HIGH HORSEPOWER CRANKSHAFT PUMP
OPTIMIZES FRACTURING OPERATIONS

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ABSTRACT

Development of a 2000 horsepower fracturing pump unit has brought greater safety, more reliability, more sand pumping capacity, reduced manpower requirements, and shorter setup time to fracturing operations. The pump is a crankshaft type, powered by a 16-cylinder diesel engine, and driven through an eight speed transmission.

For job control, the unit is provided with remote controls and a specialized pressure limiting system. The fluid end has been optimized for handling fluids with high sand concentrations.

The paper describes the design and gives pertinent specifications of the pumping unit. Selected case histories from more than a year of operation are presented, including accounts of stimulation jobs where record-high sand concentrations were pumped. Job design considerations peculiar to the use of this large-scale pumping capability are presented.

INTRODUCTION

A fracturing trailer forms the backbone of a well-servicing fleet. It primarily consists of pumps, associated prime movers, and drive trains. The early prime movers (in 1940's) were steam engines, which have been gradually replaced by modern Diesel engines. An attempt made in the 1960's to employ the more compact and weight-efficient gas turbines did not succeed owing to high initial cost, high operating costs and difficulty in maintenance.

Fracturing pumps have historically been of the crankshaft type, beginning with a simplex design and evolving through a duplex and then finally to the modern triplex configuration. A relatively small number of intensifier type pumps have also been developed. These utilize lower pressure hydraulic horsepower of other pumps to develop high pressures (in excess of 10,000 psi).

With the well depths increasing, there has been a growing trend in the petroleum industry to employ higher horsepowers in fracturing applications. This, combined with the economic pressure of operating costs, has led to the need for a single pump, high horsepower fracturing unit. This paper describes such a unit.

TRAILER

Figures 1 and 2 show a 2000 HHP trailer unit. It consists of a horizontal, single acting, crankshaft type triplex pump. It is powered by a General Motors 16V-149TI Diesel engine. It is a two-cycle, turbo-charged and inter-cooled engine which develops 2000 BHP at 2050 RPM. An eight speed, direct-mounted Allison CLT-9880 power-shift transmission provides the pump speed controls. The torque converter on the transmission is equipped with an automatic lock-up for efficient operation in the direct mode. A viscous dampener is employed in the drive train to dampen torsional vibrations in the drive train, which lowers vibration-induced stress levels in the transmission and engine components. Air-to-oil coolers are employed to cool the transmission and pump oils. All of these components are mounted on a standard gooseneck type trailer frame which is weight-legal in almost all states. An adjustable maximum pressure-limiting system is also employed which helps provide protection to the well, equipment and personnel.

Design considerations involved in the trailer are varied in nature and are briefly described here. All major components (engine, transmission, pump and coolers) are mounted onto the trailer with a "three-point" mount system. These joints allow for trailer frame deflections without creating high stress amplitudes during roading and operation. The location of various major components on the trailer was determined through a computer model of the trailer structure. Minimization of the static and dynamic stresses was used as the design criterion, while meeting highway regulations for maximum tire loading, length, width and height constraints. Inlet and discharge manifolding have been designed to facilitate easy job set up. This has been accomplished by incorporating a sufficient number of swivel joints, and by locating the inlets and outlets at easily accessible points on the trailer. One of the major design considerations, other than the cost and performance, was the total unit weight. The unit has been designed to have minimum weight per horsepower produced.

PUMP

General Considerations

Design of the pump has been based upon overall economy of operation without compromising reliability. Past approaches have primarily concentrated on initial cost without any regard to the total operating cost.[1,2] In Reference 1 a short stroke, high speed design is advocated. From initial cost and weight considerations, that is a good design. But from total operating cost and reliability considerations, a pump with relatively large displacement volume and slow speed in general is a better choice. Specific displacement volume and speed depend upon the job profile a pump will go through over its life span. This duty cycle will vary for each division, so an overall optimum for this service company as a whole had to be selected. This has resulted in a 12 in. stroke design. The large displacement, slow speed pump imposes a lower number of cycles on the pump expendables (valves, valve seats, inserts) and on the fluid end itself. This contributes to the higher reliability of the equipment. The slower pump speed does not result in poor performance while pumping solid-phase slurries as claimed in Reference 1. Cycle time (even at the lowest pump speed) is considerably smaller than the typical settling time for gelled slurries (typically the settling time is on the order of 10 seconds, determined from settling velocity tests).

The horizontal triplex, crankshaft type pump was designed based on computer modeling. These models are comprised of finite element models of various substructures of the pump, and also of the kinematics and dynamics of the slider crank mechanism. A direct result of the computer analyses is an offset slider crank mechanism that results in better loading of the pump power end structure.

Power End

The power end design incorporates force feed lubrication to all bearings (roller and journal type). This includes the crosshead shoes which normally do not receive lubricant through the entire stroke. This has been accomplished through internal passages in the crosshead and shoes connecting them to the lubricant supply through the crankshaft.

Power end design is also based on the general philosophy of direct load bearing. Load transfers across the power end sections are minimized, which helps reduce cyclic stresses in the welds. Further, the number of welds and the shimming requirements have been minimized.

Fluid End

The fluid end design is based on an empirical data base accumulated over the past 40 years of operation in the oil field. It primarily allows a prediction of the fatigue life of a fluid end with autofrettage. Aircraft quality alloy steel is used in the monoblock fluid end design. The monoblock feature results in lower stress levels compared to the three-sections approach. To help prevent erosion, valves and internal passages have been designed to minimize fluid velocities. Lower velocities also minimize NPSH (net positive suction head) requirements. Due to the relatively large size and the associated cost of the fluid end, it is imperative to minimize the cyclic stress levels. This has been accomplished through hydraulic preloading of threaded connections.

While pumping sand-laden slurries sometimes at high sand concentrations, the high pressure pumps "sandout," i.e., the fluid end gets filled up with an excessive sand volume, thereby locking the plunger. This normally results in severe damage to the pump components. A "protective cover" concept has been developed to minimize damage to the pump in the event of a sand-out. It also provides a higher degree of safety to the operating personnel. The inherent fluid end design has been affected to minimize dead spaces and right-angled turns, which result in improved sand pumping capability.

Through computer modeling, lateral deflections of the fluid end relative to the power end have been minimized, which improves plunger packing life.

LAB TESTS

Static tests were run to determine stresses at a few critical points in the pump structure. This was used to validate the theoretical predictions. Following static tests, running tests were conducted at various speeds and pressures. These early tests indicated problem areas in the speed reducer and in a few minor components. These components were redesigned and corrected. A different speed reducer was used which performed satisfactorily. Tests conducted gathered information to evaluate the thermal behavior of the pump, vibration characteristics related to cavitation, and pump performance while pumping sand. As high as 34 lb/gal (10-20) sand concentration was successfully pumped in the lab.

FIELD EXPERIENCE

Two prototype units have been in the field for a total of five years (both units combined). During this time other than normal wear of expendable parts (valves, valve seats, inserts and packing) only one journal bearing has failed prematurely. The reason for this failure has been traced to a material flaw in the bronze material used in that bearing. The units have been run at their maximum rated horsepower and at a maximum of 20 lb/gallon sand concentration. Figure 3 shows a typical pressure and sand concentration trace of a job performed by a single unit.

JOB DESIGN CONSIDERATIONS

A high horsepower single pump unit possesses the following characteristics:

1. It inherently does not offer a standby capability. Hence, for single unit jobs where the flow cannot be interrupted, adequate standby power should be available at the job site.

2. The job control function is carried out more easily because the number of pump operators involved is less. However, this also makes the operator performance more critical.

3. Due to the high flow rate capability (in excess of 20 bbl/min), suction and discharge lines should have adequate flow areas to help prevent erosion due to excessive fluid velocities.

4. If a single unit is used at low speed, resultant fluctuations in flow rate may adversely affect flow rate and rheology instrumentation.

5. If a single unit is used on a job, output characteristics (Fig. 4) of the unit should be consulted to determine feasibility of pumping. Simply calculating the hydraulic horsepower required is not sufficient to guarantee feasibility of pumping.

CONCLUSION

The high horsepower pump and unit designs appear technically sound, although based on concepts (large displacement - slow speed) which are radically different from past thinking. Field experience up to this point tends to validate assumptions made in modeling.

ACKNOWLEDGMENTS

A large number of people have materially contributed to this project. Their efforts are gratefully acknowledged.

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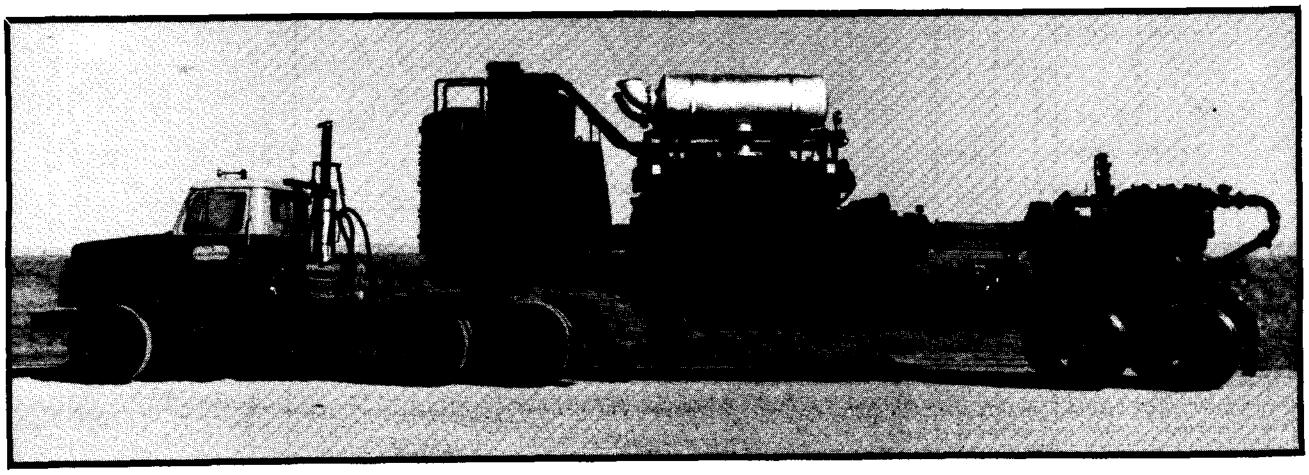

Figure 1 - Fracturing trailer - side view

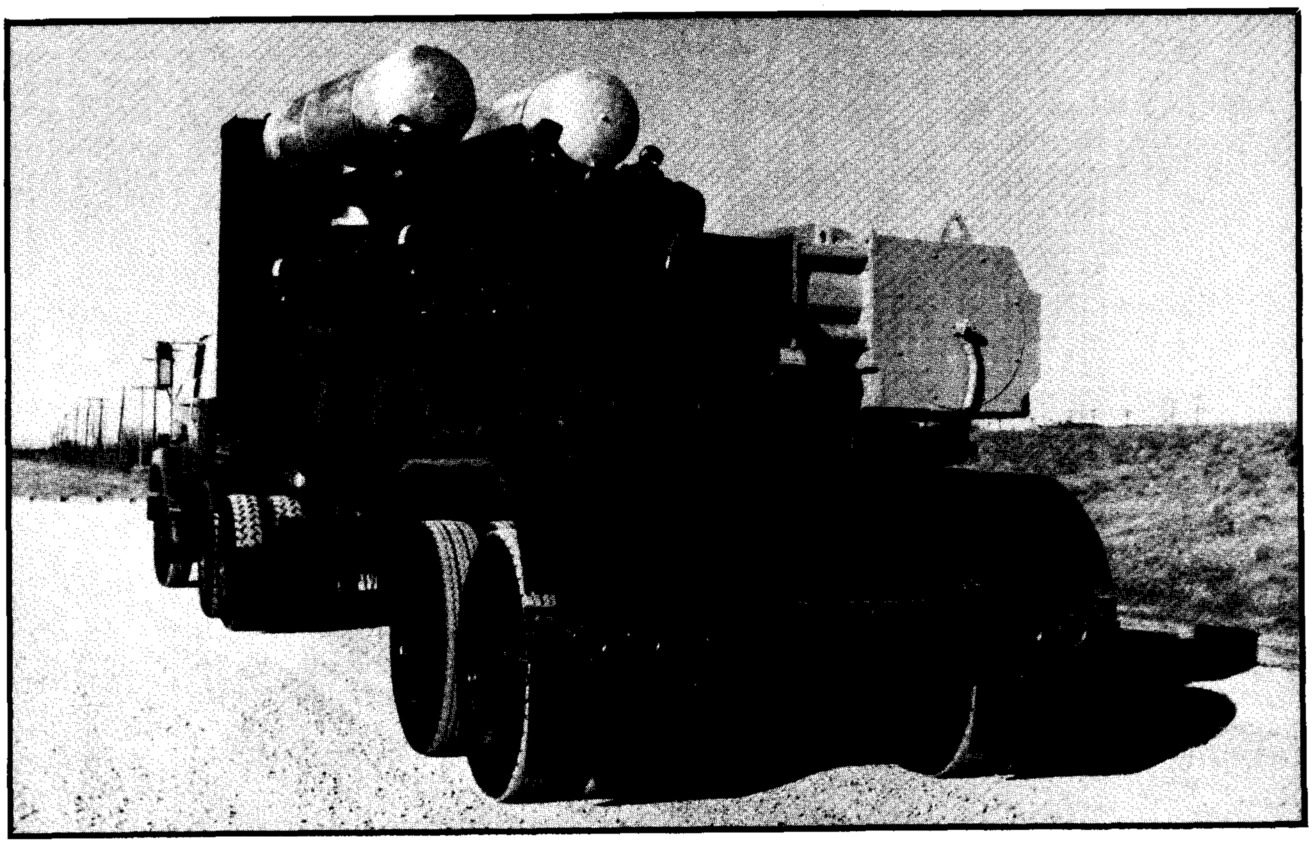

Figure 2 - Fracturing trailer - rear view

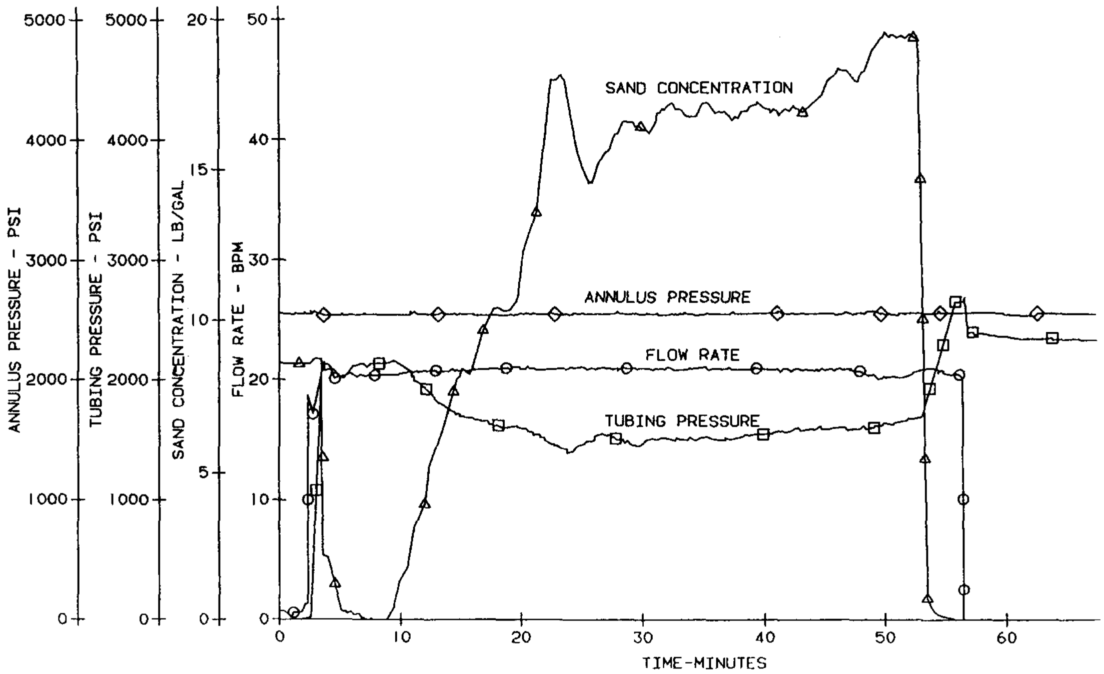

Figure 3 - Time trace of pressure and sand concentration

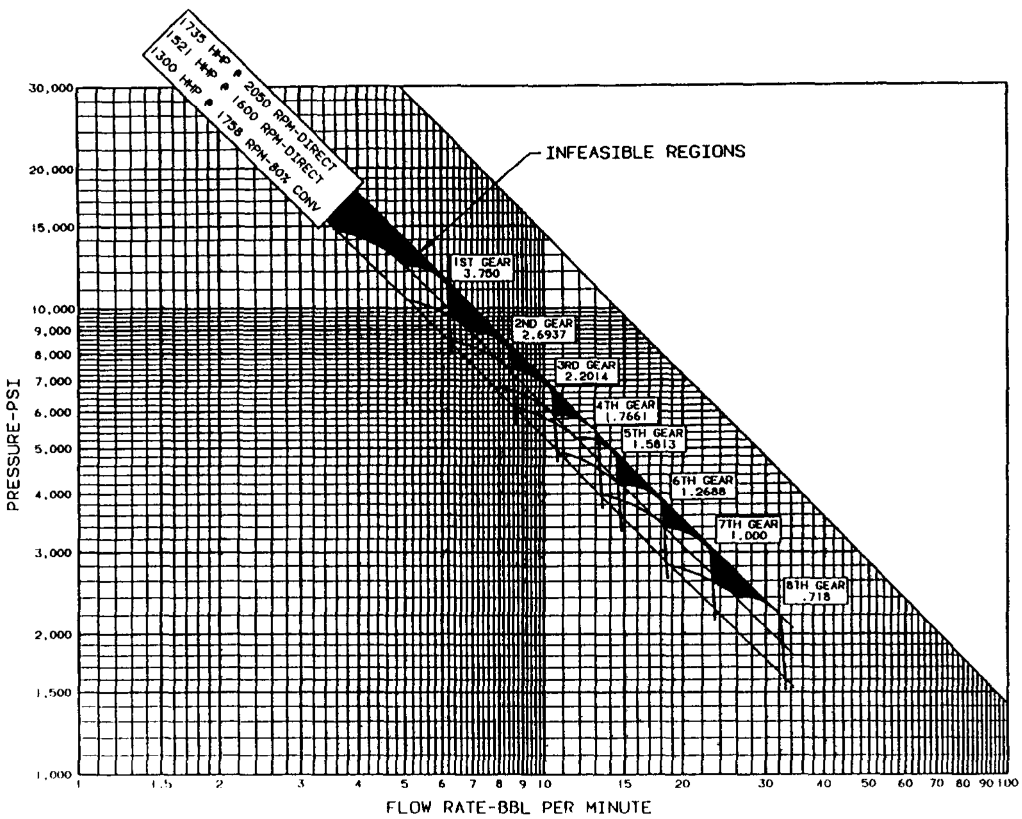

Figure 4 - Output characteristics of a pumping unit