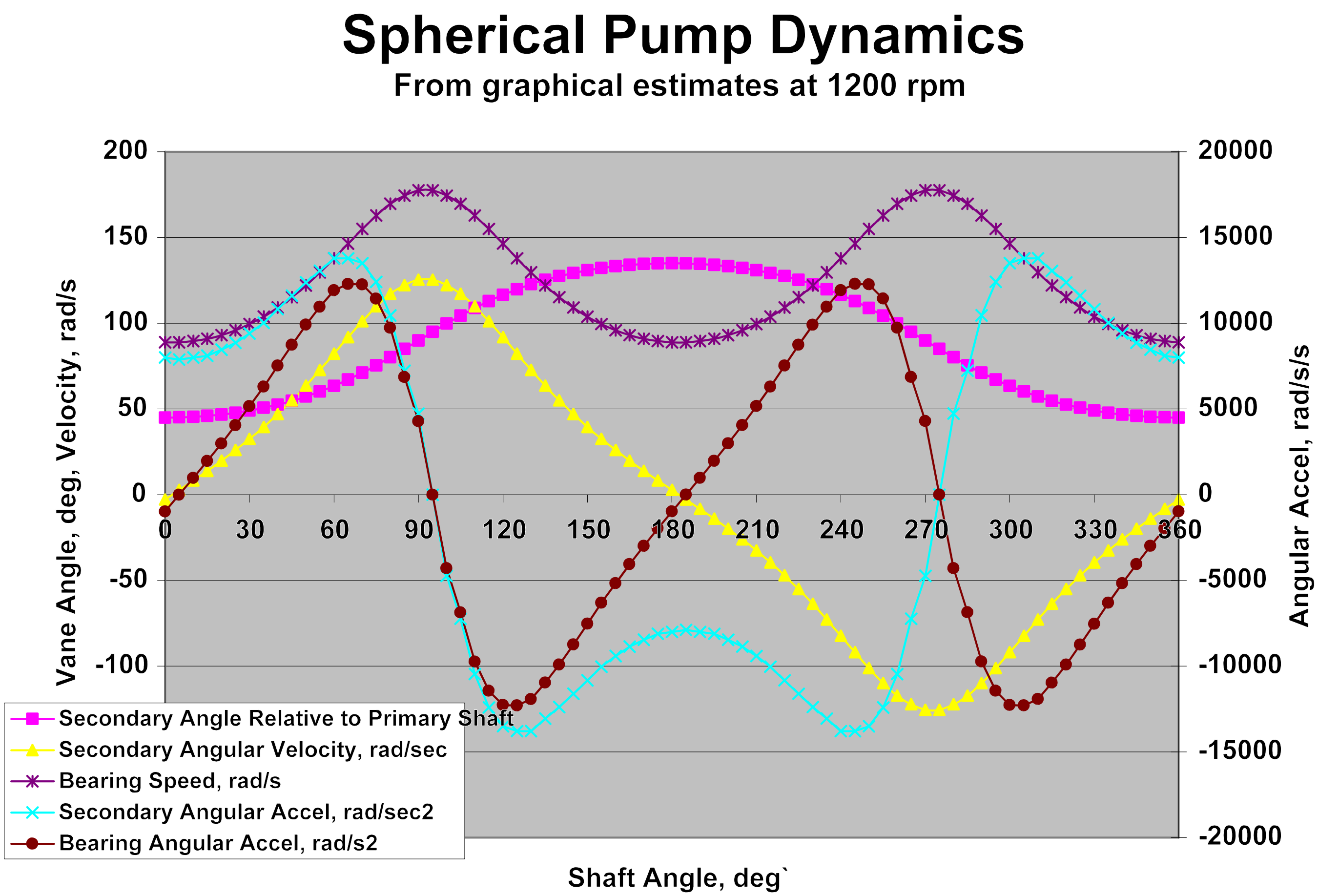

Figure 1

- Bearing Load Calculation Output for the Outside Carrier Ring

Bearing load calculations were at 50 psi and 100 psi, 800 rpm and 1200 rpm; most compressors in the oil field (positive

displacement) run from 600 rpm to 1200 rpm.
A Summary of the calculated bearing loadings and space requirements for various pump diameters are summarized in Table 1 below.

Table 1

Spherical Compressor Bearing Loading and Space Requirements

		5.0" Diameter Compressor		5.5" Diameter		6.5" Diameter		8.5" Diameter	
Bearing Moment, lb-in	Pressure, psig	Peak	Mean	Peak	Mean	Peak	Mean	Peak	Mean
	50	905	639	1145	832	1759	1339	4851	3297
	100	1840	1210	2376	1584	3709	2523	6835	5074
Speed, rpm		1697	1200	1697	1200	1697	1200	1697	1200
Bearing Envelope	Dimension Limits, inches	Internal Bearing Track	External Bearing Track	Internal Bearing Track	External Bearing Track	Internal Bearing Track	External Bearing Track	Internal Bearing Track	External Bearing Track
	ID Min	1.000	5.5	1.100	6	1.313	7	1.75	9
	OD Max	3.015	Open	3.32	Open	3.92	Open	5.125	Open
	Space Between Bearings, Min	Open	0.8125	Open	0.8125	Open	0.8125	Open	1
	Total external width, Bearings + Spacer, Max	0.625	2.25	0.75	2.5	1.85	3	2.41	4

The above results indicate that the bearings will hold up for all of the four investigated sizes.

Details of bearing designs investigated are provided in Figure 2 and Figure 3 below.

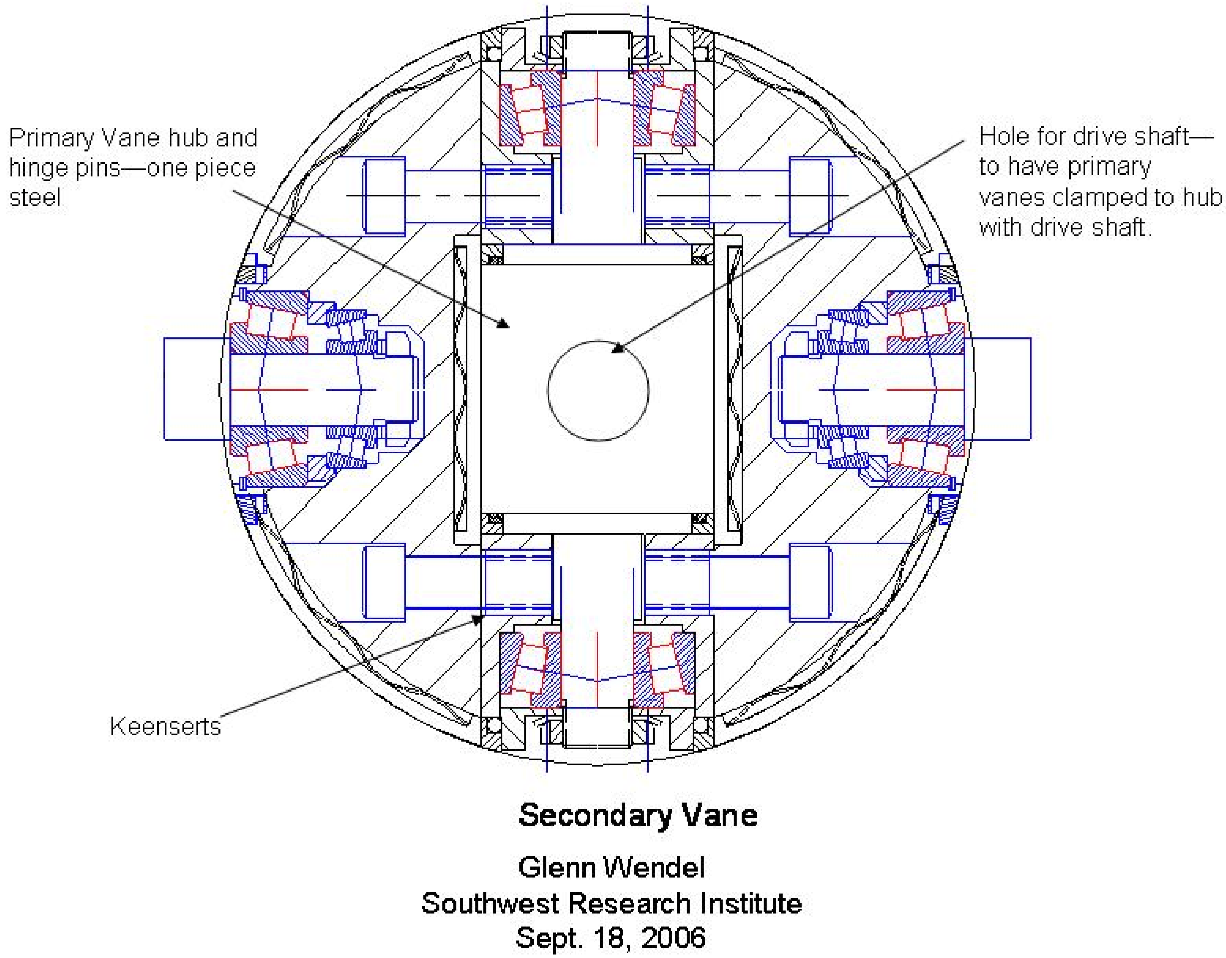

Figure 2 - Details of Components Investigated for Load Calculations

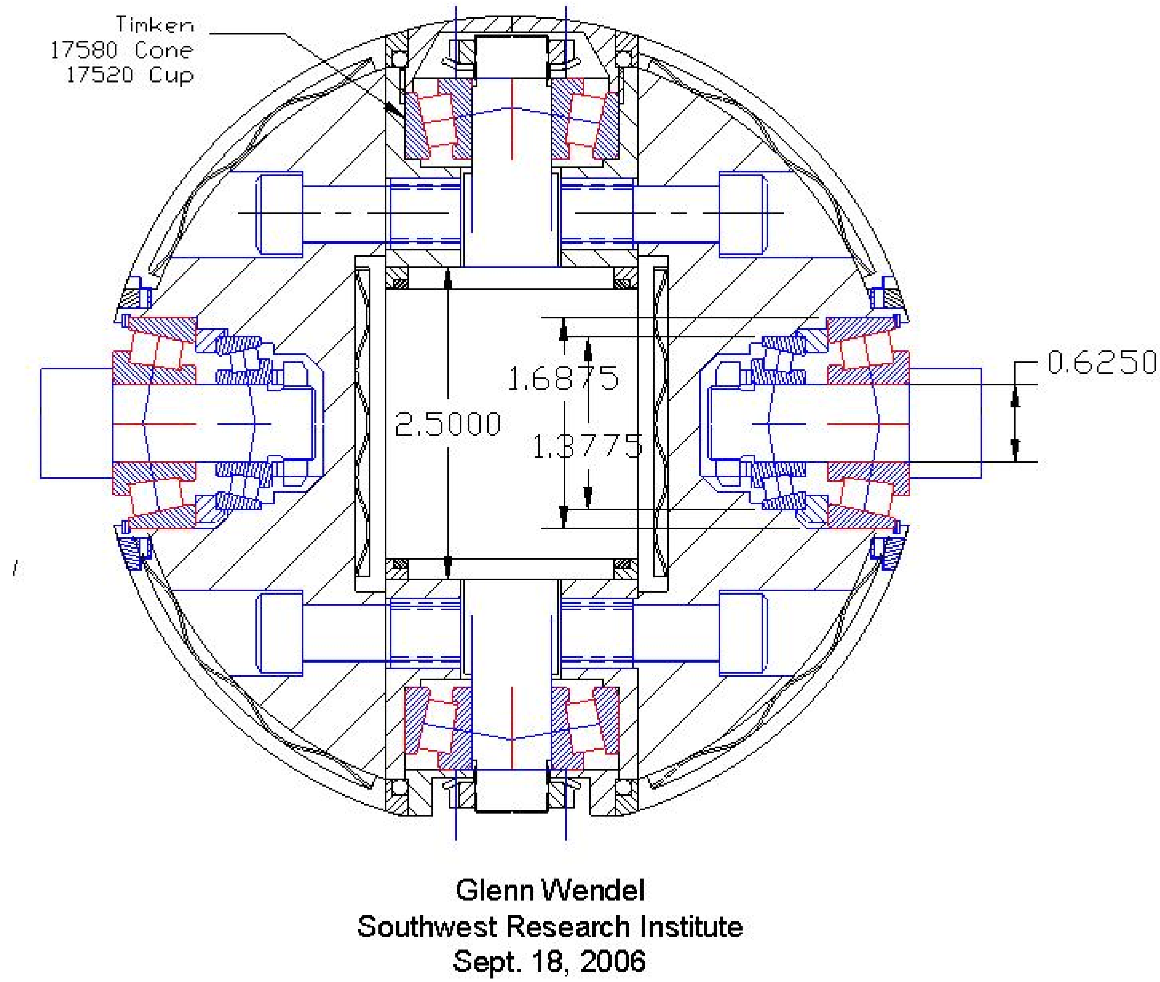

Figure 3 - Details of Components Investigated for Load Calculations

Category 4: Seal Design for Re-designed Pump. After considering the effects of pump ID, bearing spacing, energizing mechanisms, and potential for seal rotation, Glenn Wendell at Southwest Research Institute made preliminary conceptual recommendations for sealing a pump with the new design with the carrier ring to the outside. A conceptual sketch of exterior vane seals for a compressor with the new design is shown below in Figure 4.

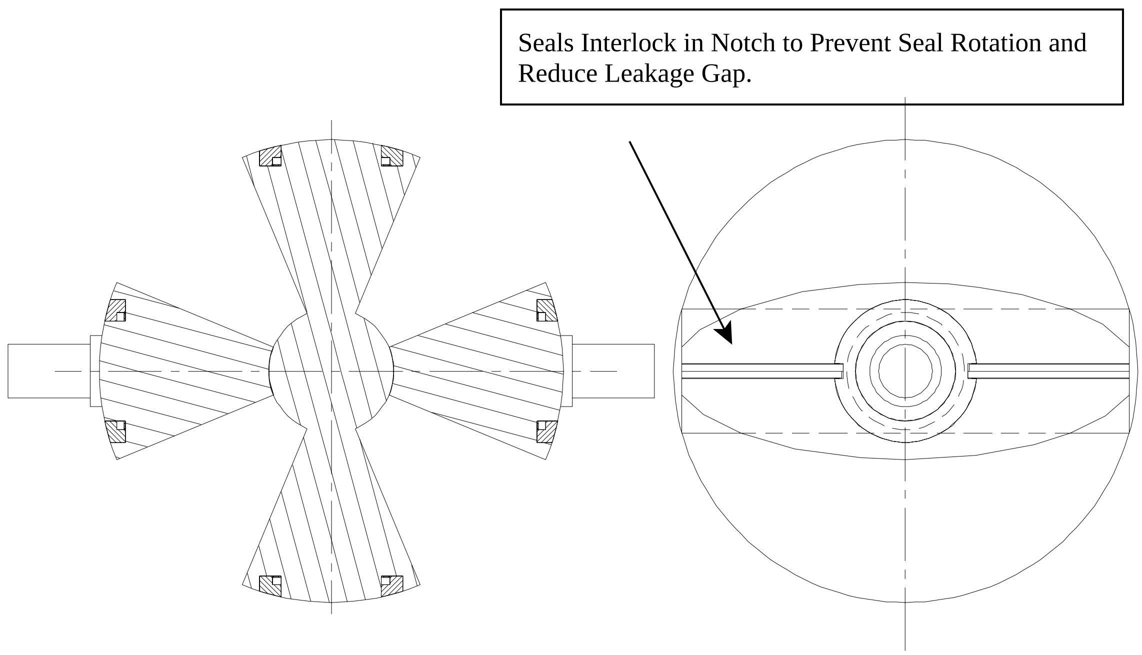

Figure 4 - Conceptual Exterior Vane Seal Design

The new pump design with the carrier ring to the outside will be easier to build and seal. Seal geometry will be simplified, manufacturing will be simplified, and in operation centrifugal force works with the components rather than against them. So for now, the design with the carrier ring to the inside has been postponed for future development. SRI has verified that taking the carrier ring to the outside would work.

The preferred seal design utilizes an energizer to close a gap and seal across a differential pressure in such a way that the differential pressure then helps hold the seal closed. Sealing is usually a compromise of completeness of seal and friction created by the seal. The approach being taken now emphasizes minimization of friction to reduce heat, while maintaining as much sealing capability as possible. The intention was to try to come up with a seal arrangement where every possible path of fluid leakage had some energized seal obstruction, which would also stay energized during operation of the pump. The intent of the seal design, all seals in the new design are energized at startup of the pump, and then centrifugal forces and pressure take over to maintain load on the seals during operation. The planned seal materials iron, PEEK, and Torlon, among other potential materials. All seals interlock or butt up to other seals to reduce fluid leakage paths.

Category 5: Fabrication of Two Pumps with New Design. Two new pumps with the carrier ring to the outside were designed and built according to the new design inputs above. One of these was demonstrated in tandem with a pump of the old design at a recent Stripper Well Consortium conference in Pennsylvania.

<u>Task 3:</u> Testing of Prototype:

Due to the necessity of implementing a major design change, it was not possible to bench test the new design prior to the December 31, 2006 deadline. However, the new designs are currently being tested on the test stand with promising results.

CONCLUSIONS

Conclusion. Further prototype testing indicated that the current design needs modifications to deal with heat prior to full-time operation in the field. In-depth investigation of alternatives for seal and bearing improvement indicate that a design with the carrier ring to the outside would be more robust and simpler to manufacture. Currently, the build of prototypes with the carrier ring to the outside are under way. Design efforts of pumps with carrier ring to the inside are not lost, for example this pump could be used as a multi-phase pump.

REFERENCES

None