Operating Efficiency of
Crankshaft Drive Pumps

By Dr. Scott Veenhuizen, OMAX Corporation

Nozzle output power, electrical input power, and pump crankshaft input power of a 380 MPa variable speed crankshaft drive 30 kW pump were measured at different flow rates and pressures to determine operating efficiency. The pump was independently mounted on flexible supports to allow limited rotation and equipped with a resisting load cell to measure input torque to the pump crankshaft. Overall efficiency was about 86 percent and efficiency of the crankshaft pump was about 93 percent. A crankshaft driven pump is shown to be significantly more efficient than hydraulically driven intensifier pumps.

1. Introduction

Experiments were conducted with a 30 kW (40 hp) crankshaft drive pump rated at 380 MPa (55,000 psi) to determine the power efficiency of the electric motor/drive, the crankshaft drive pumping component, and the overall pump unit.

2. Crankshaft Drive Pump

The pump was a triplex crankshaft drive pump designed to pump clean water. It is equipped with electronic variable frequency drive (VFD) that allows the speed of the electric motor, and therefore the crankshaft of the pump, to be continuously varied by varying the frequency of the electric power to the motor. The advantages of the variable speed drive are accommodation of slight changes in nozzle flow characteristics (discharge coefficient and nozzle wear), no current surge upon starting, and high power factors and VFD/motor efficiencies throughout the useful power range. A high efficiency gear belt drive with a ratio of 80/36 is used to transmit power from the faster turning electric motor to the slower turning crankshaft of the power end of the pump. The major drive components are depicted in the block diagram in figure 1.

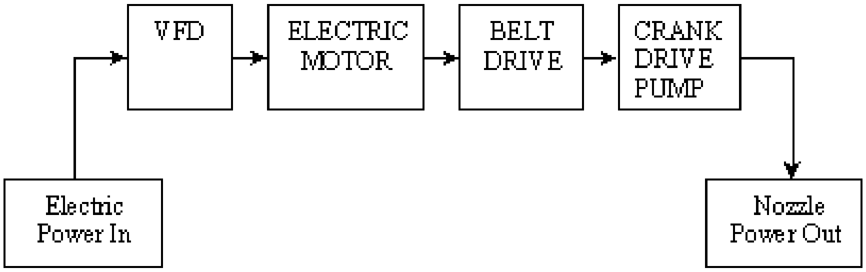

Figure 1. Major Drive Components of Crankshaft Drive Pump

3. Experimental Approach

Efficiency, h, was defined as the ratio of power out to power in: h = 100(kWout)/(kWin). Power was measured at three locations of Figure 1: at the electrical input to the VFD (Electric Power In), between the electric motor and belt drive, and at the nozzle (Nozzle Power Out). This allowed for calculation of the power efficiency of the VFD/motor, the belt drive/crankshaft drive pump, and the overall unit.

3.1. Output Power

Hydraulic output power of the nozzle was calculated from measured values of the pump output pressure and flow rate. Pressure was measured with an electronic pressure transducer calibrated against the laboratory pressure reference. The transducer was immediately at the outlet of the high-pressure manifold of the pump wet end. The laboratory pressure reference was a digital pressure gauge that had recently been calibrated by the manufacturer. Flow rate was determined by capturing flow from the nozzle in a 4,000 ml graduated cylinder for 60 seconds.

3.2 Electrical Input Power

Electrical power to the pump unit was delta configuration, balanced, 480 volt, 60 Hertz, three-phase. Electrical input power to the VFD/Motor was measured using a current shunt and a two-channel digital storage oscilloscope. In addition, voltage was measured with a voltage meter and current with a true rms current meter. Measurements were made on all three legs to confirm balanced three-phase power. The shunt was calibrated for a 50 mV drop at 30 amps. It was placed in series with an input lead to the VFD. The oscilloscope was specifically designed for high power systems with isolated inputs. It could store up to four signals that could then be downloaded into a PC. Signals recorded were voltage drop across the shunt and the corresponding phase voltage (277 V line to ground for 480 V line-to-line voltage). The power and corresponding power-factor was calculated from a numerical integration of the voltage and current signals.

3.3 Pump Crankshaft Torque

To obtain the output power at the motor shaft, or input to the belt drive and crankshaft, a load cell was used to measure shaft torque. The power end crankcase of the pump is normally foot mounted on the upper side of a tray-like base. On the under side of the base, the electric drive motor is mounted. The whole base, with the motor and crankshaft drive pump assembly is then mounted in the pump frame on vibration mounts. The crankcase was raised off the base and supported with brackets that were flexible in torsion around the longitudinal axis of the pump crankshaft. When the pump was run under load, the flexible brackets allowed the whole pump assembly to rotate slightly in the direction of crankshaft rotation. The slight rotation was resisted by the load cell at a fixed moment arm and measured the torque to resist the slight rotation. The arrangement is shown in Figure 2.

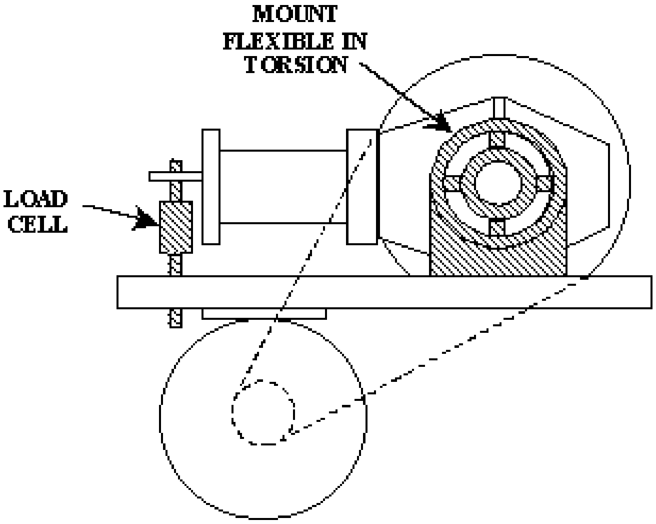

Figure 2. Arrangement for Torque Measurements

The load cell was calibrated directly in torque units up to 610 N-m (444 ft-lb) using a certified calibration tool that is specifically for calibrating torque wrenches. The torque calibration tool was used to apply a known input torque to the end of the motor shaft such that the drive belt was under proper tension during the torque calibration. The load cell was a strain gauge type force transducer rated at 890 N (200 lbs) in either tension or compression with a side load rejection of 500:1. It was excited with a 15 VDC signal and produced 30 mV signal at 890 N (200 lbs), which was amplified 100 times through a DC amplifier.

4. Experimental Results

Data was acquired for four different nozzle sizes: 0.25, 0.30, 0.36 and 0.38 mm (0.015, 0.014, 0.012, and 0.010 inch). Data was taken for each nozzle at nominal pressures of 138, 207, 276, 345, 380 MPa (20,000, 30,000, 40,000, 50,000, and 55,000 psi). No data was obtained, however, for the smallest nozzle at the lowest pressure because motor speed was less than the minimum allowed by the VFD. Motor and pump crankshaft speeds ranged from 560 and 260 rpm at the lowest pressure for the smallest nozzle to 1850 and 830 rpm at the highest pressure for the largest nozzle. Flow rates ranged from 1.3 to 3.95 liter per minute (0.34 to 1.04 gpm).

4.1. Crankshaft Pump Efficiency

Torque was measured with the load cell as the input torque to the belt drive-crankshaft. Although the belt and sprocket drive manufacturer stated its efficiency as 98 %, it was assumed to be 100 %. That is, the power measured at the motor belt sprocket was assumed to be the same as the input to the crankshaft of the pump. The crankshaft input torque values are plotted in Figure 3 with pump pressure. Input torque is the same at each pressure independent of nozzle size. The curve bends over slightly at higher pressures due to decreasing volumetric efficiency of by the compressibility of water at higher pressures. The input power to the belt drive/crankshaft pump combination was calculated from the data shown in Figure 3 by multiplying the torque value by the corresponding crankshaft rpm.

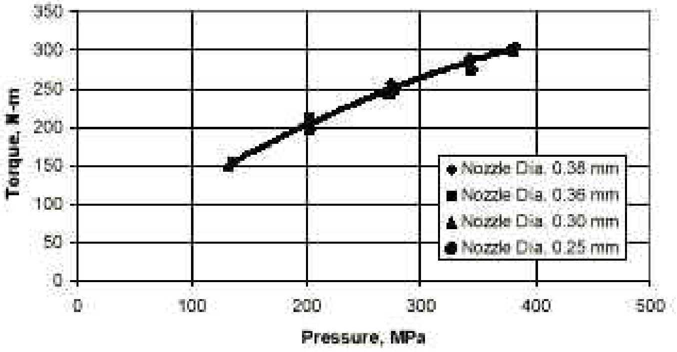

Figure 3. Pump Crankshaft Input Torque versus Pump Pressure

Nozzle power was calculated as the product of pressure times flow rate. Nozzle power is shown in Figure 4.

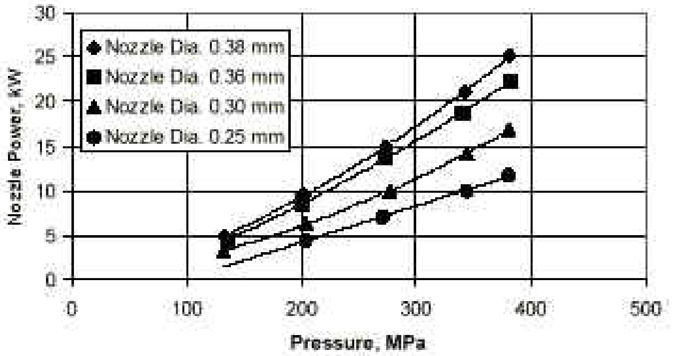

Figure 4. Output Hydraulic Power versus Pump Pressure

Shown in Figure 5 is the crankshaft pump efficiency as calculated from the nozzle output power values and the input torque values. It appears to vary linearly with pressure, and like the input torque, appears independent of nozzle size when plotted with nozzle pressure.

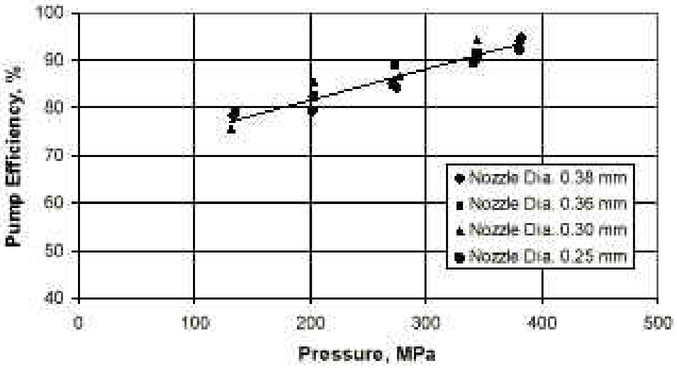

Figure 5. Crankshaft Drive Pump Efficiency

4.2. VFD/Motor Efficiency

Electrical input power is shown in figure 6. Using the measured torque values and pump crankshaft rpm, and assuming 100 % power transmission from the belt drive, the efficiency of the VFD/Motor could be determined. The efficiency of the VFD/Motor is shown in Figure 7.

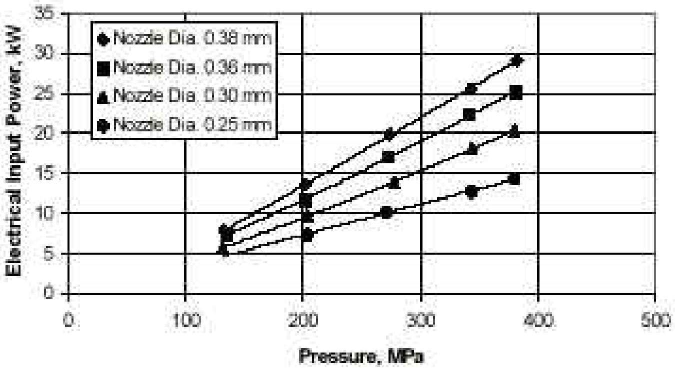

Figure 6. Input Power to the Variable Frequency Drive and Motor

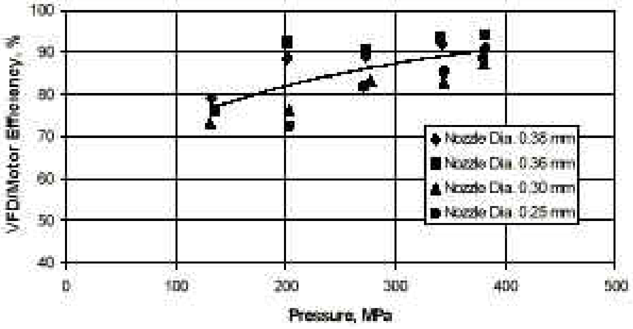

Figure 7. VFD/Motor Efficiency versus Nozzle Pressure

Although there is significant scatter in the data, particularly at 200 MPa, the overall trend indicates that the VFD/Motor combination was very efficient, even at low speed under high torque. This corresponds to lower voltage and higher current to the motor from the VFD. The scatter is due to the uncertainty in calculating real power from noisy current signals, particularly at lower current draw.

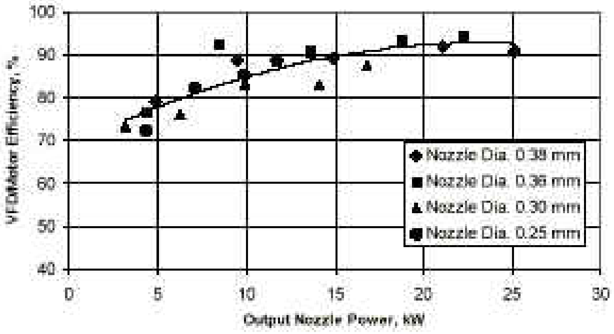

Figure 8. VFD/Motor Efficiency versus Output Nozzle Power

Shown in Figure 8 is the VFD/motor efficiency with power at the nozzle. Peak efficiencies calculated using the input electrical power values and the crankshaft input values were approximately 93 percent. The peak value published values by the motor at rated motor speed of 1760 rpm was 92 percent. The experimental values can be affected by power factor. Lower power factor values result in higher efficiency. Calculated power factors from the voltage and current data ranged from 0.86 to 0.93 at the highest input power. The VFD manufacturer indicated that the input power factor should be 0.95 for the VFD/motor.

4.3 Overall Efficiency

Overall efficiency is shown in Figure 9 with nozzle pressure. It range from a low of about 60% to almost 85%.

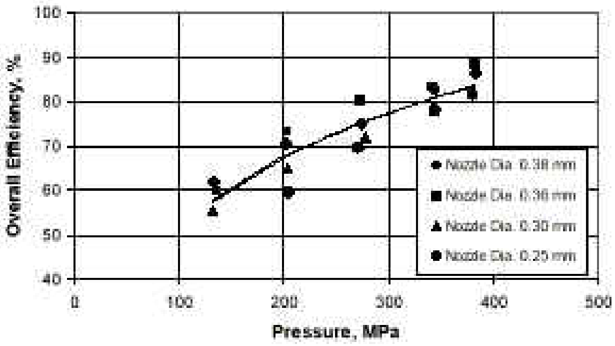

Figure 9. Overall Efficiency versus Nozzle Pressure

5. Comparison with Intensifier Type Pump Efficiencies

The efficiencies determined during these experiments can be compared with those obtained for intensifier style high-pressure pumps by Chalmers 1 and more recently by Herbig and Trieb 2 . Chalmers measured the operating efficiency of a 30 kW (40 hp) double acting intensifier pump that utilized a Pressure Compensated Variable Displacement (PCVD) hydraulic drive pump. Overall efficiency obtained by Chalmers was 60 percent between 10 kW nozzle power and the maximum nozzle power of 20 kW. Below 10 kW, the efficiency decreased rapidly with decreasing nozzle power. The efficiency of 60 percent is comparable to the value of 80 to 86 percent for overall efficiency shown in Figure 10 between 15 and 25 kW. Indirectly, assuming an average motor efficiency of 90 percent, Chalmers calculated a maximum operating efficiency for the hydraulic drive and double acting intensifier pump as 67 percent. This is comparable to the value of 93 percent for the crankshaft pump shown in Figure 10.

Herbig and Trieb also measured the overall efficiency of a double acting intensifier pump comparable in rated power to Chalmers' (probably 37 kW (50 hp) although it was not stated). Overall efficiency varied from 45 percent at about 12 kW to 70 percent at the maximum nozzle power of 23 kW. Corresponding efficiency values for the crankshaft pump shown as the lower curve in Figure 10 were 75 and 86 percent respectively. In comparison then, the belt driven crankshaft pump is significantly more efficient than the hydraulically driven intensifier pumps.

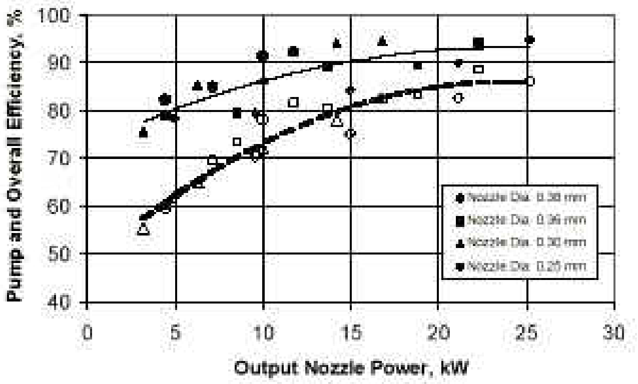

Figure 10. Crankshaft Driven Pump and Overall Unit Efficiencies. Filled Symbols = Crankshaft Pump Efficiency, Open Symbols = Overall Efficiency

6. Conclusions

Nozzle output power, electrical input power, and pump crankshaft input power of a 380 MPa variable speed crankshaft drive 30 kW pump were measured at different flow rates and pressures to determine operating efficiency. The pump was independently mounted on flexible supports to allow limited rotation and equipped with a resisting load cell to measure input torque to the pump crankshaft. The conclusions from this effort may be summarized as follows:

1. Efficiency of the Variable Frequency Drive/Motor varied from almost 80 percent to 93 percent in the range between 5 kW and 25 kW output nozzle power. It was above 90 percent in the range above 15 kW output nozzle power.
2. The crankshaft pump efficiency varied between 80 percent and 93 percent in the range between 5 kW and 25 kW output nozzle power and was also above 90 percent above 15 kW.
3. Overall operating efficiency of the entire unit varied from 60 percent to 86 percent in the same range. It was above 80 percent in the range above 15 kW.
4. In comparison to published overall efficiencies of 60 to 70 percent for hydraulically driven intensifier pumps, the crankshaft driven pump is significantly more efficient.

References

1. E. J. Chalmers, Pressure Fluctuation and Operating Efficiency of Intensifier Pumps, *Proc. of the 7 th American Water Jet Conference, vol. I, ed. Mohamed Hashish, 1993, 327-336.*
2. N. Herbig and F Trieb, Calculation of the Pumps*, Proc. of the 10 th American Water Jet Conference, vol. II, ed. Mohamed Hashish, 1999, 507-522.*
3. Proceedings from The 6th Pacific Rim International Conference on Waterjet Technology 9-11 October 2000. Sydney, Australia