Development of a Test Apparatus for Measurement of Hydraulic Fluid Efficiency

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ABSTRACT

With increasing demand for nonrenewable resources, energy conservation is critical. Efficiency gains allow more work to be performed while maintaining or even decreasing the energy expended in the process. Reducing the energy consumed by a system results in favorable economic and environmental impact.

An apparatus has been developed to measure hydraulic fluid efficiency in a stationary application. The system can be used to develop more efficient fluids, leading to increased work output or decreased energy consumption.

INTRODUCTION

Efficiency in hydraulic systems may be expressed in terms of overall efficiency by comparing the power output of the pump to the amount of power required to drive it. Overall efficiency can, in turn, be further analyzed in terms of both mechanical and volumetric efficiency.

High-accuracy measurements of input (driving) torque, pump shaft speed, output flow and pressure are all necessary for the calculation of overall hydraulic efficiency. An inline torque sensor of appropriate range, along with an accurate transducer at the pump outlet will provide the torque and pressure data necessary to calculate mechanical efficiency. Pump shaft speed can be measured via a magnetic pickup and a Coriolis mass flow meter provides precise measurement of pump output flow. Volumetric efficiency can be derived from the resulting speed and output flow data.

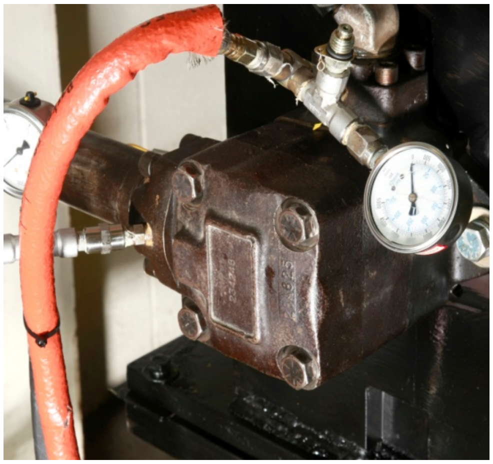
Pump speed was controlled using an electric motor and variable frequency drive. Output pressure was set through the use of a proportionally-controlled relief valve downstream of the pump. Fluid temperature was controlled with a shell-and-tube heat exchanger.

The focus of this study was the development of a stationary hydraulic test apparatus that, incorporating precise measurements of speed, torque, pressure, flow, and temperature, can be used to determine efficiencies of hydraulic fluid. Application of this apparatus will allow comparative fluid studies for the optimization of fluid formulations and the possible prediction of fluid behavior in both stationary and mobile equipment, without the need for costly field trials.

APPARATUS

The pump selected for development of the hydraulic efficiency apparatus was the Eaton 35VQ25A vane pump. This pump, shown in Figure 1, was selected due to its wide use as a fixed displacement test pump for fluid performance testing. To reduce wear-in effects, a unit with over 100 hours of operation at a broad range of conditions was selected. The pump was inspected before and after testing to ensure proper function and to determine that no mechanical degradation had occurred.

Figure 1: Eaton 35VQ25A Test Pump



A modified ASTM D6973-08 apparatus was used as the test bench. Input power was provided by an electric motor and shaft speed was controlled using a variable frequency drive. Closed loop output pressure control was accomplished using an electronically operated proportional valve. Temperature was controlled by routing the fluid through a shell-and-tube heat exchanger in a low pressure portion of the circuit. A full-flow filter was used to minimize particulate contamination.

Torque was measured at the pump input shaft using a 0.05% accuracy class digital flange-mounted, non-contact torque meter.[1] A magnetic pickup on a 60-tooth gear measured shaft speed with a total system accuracy of 0.1%.

A Coriolis mass flow meter was used to provide direct measurement of mass flow and density. The Coriolis meter is immune to flow profile effects[2] and has a volume flow accuracy of ± 0.10% of rate and repeatability of ± 0.05% of rate.[3] Electronic transducers were used to measure pressure at the pump inlet and outlet. The outlet transducer has a range of 0 to 690 bar and an accuracy of 0.1%. The inlet transducer has a range of 0 to 1.7 bar and an accuracy of 0.1%.[4]

A stainless steel K-type thermocouple was mounted 15 mm from the pump inlet to measure inlet temperature. In order to eliminate inlet system and reservoir design variables, the pump inlet pressure was controlled. The inlet pressure was set at 0.21 bar to simulate elevation of the reservoir above the pump in mobile or stationary applications. The system is shown in Figure 2.

Figure 2: SwRI Hydraulic Efficiency Test Rig

TEST FLUIDS

Hydraulic efficiency depends largely on fluid viscosity. Higher viscosity fluids tend to produce higher volumetric efficiencies due to reduced internal pump leakage and lower mechanical efficiencies due to the increased power required to pump them. Conversely, lower viscosity fluids give higher mechanical efficiencies because they are easier to pump, but also produce lower volumetric efficiencies due to increased leakage within the pump.

Three fluids were evaluated in developing the hydraulic efficiency test apparatus. Fluids A and B were ISO-32 grade fluids that were tested to determine the ability of the apparatus to measure efficiency differences between fluids of the same nominal viscosity. Fluid C was an ISO-46 grade fluid tested to compare efficiency across viscosity grades.

Table 1 provides a summary of the test fluid viscometric characteristics.

Table 1: Test Fluid Viscometrics

	Fluid A	Fluid B	Fluid C
ASTM D 445 (40°C) [cSt]	32.17	31.88	45.69
ASTM D 445 (100°C) [cSt]	6.32	6.25	6.96
ISO Grade	32	32	46

TEST PROCEDURE

The test fluids were evaluated at conditions representative of field operation. Shaft speeds (1200, 1800, and 2400 rpm) and output pressures (103, 155, and 207 bar) were selected to reflect field conditions near the upper end of equipment ratings. Temperature conditions ranging from 50 °C to 90 °C represent the widest practical range for the apparatus.

A summary of the test conditions is presented in Table 2.

Table 2: Test Conditions

Temperature [°C]	Shaft Speed [rpm]	Output Pressure [bar]
50		
55	1200	103
60		
65		
70	1800	155
75		
80		
85	2400	207
90		

Prior to each test sequence, the test conditions were stabilized for one hour at the maximum speed, pressure, and temperature combination. All shaft speeds were evaluated at a given temperature and pressure before continuing to the next pressure, and all pressures were tested before proceeding to the next temperature.

Test conditions were stabilized before and after the data acquisition period. Data was recorded once per minute following stabilization. The data used for efficiency calculation was an average of the first five minutes of stable operation at each condition. Test conditions were maintained for a minimum of one minute after data acquisition was complete before proceeding to the next combination in order to eliminate the possibility of transients in the data set.

DATA ANALYSIS

Overall, mechanical, and volumetric efficiencies were calculated for each operating condition according to the following equations:

Overall Efficiency[5]:

$$\eta_O = \frac{P \times Q}{T \times N} \times 15.92 \times 100\% \qquad (1)$$

Mechanical Efficiency[5]:

$$\eta_m = \frac{P \times V_p}{T \times 2\pi \times 10} \times 100\% \qquad (2)$$

Volumetric Efficiency[5]:

$$\eta_v = \frac{Q \times 1000}{V_p \times N} \times 100\% \qquad (3)$$

where

P = Pump outlet pressure [bar]

Q = Pump output flow [liters/min]

T = Shaft torque [Nm]

N = Shaft speed [rev/min]

V_p = Pump displacement [81.6 ml/rev][6]

Based upon the calculated efficiencies, some general trends were observed. Fluids were more volumetrically efficient at higher speeds (Figures 3 through 5, Appendix) and more mechanically efficient at higher pressures (Figure 6 through 8, Appendix).

Increases in temperature had mixed results for mechanical efficiency. At the lower end of the temperature range, mechanical efficiency increased with temperature. At higher temperatures, however, the rate of increase became more gradual and seemed to indicate that mechanical efficiency might stabilize or even decrease at higher temperatures. This phenomenon was most pronounced at lower speeds and higher pressures (Figure 8, Appendix).

Lower temperatures produced higher volumetric and overall efficiency (Figures 3 through 5 and Figures 9 through 11, Appendix).

Overall efficiency trends followed those exhibited by the volumetric efficiency data, suggesting that volumetric effects have a larger impact on overall efficiency.

Table 3 summarizes the average efficiency differences among the three fluids across all conditions.

Table 3: Average Efficiency Differences

	Volumetric	Mechanical	Overall
Oil B vs. Oil A	-0.49%	0.10%	-0.39%
Oil C vs. Oil A	4.04%	-0.11%	3.93%
Oil C vs. Oil B	4.56%	-0.21%	4.35%
Uncertainty	± 0.14%	± 0.11%	± 0.18%

Fluids A and B did not exhibit significant differences in efficiency. Fluid C produced the highest volumetric and overall efficiencies of the three fluids (Figures 3 through 5 and Figures 9 through 11, Appendix). Fluid C exhibited notable separation from Fluids A and B, likely due to the differences in their viscosities (Figures 3 through 5 and Figure 11, Appendix).

UNCERTAINTY AND REPEATABILITY

Uncertainties for the results of the efficiency investigations were quantified using the root sum of squares rule[7] The published torque meter accuracy was 0.05%. The accuracy of the flow meter, pressure transducer, and speed sensor was 0.1% for each device. The square root of the sum of the squares of the accuracy values yields uncertainty values of ±0.18% for overall efficiency, ±0.11% for mechanical efficiency, and ±0.14% for volumetric efficiency.

An example calculation for uncertainty in the overall efficiency values is shown in Equation 4.

$$RSS = \sqrt{(0.1\%)^2 + (0.1\%)^2 + (0.1\%)^2 + (0.05\%)^2} \quad (4)$$

Repeatability was determined by recording data multiple times on a single fluid at one set of conditions (1800 rpm, 155 bar output pressure, 85 °C inlet temperature). Four data sets were recorded on two different days with a minimum of three hours of cold soak time between each set. Data was recorded and averaged according to the same procedure used for the full investigation. These repeatability tests resulted in a range of 0.15% for overall efficiency, with the ranges for mechanical and volumetric efficiency at 0.09% and 0.14%, respectively.

It should be noted that overall accuracy will also be influenced by the accuracies of all devices in each signal path. Further quantification of these influences is warranted.

CONCLUSIONS

From the data gathered during the development of the hydraulic fluid efficiency apparatus, it can be concluded that changes in volumetric efficiency have a much larger effect on overall efficiency than changes in mechanical efficiency.

Efficiency results for fluids in the same viscosity grade are similar, while differences in efficiency are much more pronounced for fluids of differing viscosity grades. There were no appreciable differences measured in terms of mechanical efficiency.

The hydraulic fluid efficiency apparatus can be a useful tool for comparing efficiency among fluids. As with any new test, additional follow up work is warranted to enhance the method and more fully develop the data set.

Additional fluids of differing characteristics should be evaluated. Data to date was recorded over multiple combinations of temperature, pressure, and speed conditions in order to provide a general overview of the performance of the apparatus. Repeated tests of the same fluids at each condition should be performed in order to establish a confidence interval for the efficiency measurements.

Further investigation may involve variable displacement pumps or pumps of other design or manufacture. Testing across a wider range of fluid temperatures in order to assess efficiency at extreme operating conditions is another avenue worthy of exploration. Wear measurements of pump components before and after testing will better quantify pump degradation effects.

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REFERENCES

1. HBM Inc., Data sheet B2484-4.0 en for T40 torque flange, 1 kNm size.

2. T. Patten “The Micro Motion® ELITE® Promise”, Micro Motion White Paper WP-00842, 2005.

3. “Micro Motion® ELITE® Coriolis Flow and Density Meters”, Emerson Process Management Product Data Sheet PS-00374, Rev. L (2009).

4. Honeywell, Model TJE, 008606-1-EN IL50 GLO data sheet, May 2008

5. James A. Sullivan, Fluid Power theory and Applications, 3rd edition, Prentice Hall, Englewood Cliffs, New Jersey, 1989, pages 123 through 128. (equations adjusted for units)

6. EATON Vane Pump and Motor Design Guide for Mobile Equipment, Revised 08/98, page 47.

7. H. Coleman, W. Steele, Experimentation and Uncertainty Analysis for Engineers, John Wiley & Sons 1989.

APPENDIX

Figure 3: Volumetric Efficiency vs. Temperature, 2400 rpm

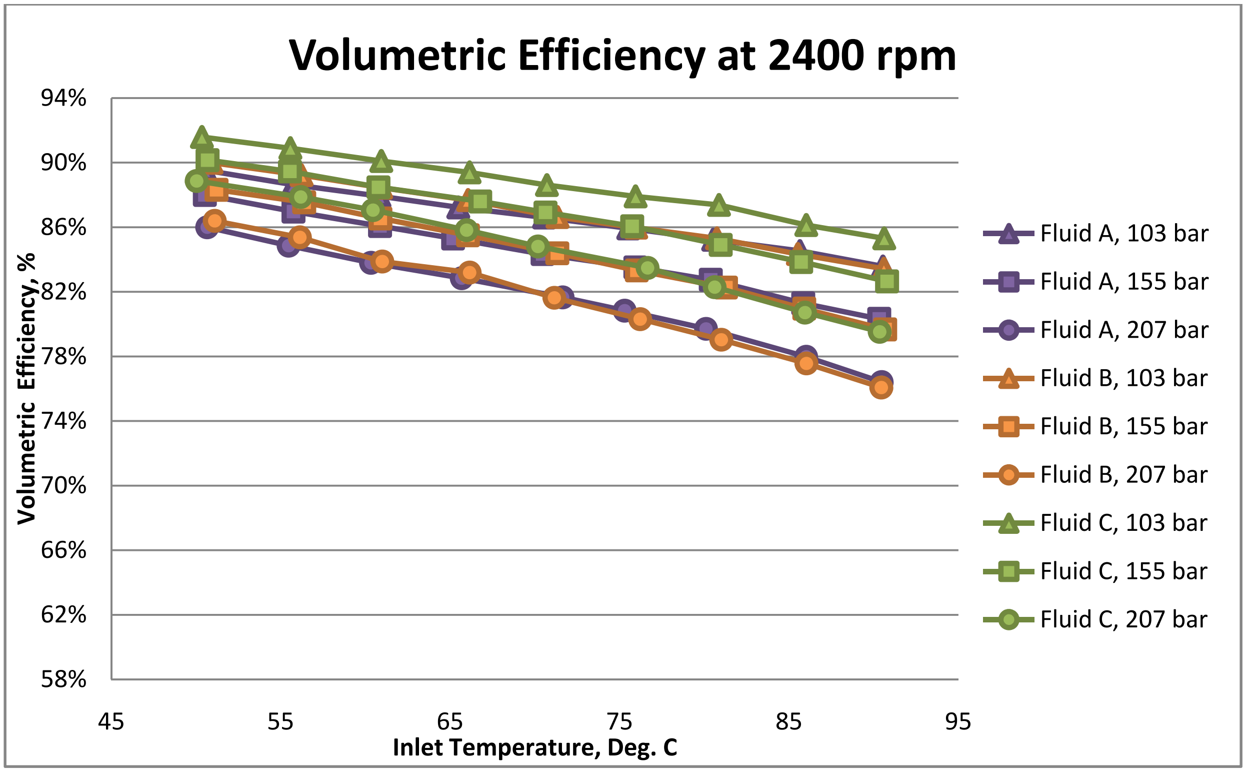

Figure 4: Volumetric Efficiency vs. Temperature, 1800 rpm

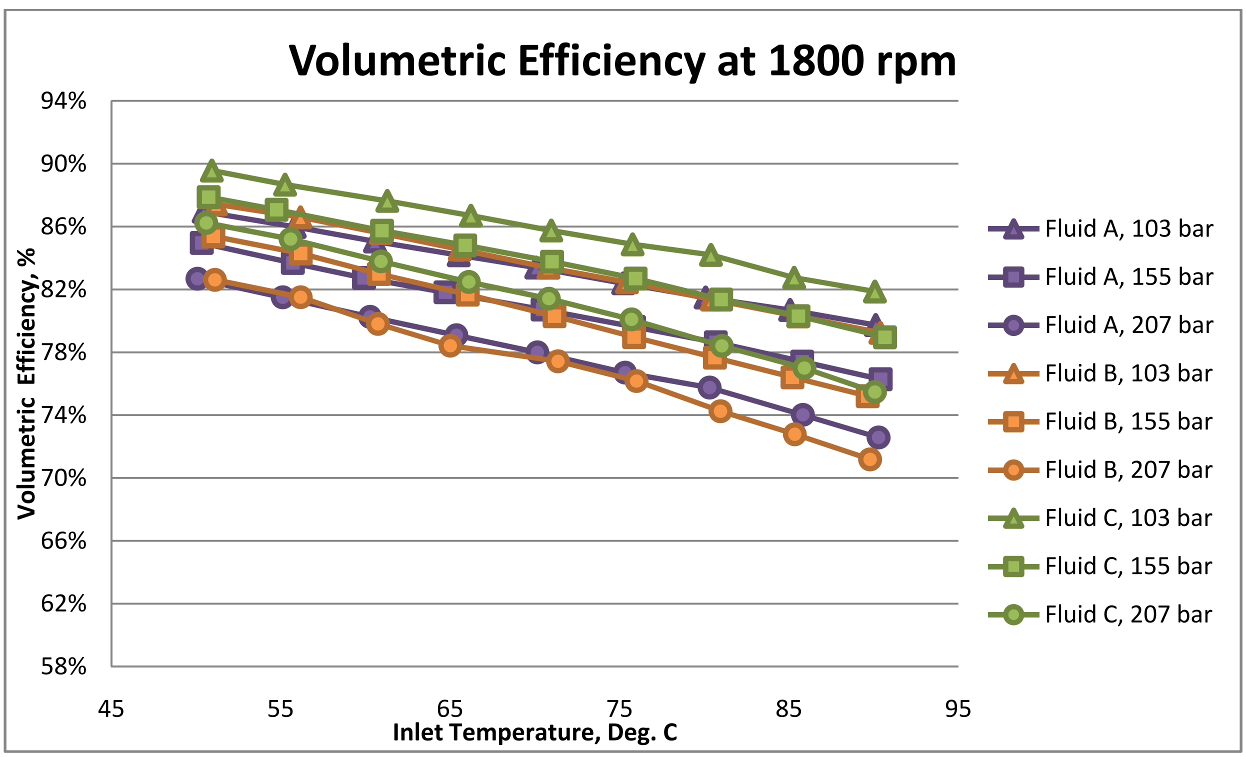

Figure 5: Volumetric Efficiency vs. Temperature, 1200 rpm

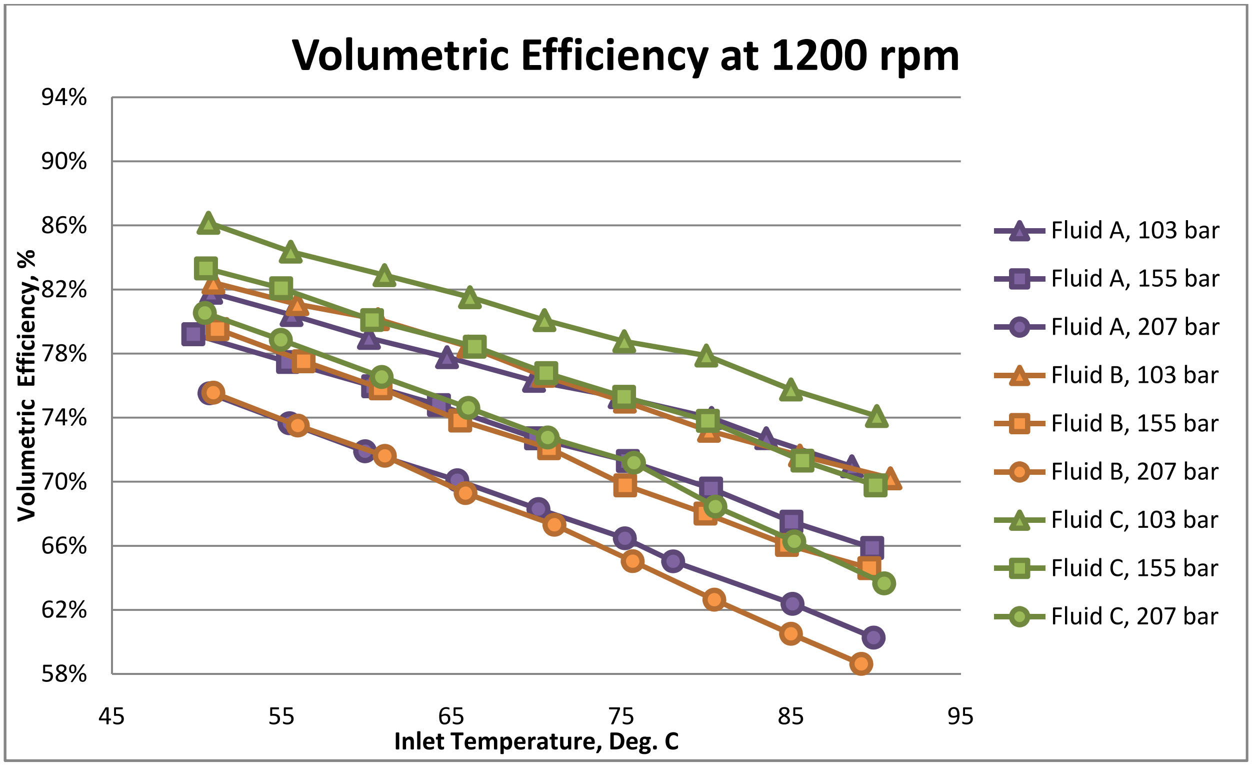

Figure 6: Mechanical Efficiency vs. Temperature, 2400 rpm

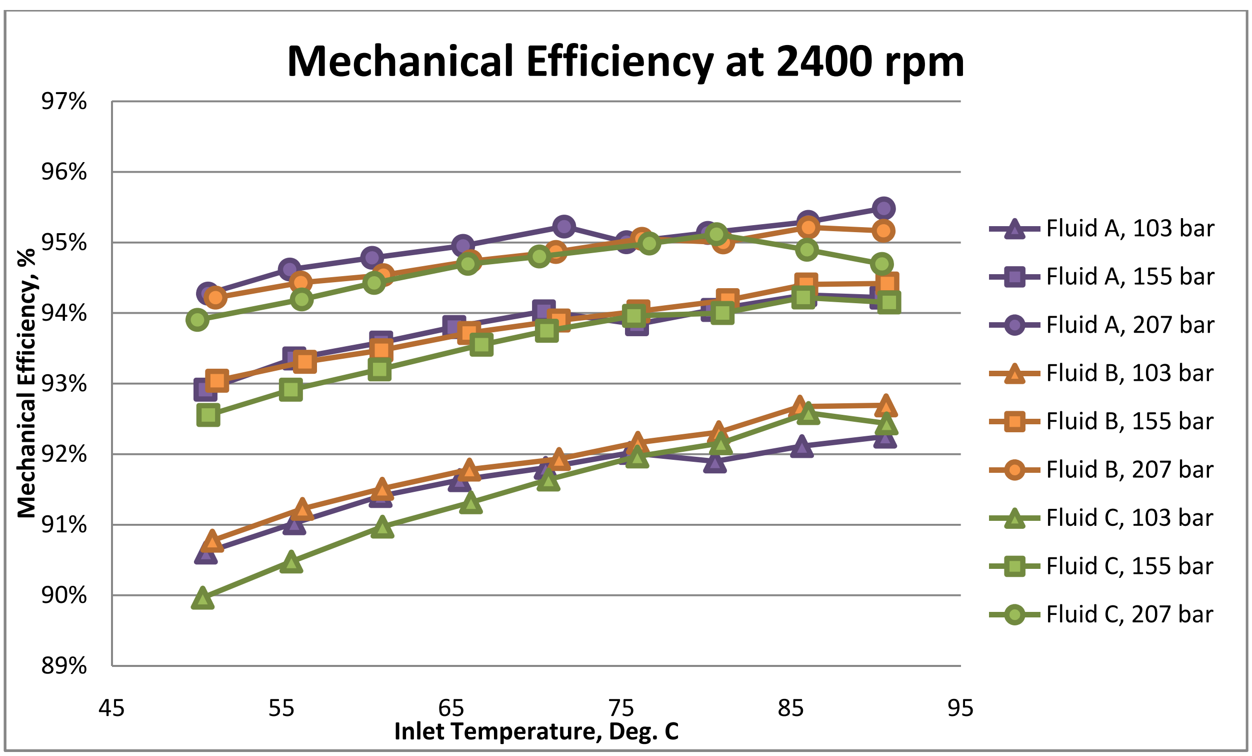

Figure 7: Mechanical Efficiency vs. Temperature, 1800 rpm

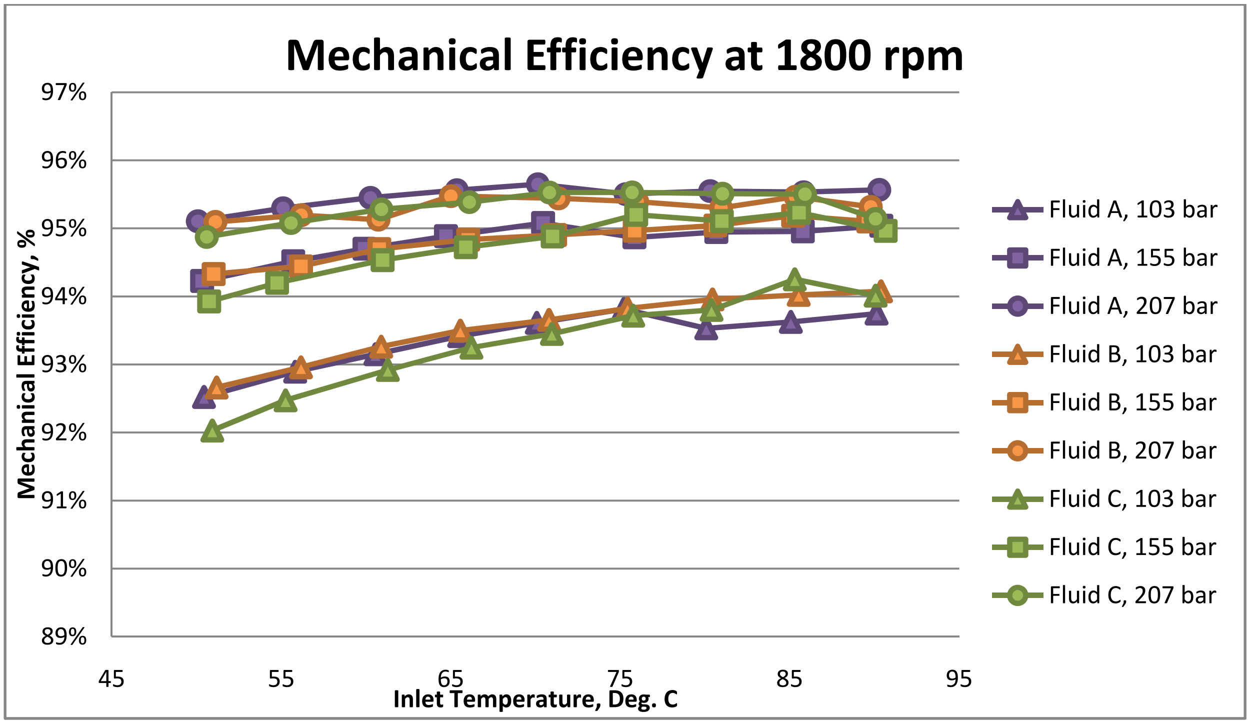

Figure 8: Mechanical Efficiency vs. Temperature, 1200 rpm

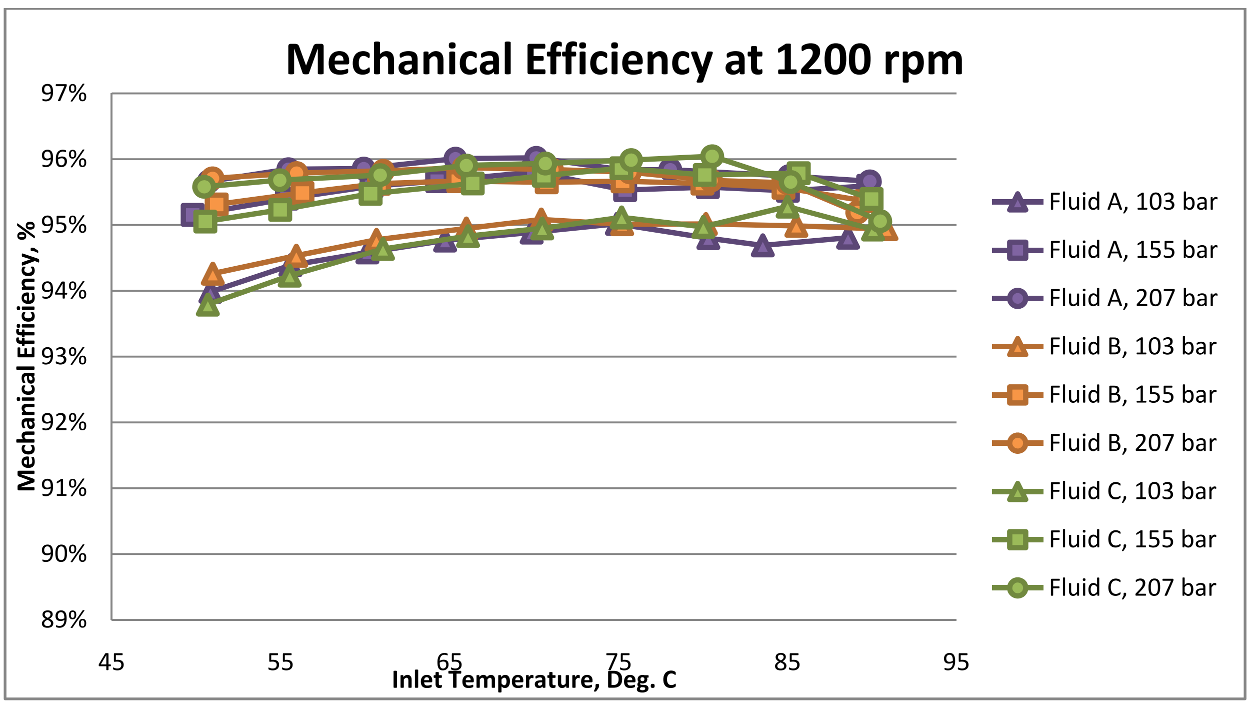

Figure 9: Overall Efficiency vs. Temperature, 2400 rpm

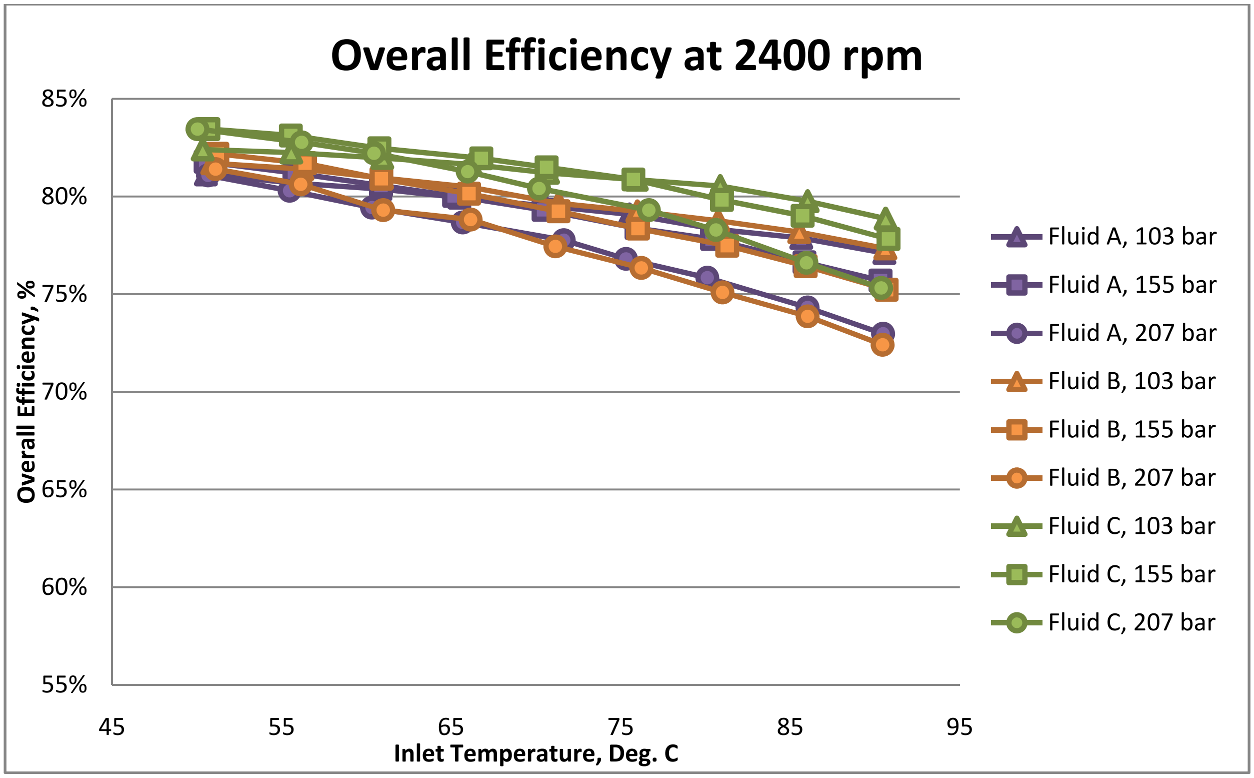

Figure 10: Overall Efficiency vs. Temperature, 1800 rpm

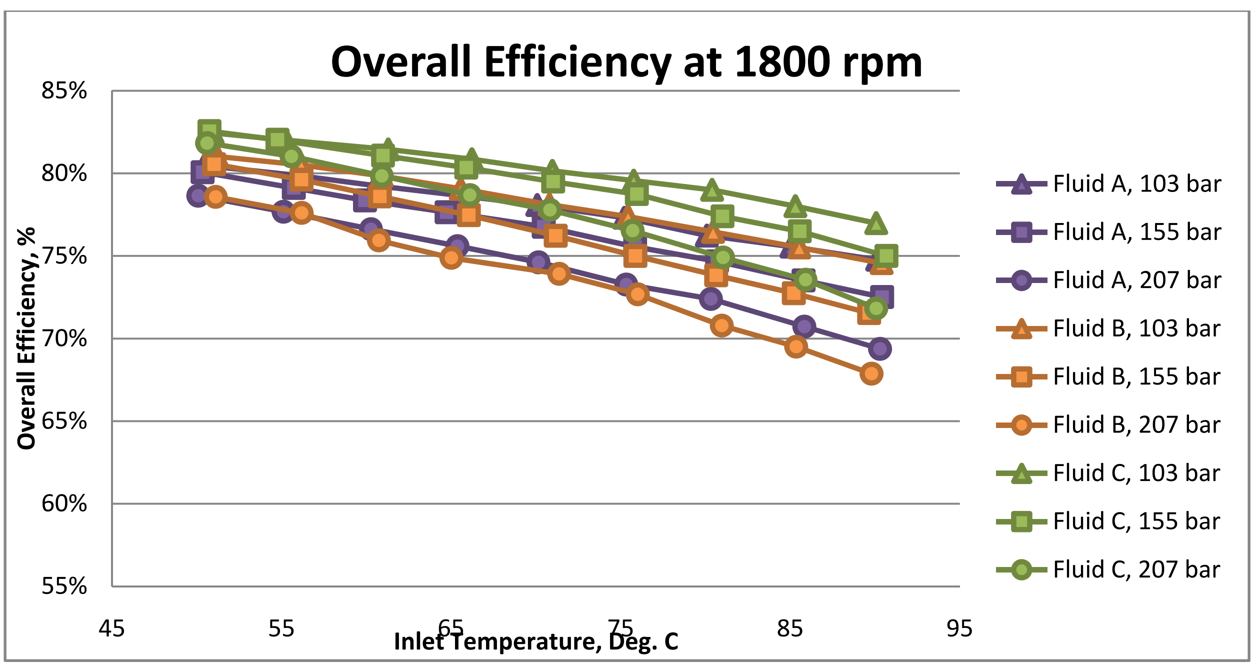

Figure 11: Overall Efficiency vs. Temperature, 1200 rpm