

CENTRIFUGAL PUMP PERFORMANCE UNDER STABLE AND UNSTABLE OIL-WATER EMULSIONS FLOW

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ABSTRACT

The performance change in centrifugal pumps operating under flow of stable and unstable oil-in-water emulsions is determined experimentally. The head-flow rate is measured at different temperatures and holdup values and represented by empirical relations. A comparison between head-flow curves for centrifugal pumps under water and emulsions flow results in coefficients relating the flow rate and head for water and emulsions. The hydraulic efficiency is calculated for stable and unstable emulsions and compared with water flow.

The oil-in-water emulsion flow reduces the head and flow rate of the centrifugal pumps. As the holdup increase and temperature decrease, the reduction in head and flow rate increase. The unstable oil-in-water emulsions show less decrease in head-flow rate, while stable emulsions show bigger decrease in head-flow rate compared with water. The surfactant type affects the head-flow rate, where the stable emulsions with Sodium Dodecyl Sulfate (SDS) show less reduction in head-flow rate than stable emulsions with Fatty Acid and Amine (FAA). The change in the rheology of the emulsions with holdup, temperatures, and addition of surfactant is responsible for increasing the losses inside the pump and change in performance. Thus, the hydraulic efficiency decreases as the holdup increase and temperature decrease.

Keywords: Centrifugal pumps, oil-in-water emulsions, energy losses, Hydraulic machines, Two-Phase flow.

INTRODUCTION

A centrifugal pump is one of the simplest pieces of equipment in any process plant. Its purpose is to convert energy of a prime mover (electric motor or turbine) first into velocity or kinetic energy and then into pressure energy of a fluid that is being pumped. The energy changes occur by virtue of two main parts of the pump, the impeller and the volute or diffuser, Fig. 1. The impeller is the rotating part that

converts driver energy into the kinetic energy. The volute or diffuser is the stationary part that converts the kinetic energy into pressure energy.

The hydraulic performance of centrifugal pumps can be illustrated by using three curves: head-flow rate, power-flow rate, and efficiency-flow rate. The performance can be given exactly through experiment only. Li (2004) calculated slip factor, hydraulic, mechanical and volumetric efficiencies based on the experimental performance of centrifugal pump. The slip factor grows and then drops while the viscosity increases. The hydraulic efficiency decreases while the viscosity gets higher due to larger friction loss over flow passage surfaces of impeller, inlet casing and volute. The volumetric efficiency keeps constant or increases while the viscosity grows because of less leakage through the wearing-ring. The mechanical efficiency reduces quickly as the viscosity grows. The reduction of the disc friction loss of impeller is a key issue to improve the performance of centrifugal pumps.

Ogata et al. (2006) measured the performance of a centrifugal pump when handling surfactant solutions. It was clarified that the pump efficiency with surfactant solutions was higher than that with tap water and increased with an increase in surfactant concentration. The value of maximum flow rate also increases. The total pump head increased with an increase in concentration, and shaft power decreased with a decrease in the impeller rotating speed.

Performance prediction of centrifugal pumps handling two-phase mixtures has been an active area of research in both nuclear and petroleum industries. The nuclear industry is concerned with the loss of coolant accidents in nuclear reactors due to safety purposes. In such accident, the reactor centrifugal pumps need to handle two-phase mixtures because of a rapid depressurisation of the coolant. As for the petroleum industry, fair amounts of gas are entrained in the oil handled by electric submersible pump applications. In both cases, the pump undergoes a decrease in the head delivered, which must be estimated. Thus, several investigators [Cardidad et al. 2005, Cao et al. 2005, Noghrehkar et al. 1995, Minemura et al. 1993, Pak et al. 1998, Minemura et al. 1983] have undertaken the performance of pumps operating under two-phase (gas-liquid) flow conditions.

It is well established that the performance of centrifugal pumps with slurries gets reduced in the presence of solids in the carrier liquid. The magnitude of the reduction is a function of concentration of solids in the mixture, physical properties of solids like their specific gravity, size and size distribution of particles, and pump size [Engin et al. 2003, Bross et al. 2002].

Knowledge of centrifugal pump performance handling emulsions is required for the design, selection, and operation of the centrifugal pump used in the petroleum industry such as oil well drilling, and improving oil recovery. Despite the importance of emulsion flows, such flows have not been explored to the same extent as gas-liquid and solid-liquid flows. Thus, the purpose of the present study is to investigate the

performance characteristics of the centrifugal pump with stable and unstable oil-in-water emulsions at different holdup and temperature values.

EMULSION PREPARATION

Three different sets of emulsions were prepared using tap water and a refined white mineral oil. The white mineral oil is low viscosity colorless, tasteless and odorless highly refined paraffinic oil, supplied by CO-OP Company, Alexandria, Egypt. Its density is 0.9 gm/cm^3 at $25 \text{ }^\circ\text{C}$ and viscosity shown in Fig. 2. In one set of emulsions, no chemical-emulsifier (surfactant) added so that unstable emulsion is produced. The unstable emulsion separated into oil and water if left without agitation for sometime. The experiment in this set began with tap water into which a required amount of oil varies from 0 to 65% by volume, was added to prepare emulsion.

In the second set of emulsion a surfactant namely Fatty Acid and Amine which prepared as follows, 3% Oleic Acid [$\text{C}_{18}\text{H}_{34}\text{O}_2$] from the total volume of mineral oil, and 1% Trimethylamine [$\text{C}_3\text{H}_9\text{N}$] from the total volume of water. In the third set of emulsion an ionic surfactant namely Sodium Dodecyl Sulfate [$\text{CH}_3(\text{CH}_2)_{11}\text{OSO}_3\text{Na}$] is added to the oil with 1.5% by weight based on the water.

TEST PUMP

The test pump is a centrifugal pump of type Calpeda (NMM, 1/A), with impeller (NM20/110 MOD0120) (Fig. 2), of 1”/1” suction/delivery diameters. The parameters of the pump are: flow rate $Q_{\min}=1 \text{ m}^3/\text{h}$, $Q_{\max}=4.2 \text{ m}^3/\text{h}$, head $H_{\max}=22 \text{ m}$, $H_{\min}=15.5 \text{ m}$, rotating speed $N=2900 \text{ RPM}$, and power = 0.5 HP. The geometry of impeller are: impeller diameter $D_0=130 \text{ mm}$, eye diameter $D_1=32 \text{ mm}$, blade outlet width $b=2 \text{ mm}$, blade outlet angle $\beta_2=45^\circ$, and number of blades $z=6$.

ENERGY CONSERVATION EQUATION

The energy conservation equation in centrifugal pumps can be written as [Li, 2004]:

$$H = H_{th} - K Q^2 \quad (1)$$

Where H stands for the known head of the pump, H_{th} presents the theoretical head, Q denotes the known flow rate and K is hydraulic loss coefficient, which can be considered as a constant. The experimental data of performance and the geometry of impeller can determine H_{th} and K . The hydraulic losses in centrifugal pumps include friction loss along flow channel wall, shock loss at leading edge of blade and local vortex loss. Friction losses due to high viscosity of liquid and shock loss have large percentage in total, so the local vortex can be neglected.

The theoretical or ideal head rise, H_{th} , for a centrifugal pump varies linearly with Q for a given blade geometry and angular velocity, and expressed as [Munson et al., 1998, Li, 2004]

$$H_{th} = \frac{U_o}{g} \left(\sigma U_o - \frac{Q}{2\pi r_o b \eta_v \tan \beta_o} \right) \quad (2)$$

Where b is the impeller blade height at the radius r_o , and U_o is impeller tip speed, $U_o = (r_o \omega)$. In Eq. 2, it is assumed that the fluid has no tangential component of velocity $V_{\theta i}$, or swirl, as it enters the impeller; i.e., the angle between the absolute velocity and the tangential direction is 90° ($\alpha_i = 90^\circ$ in Fig. 2). σ stands for the slip factor, where σU_o represents the impeller tip speed after considering the slip of flow at impeller discharge. η_v is volumetric efficiency of pump. For actual pumps, the blade angle β_o falls in the range of 15° - 35° , with a normal range of $20^\circ < \beta_o < 25^\circ$, and with $15^\circ < \beta_i < 50^\circ$. Blades with $\beta_o < 90^\circ$ are called backward curved, whereas blades with $\beta_o > 90^\circ$ are called forward curved. Pumps are not usually designed with forward curved vanes since such pumps tend to suffer unstable flow conditions.

The performance of centrifugal pumps is affected when handling viscous liquids. A marked increase in brake horsepower, a reduction in head, and some reduction in flow rate occur with moderate and high viscosity. The following equations are used for determining the viscous performance when the water performance of the pump is known:

$$Q = C_Q Q_w \quad (3)$$

$$H = C_H H_w \quad (4)$$

$$\eta = C_\eta \eta_w \quad (5)$$

Where Q , H , and η are the flow rate, head, and efficiency of emulsion. Q_w , H_w , and η_w are the flow rate, head, and efficiency of water. C_Q , C_H , and C_η are flow rate, head, and efficiency correction factors.

Another form for Eq. (1), such that in normal operation, a pump characteristic is given by Ntoko (1996):

$$H = a + bQ + cQ^2 \quad (6)$$

Empirical equation similar to Eq. (6) is obtained using Excel software to fit the experimental results of centrifugal pump, Head and flow rate, for stable and unstable o/w emulsion at different holdup and temperature values. The constants a , b , and c are given in table 1. R^2 is the square of the correlation between the experimental data values and the predicted fit values. R^2 can take any value between 0 and 1, with a

value closer to 1 indicating a better fit. Thus, R^2 is a measure of how successful the fit is in explaining the variation of the experimental data.

The hydraulic efficiency, η_h , is calculated as follows:

$$\eta_h = \frac{H}{H_{th}} \quad (7)$$

Where H stands for the measured head of the pump, and H_{th} presents the theoretical head in Eq. (2). Based on Li (2004), the slip factor is assumed 0.62 and volumetric efficiency 0.85.

The ratio of the oil volume to the total volume of emulsion is called holdup and represented as:

$$Holdup = \frac{\text{Oil Volume}}{\text{Oil Volume} + \text{Water Volume}} \quad (8)$$

RESULTS AND DISCUSSION

Head-flow rate and hydraulic efficiency-flow rate curves at different values of holdup and temperatures represent the performance of the centrifugal pump used in this study. Head correction factor, C_H , and flow rate correction factor, C_Q , represent a comparison of pump performance for emulsion and water. In Figs. (3-5) the performance under stable o/w emulsion (SDS), Stable o/w emulsion (FAA), and unstable o/w emulsion at 25 °C, show that the emulsion gives lower head and flow rate than water. As the holdup increase, the difference between emulsion and water increase. Thus, the hydraulic efficiency for emulsion is lower than water, and decreases as the holdup increase. This behaviour can be explained in terms of rheology, where the emulsion rheology studied in literature [Khalil et al., 2006]. This study showed that the emulsion viscosity increases as holdup increase and temperature decrease. Furthermore, the unstable emulsion viscosity was found less than stable one, and type of surfactant affects the emulsion viscosity. The hydraulic losses inside the pump increases as the viscosity increase, this explains the drop in head for emulsion compared with water.

Figures (6-8) show that the centrifugal pump performance at 50 °C is enhanced, where the expected decrease in emulsion viscosity with temperature rise leads to decrease the hydraulic losses inside the pump.

CONCLUSIONS

- 1- The oil-in-water emulsion flow reduces the head, flow rate and hydraulic efficiency of the centrifugal pumps. As the holdup increase and temperature decrease, the reduction in head and flow rate increase.
- 2- The unstable oil-in-water emulsions show less decrease in head-flow rate, while stable emulsions show bigger decrease in head-flow rate compared with water.
- 3- The surfactant type affects the head-flow rate, where the stable emulsions with Sodium Dodecyl Sulfate (SDS) show less reduction in head-flow rate than stable emulsions with Fatty Acid and Amine (FAA).
- 4- The change in the rheology of the emulsions with holdup, temperatures, and addition of surfactant is responsible for increasing the losses inside the pump and change in performance. Thus, the hydraulic efficiency decreases as the holdup increase and temperature decrease.

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NOMENCLATURE

D_i	impeller inner diameter, mm
D_o	impeller outer diameter, mm
H	head, m
Q	flow rate, m ³ /s
T	temperature, °C
V	mixture absolute velocity, m/s
U	impeller speed, m/s
W	mixture relative velocity, m/s
H_{th}	theoretical head, m
Φ	holdup
μ	mixture viscosity, cp
ρ	mixture density, kg/m ³
α	angle between the absolute velocity and the tangential direction
β	blade angle
η	efficiency
σ	slip factor

List of Abbreviations

o/w	oil-in-water emulsions
SDS	sodium dodecyl sulfate
FAA	fatty acid and amine

List of Chemical Symbols

$C_{18}H_{34}O_2$	oleic acid
C_3H_9N	trimethylamine
$CH_3(CH_2)_{11}OSO_3Na$	sodium dodecyl sulfate

Table 1. $H = cQ^2 + bQ + a$

Holdup	Temp. (°C)	c			b			a			R ²	R ²	R ²
		Stable (SDS)	Stable (FAA)	Unstable	Stable (SDS)	Stable (FAA)	Unstable	Stable (SDS)	Stable (FAA)	Unstable	Stable (SDS)	Stable (FAA)	Unstable
15%	25	3.66E5	-5E6	3.26E5	-16746	-9822.4	-15795	18.1	15.67	18.42	0.96	0.97	0.96
30%	25	-7E6	-6E6	-6E6	-8742.1	-10444	-8993.1	15.7	15.74	16.13	0.96	0.97	0.97
45%	25	-1E7	-1E7	-1E7	-6154.2	-7604.4	-6295.6	14.65	14.8	15.48	0.98	0.98	0.98
50%	25	-2E7	-2E7	-2E7	-2951.5	-4237.4	-2997.2	13.84	13.99	14.73	0.99	0.98	0.99
55%	25	-3E7	-3E7	-3E7	8686.5	8709.5	8997.7	10.64	10.79	11.46	1	1	1
60%	25	-3E7	-4E7	-3E7	12948	13157	13175	9.777	9.81	10.4	1	1	1
65%	25	-3E7	-6E7	-5E7	10991	24190	21432	8.84	8.86	9.2	1	1	1
Water	25	-1E7			-3841			19.75			0.99		
15%	50	-4E6	-4E6	3.11E5	-10746	-1.25E4	-13974	17.78	17	18.18	0.97	0.95	0.96
30%	50	-5E6	-6E6	-2E6	-7844.8	-8818	-10830	15.5	15.52	17.15	0.97	0.97	0.96
45%	50	-7E6	-9E6	-5E6	-4951.1	-5.62E3	-8029.4	14.47	14.59	16.5	0.98	0.98	0.97
50%	50	-9E6	-1E7	-6E6	-5420.6	-2.68E3	-8935.6	14.8	13.78	17	0.98	0.99	0.97
55%	50	-1E7	-1E7	-1E7	-2577.5	-2777.7	-6000	13.74	13.9	16	0.99	0.99	0.98
60%	50	-2E7	-2E7	-4E6	-1344.8	-3024.9	-10035	14.01	14	14.99	0.97	0.99	0.98
65%	50	-2E7	-2E7	-4E6	-2912.1	-3153.2	-10764	14.27	14.28	15.08	0.99	0.99	0.98
Water	50	-1E7			-6688.4			20.186			0.98		

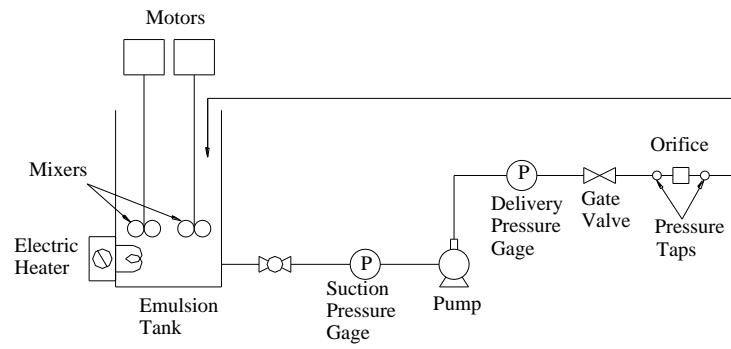
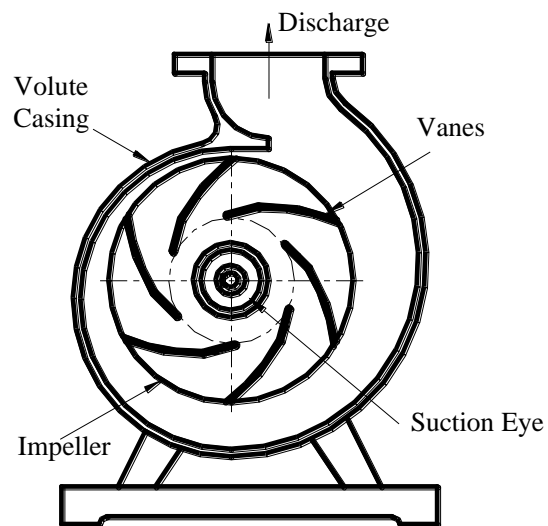


Fig. 1 The flow rig



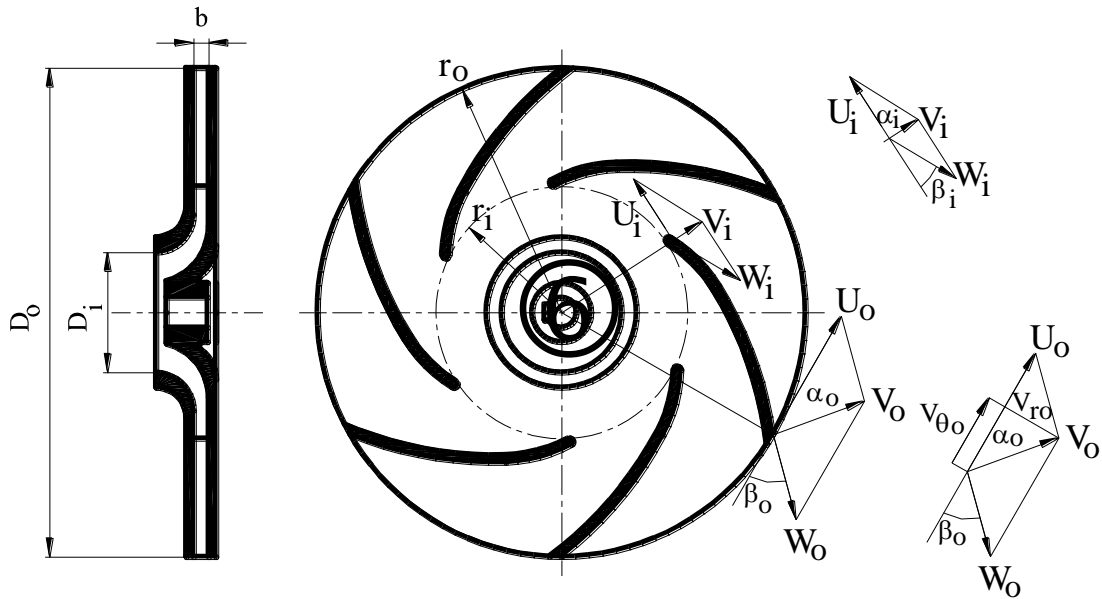
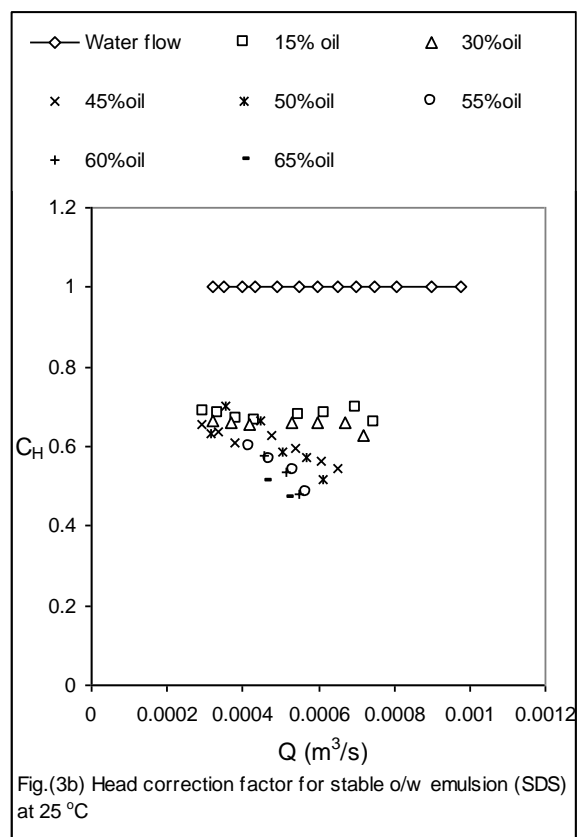
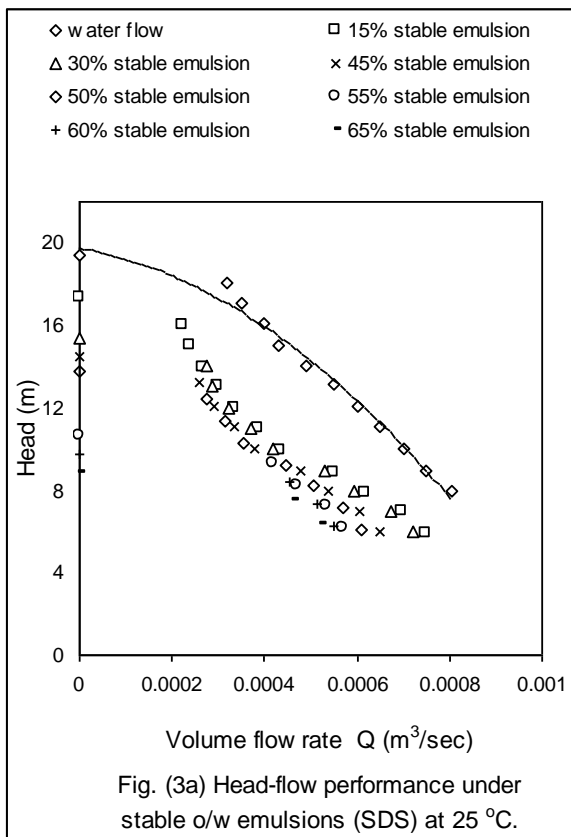


Fig. 2 Centrifugal Pump



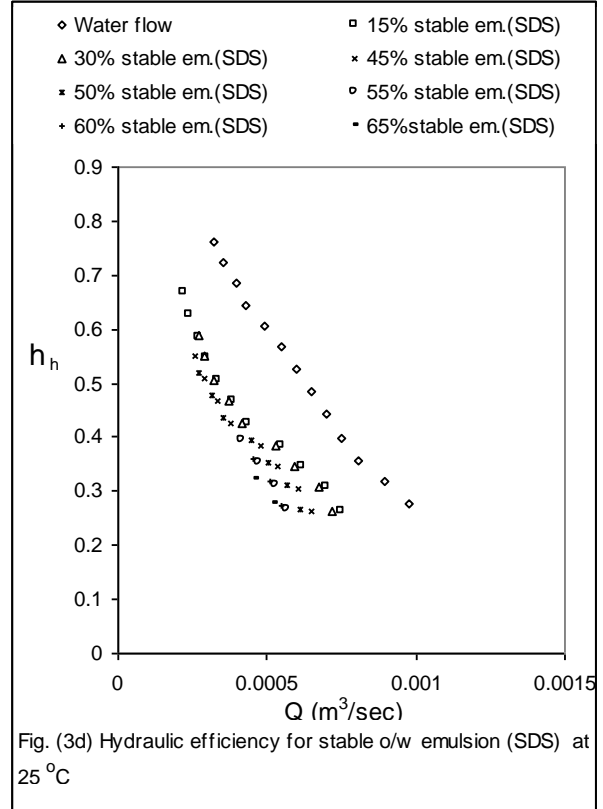
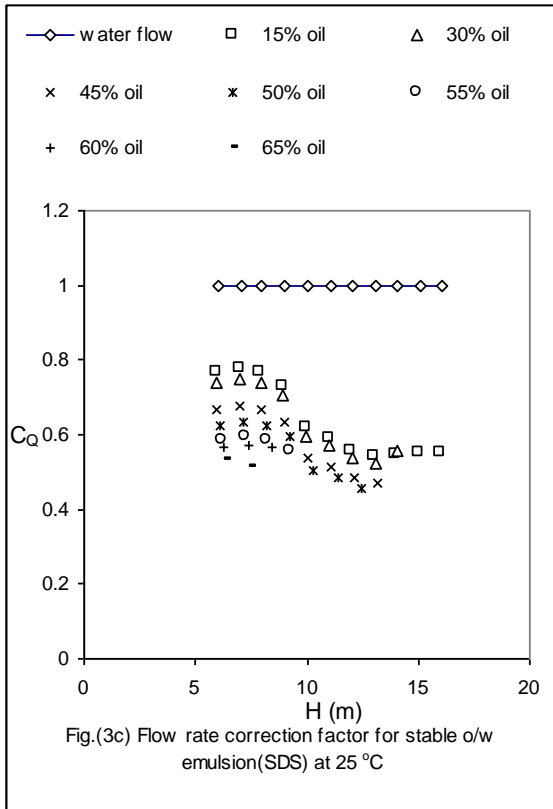
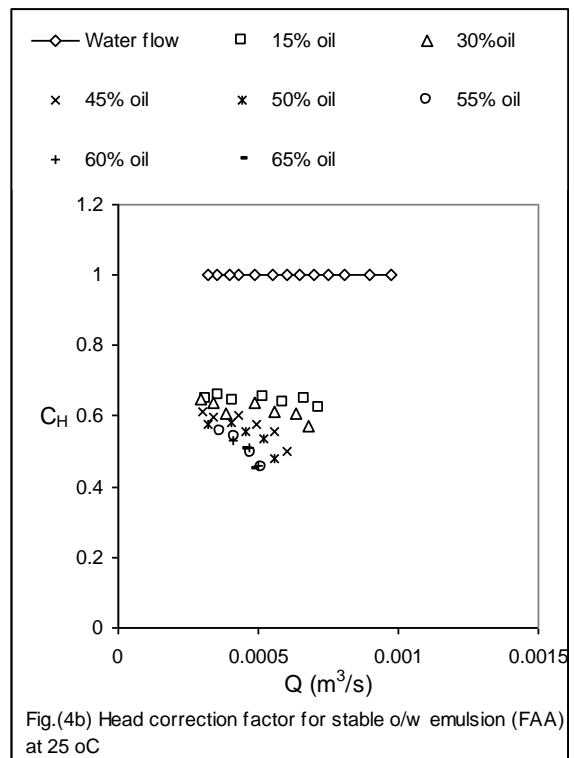
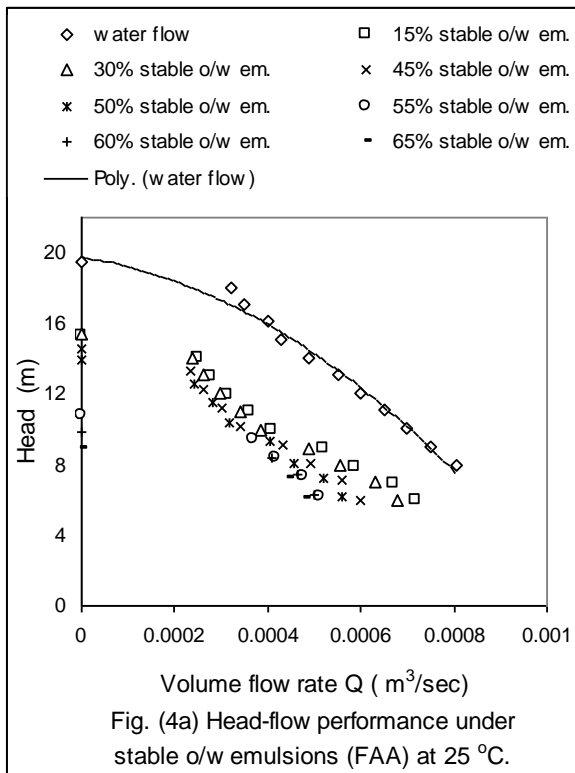


Fig.3 Pump performance for stable o/w emulsion (SDS) at 25 °C



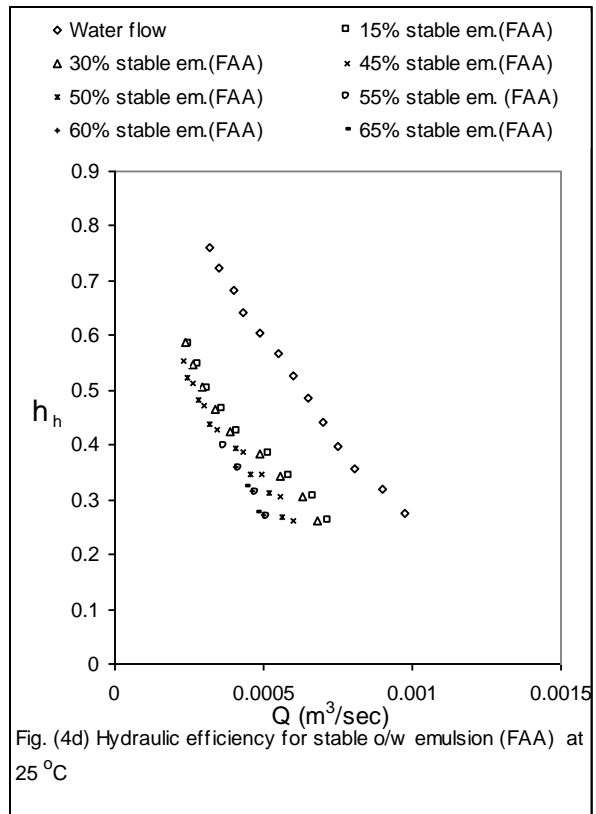
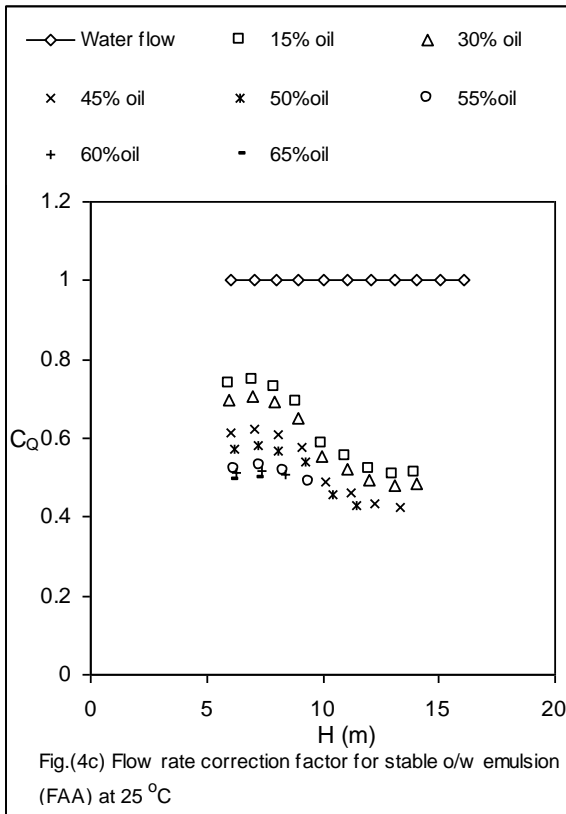
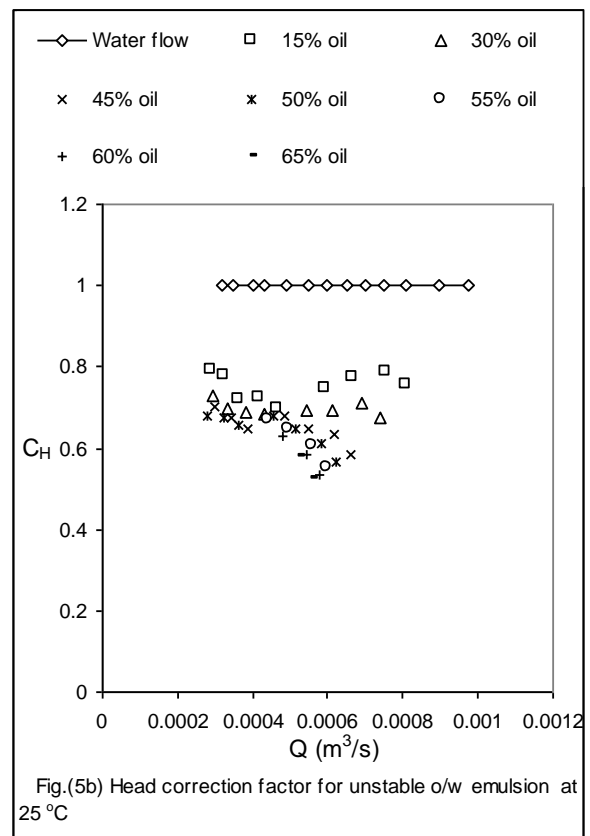
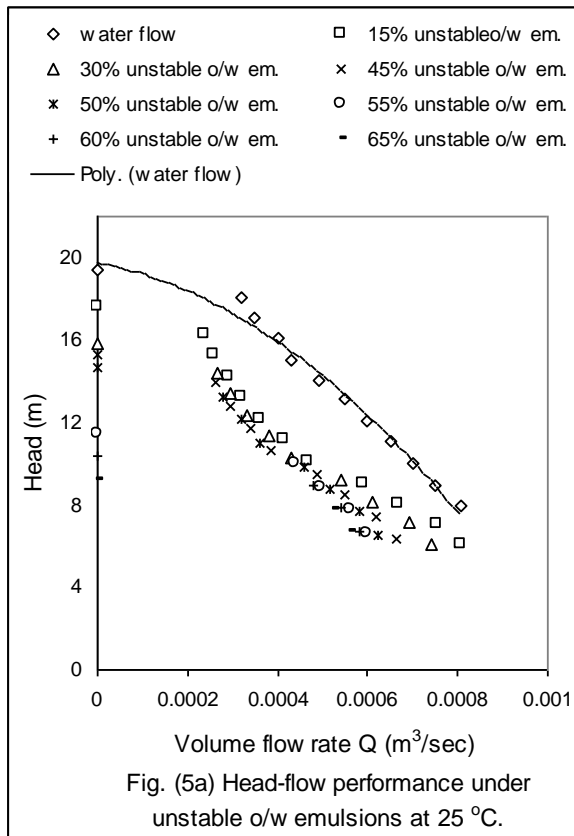


Fig. 4 Pump performance for stable o/w emulsion (FAA) at 25 °C



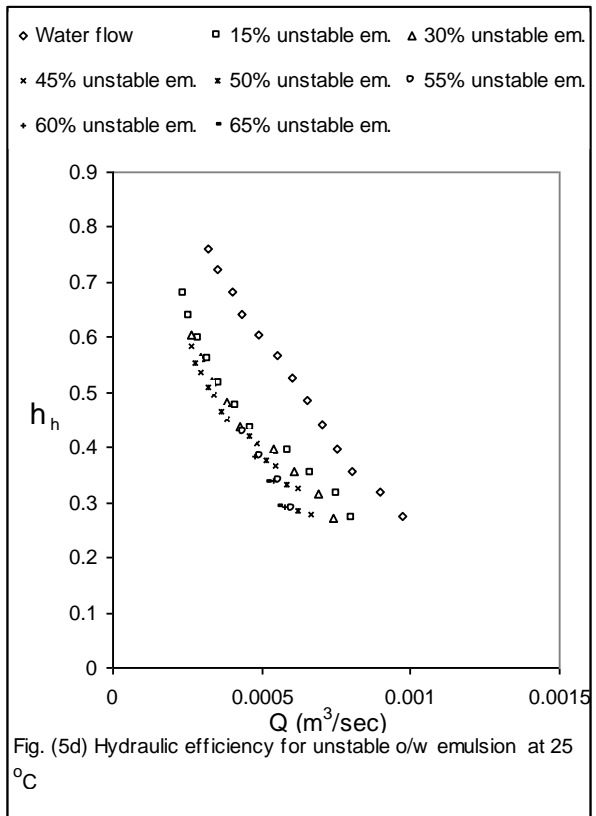
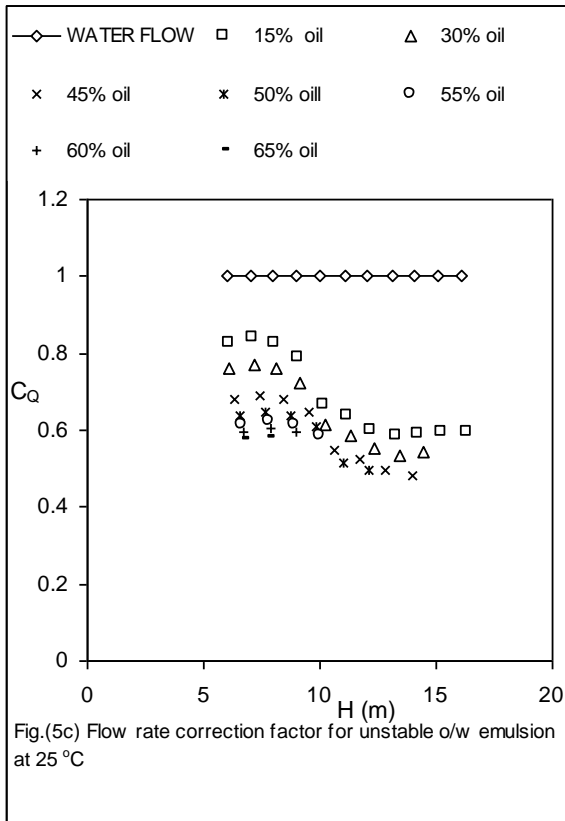
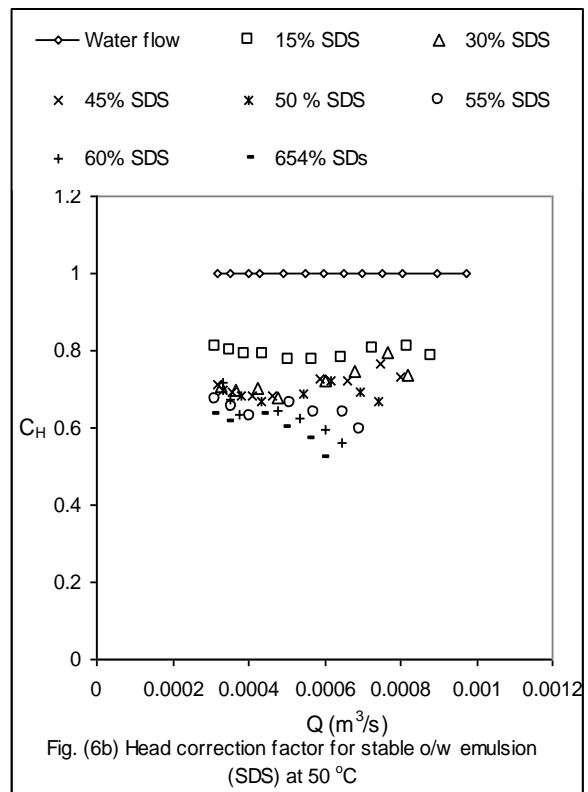
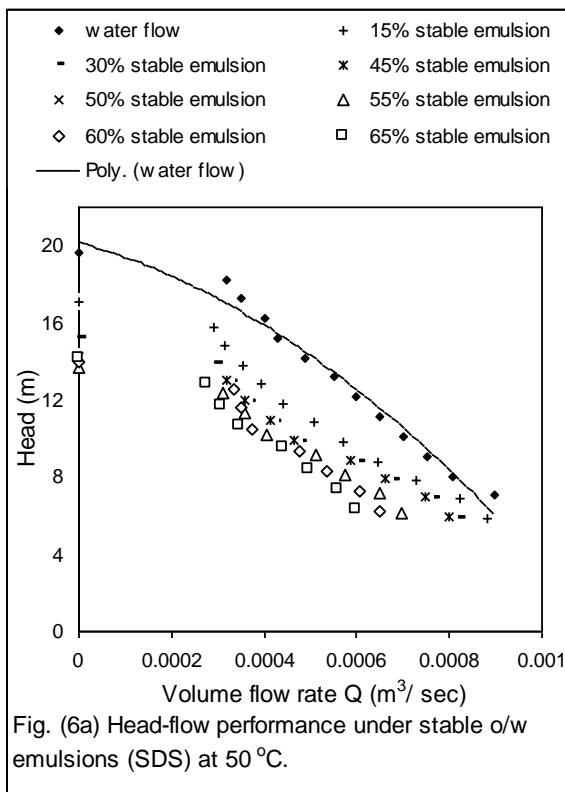


Fig. 5 Pump performance for unstable o/w emulsion at 25 °C



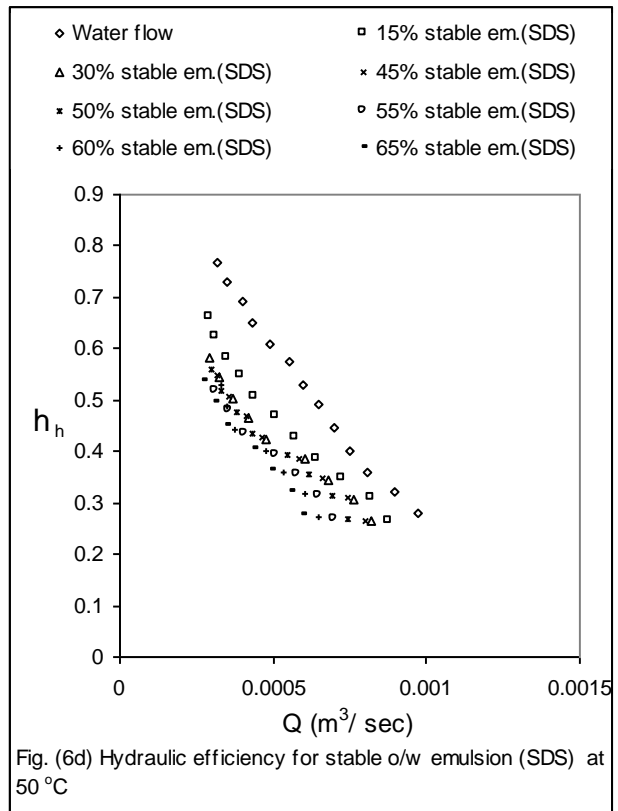
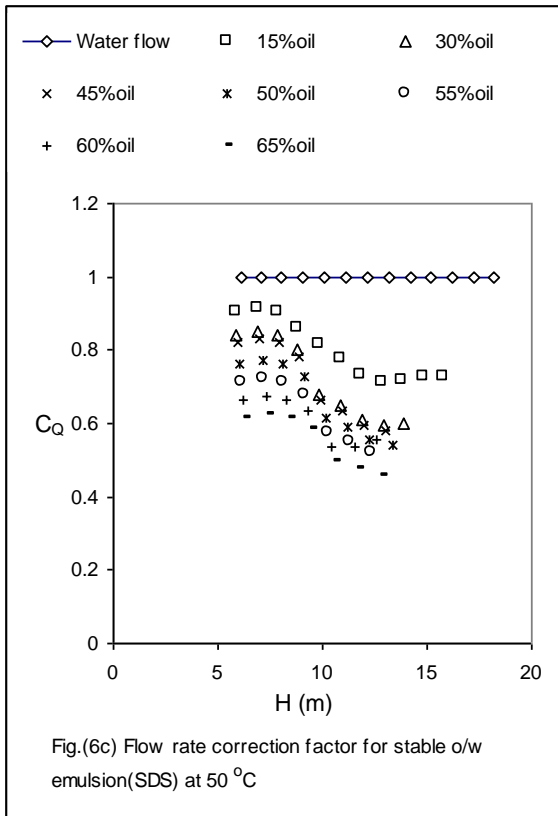
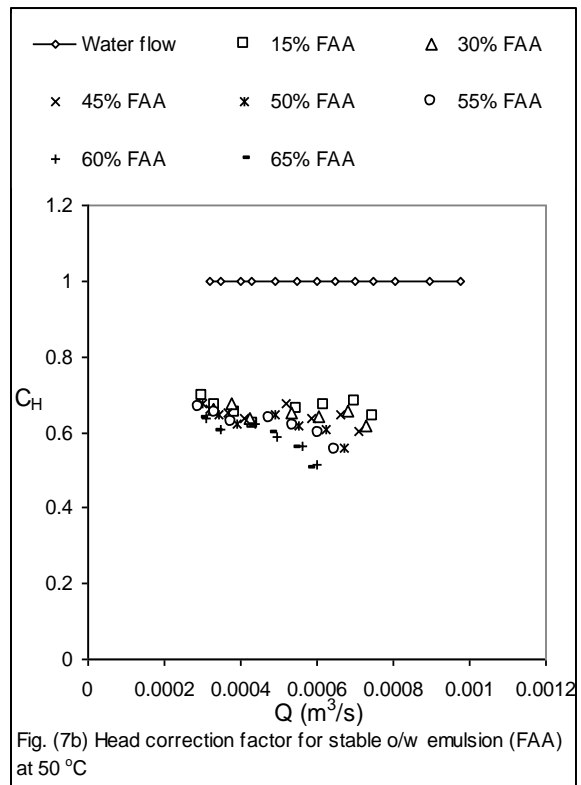
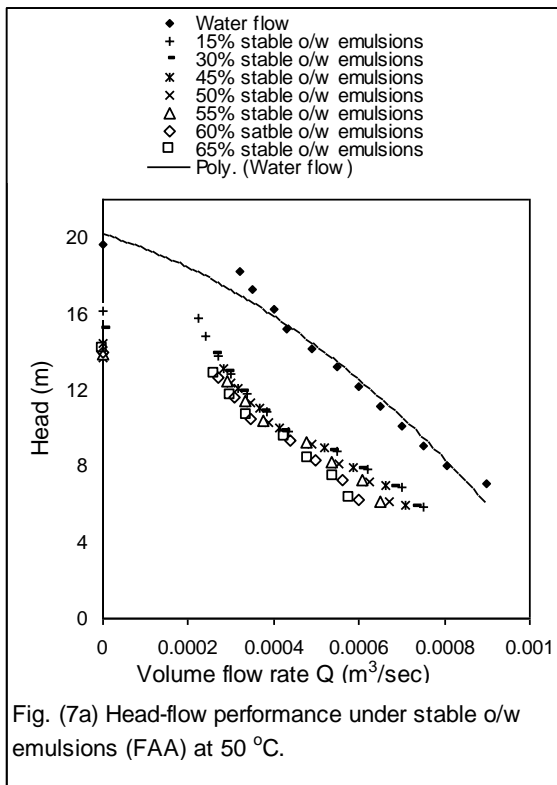


Fig. 6 Pump performance for stable o/w emulsion (SDS) at 50 °C



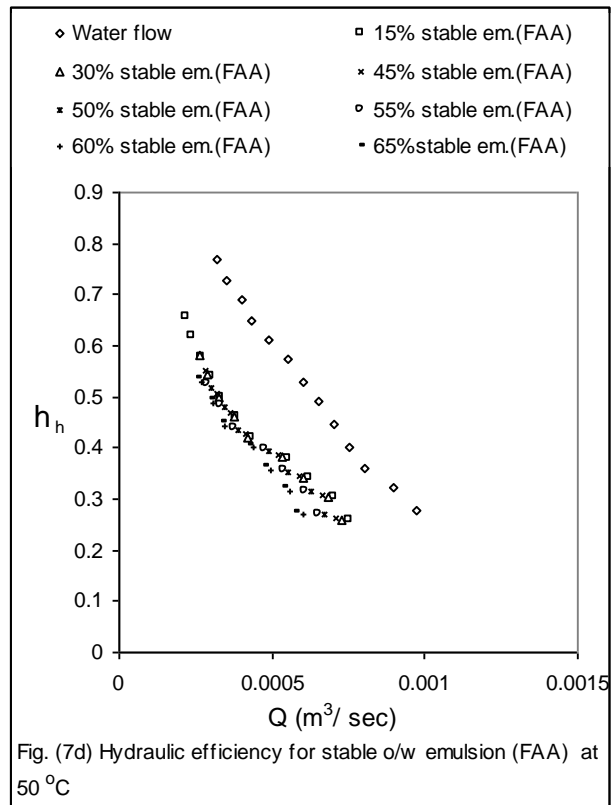
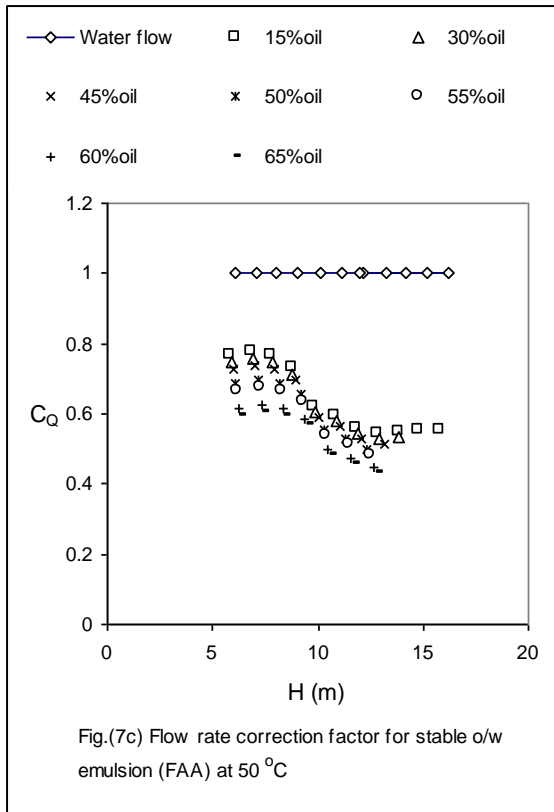
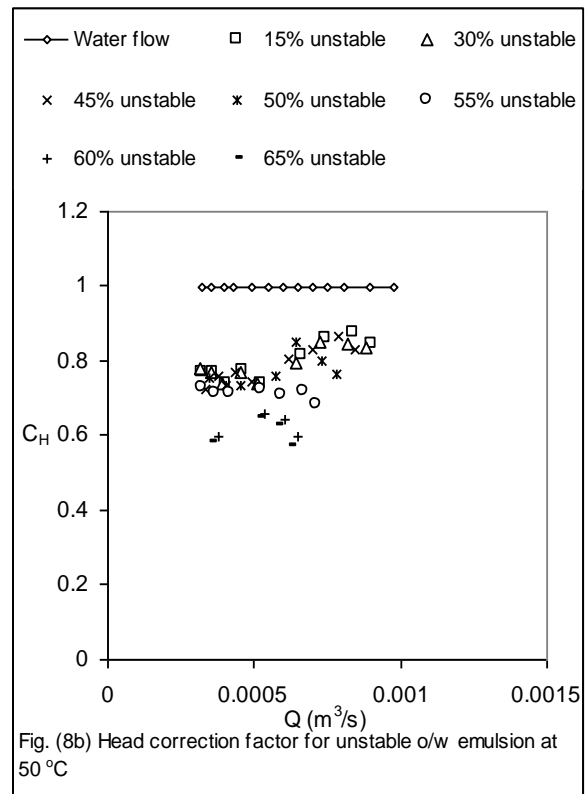
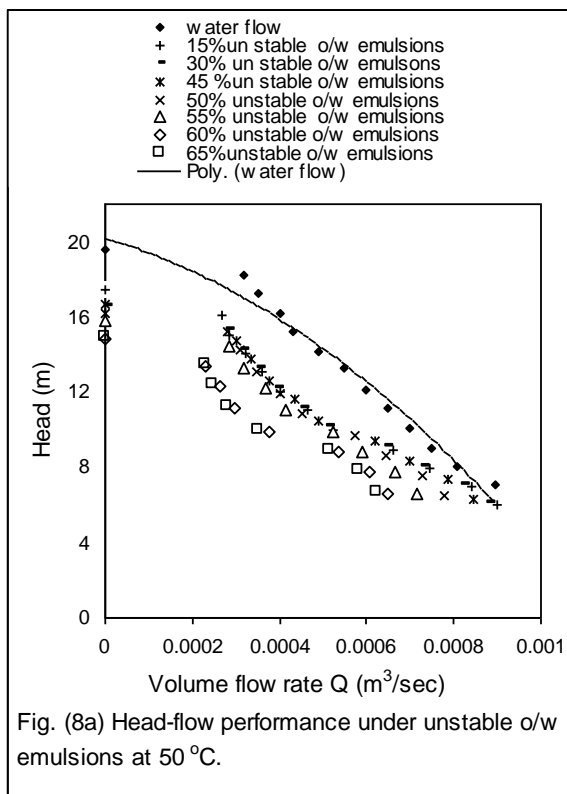


Fig. 7 Pump performance for stable o/w emulsion (FAA) at 50 °C



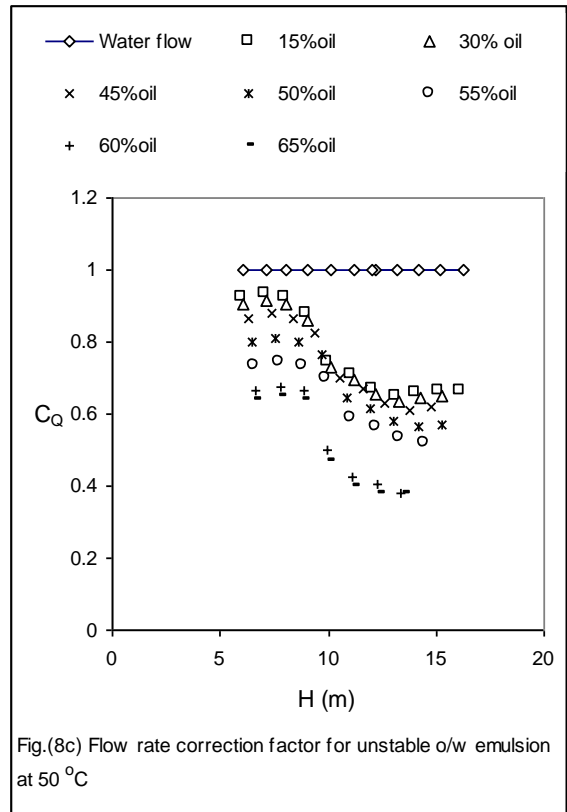
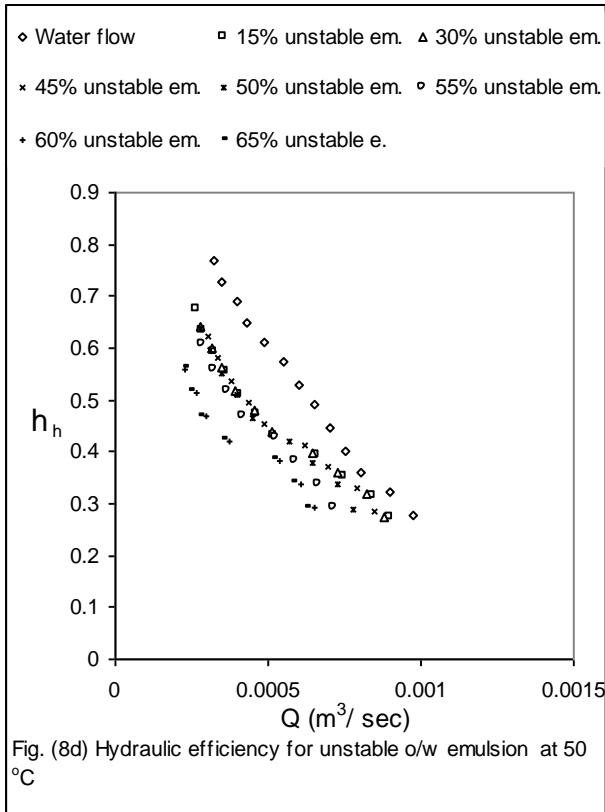


Fig. 8 Pump performance for unstable o/w emulsion at 50 °C