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ENTHALPY BALANCE MODEL LEADS TO MORE ACCURATE MODELLING OF HEAVY OIL PRODUCTION WITH AN ELECTRIC SUBMERSIBLE PUMP

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An investigation of the impact of heat dissipated by a pump system on the overall performance of the oil well has been carried out using a dynamic, non-linear electric submersible pump (ESP)/well model based on a systems dynamics modelling package. This Package allowed rapid model development for complex well situations. It included building a detailed ESP model. Comparison of results from the detailed ESO model with commercial software gives good agreement. The dynamic ESP/well model also gives good numerical agreement during steady-state production as well as reproducing the unstable production behaviour observed in the field. A detailed study showed that the heat dissipated by a down-hole motor and pump had a significant impact on the overall production performance of the extra heavy and heavy oil wells when the magnitude of the change in viscosity of the crude oil with respect to change in temperature was sufficiently large. This was particularly true for viscous oils, but this effect could be important even for lighter North Sea crude oils under unusual circumstances, e.g. pressure boosting with an ESP in a subsea pipeline when the crude oil is cold.

Keywords: electric submersible pump; viscous oil; dynamic modelling; pump performance.

INTRODUCTION

The behaviour of viscous oils and oil–water mixtures as they pass through an electric submersible pump (ESP) is poorly understood. The literature contains a number of examples illustrating unusual behaviour by ESPs e.g.

(1) observations of pump 'cycling' during very heavy oil production from the Beta Field, Offshore California by Carpenter and McCrea (1995);
(2) Lagoven, S. A. (Venezuela) successfully produced extra-heavy oil (API gravity 8.5 and viscosity 2500 cp at reservoir conditions) from the Jobo and Cerro Negro Fields at 700–1300 bopd/well using ESPs when Joubet *et al.* (1996) showed that current pump design procedures suggest that the overall pump efficiency should have been virtually nil.

Figure 1 is a simplified view of the hydraulic, heat and electrical energy flows that make up an ESP system operating in a well. The reservoir fluid flows into the well at the bottom (through the perforations) and leaves the system through the wellhead. Electrical energy is supplied to the system, part of which is converted into mechanical energy by the centrifugal pump, driving the liquid to the surface. The remainder of the energy is dissipated in form of heat, warming the fluid as it passes through the ESP. Centrifugal pumps become less efficient as the viscosity of the fluid being pumped increases. Hence the efficiency of the pump (converting shaft energy to hydraulic-potential energy) will

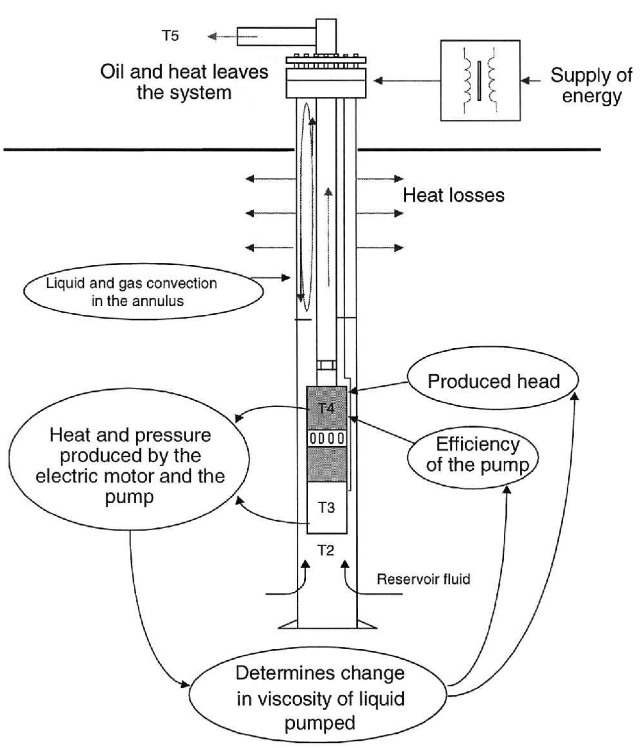

Figure 1. Energy balance in a well equipped with an ESP.

increase as the temperature rises and the fluid viscosity decreases, increasing the oil production.

Field experience has shown that existing ESP performance models often under-predict their heavy oil pumping performance. This paper presents the outcomes of a study aimed at determining if the reduced viscosity, due to heat generated by the pump, could significantly increase the oil production from the Jobo and Cerro Negro Fields while also causing the unstable oil production observed in the Beta Field. An improved model will lead to better pump sizing for field operations while reducing the overall cost of production and increasing the recoverable oil reserves.

The standard ESP design procedure for viscous fluids does not take into account the heat produced by ESP on the pump performance (Brown, 1985; Golan and Wilson, 1991). Little data is available in the literature concerning the impact of heat dissipated by ESP pump system on the overall performance of the oil well. Hence the study had to develop a methodology to allow improved understanding of the effects of the heat phenomena on ESP performance.

Before building the integrated ESP well model the system details are examined. These are: (i) reservoir fluid properties; (ii) centrifugal pump; and (iii) electric motor.

THE RESERVOIR FLUID PROPERTIES

The properties of the fluid for a particular reservoir are described by the fluid's pressure–volume–temperature (PVT) properties. Published correlations were used, most of which rely on the oil's API gravity as their basis. One (arbitrary) classification system for crude oils is by De Ghetto *et al.* (1994):

(1) extra-heavy crude oil—API° <10;
(2) heavy crude oil—10< API° <22.3;
(3) medium crude oil—22.3< API° <31.1;
(4) light crude oil—API° >31.1.

Oil PVT Correlations

The PVT correlations to estimate fluid viscosity prediction for heavy oil have been summarized by Bennison (1998). The equations used are given in Appendix 1 and are summarized as follows:

(1) extra heavy dead-oil viscosity is described by the modified Egbogah–Jack's correlation;
(2) gas-saturated oil viscosity is described by the modified Kartoatmodjo's correlation;
(3) undersaturated oil viscosity is described by the modified Labedi's correlation; and
(4) bubble point pressure is calculated according to Standing's correlation.

Figure 2 compares viscosity vs temperature prediction with actual data for an extra heavy, dead, 9° API gravity crude oil from the Orinoco Bitumen Belt crude. This figure shows there can be large errors between published correlations and data measured on actual field samples (Torres, 1997).

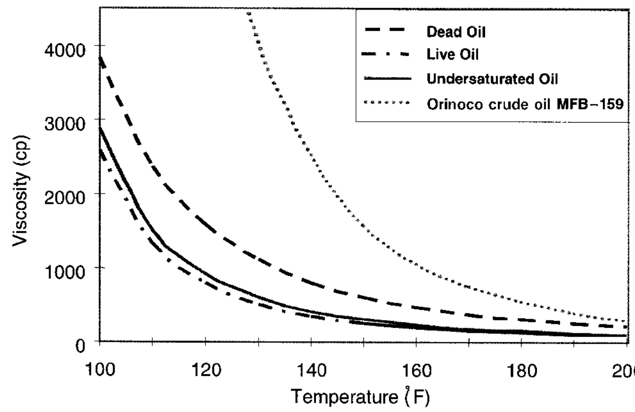

Figure 2. Comparison of viscosity predicted for extra heavy oil by De Ghetto *et al.* (1994) as a function of temperature with actual Orinoco crude oil data (Torres, 1997).

Specific Heat of the Oil

The specific heat is required for calculating the temperature following the input of a given amount of heat. It has been evaluated using the following equation quoted by Aquilera (1991):

$$C_v = \frac{0.388 + 0.00045 \cdot T}{\sqrt{\begin{array}{l}(141.5/131.5 + \mathrm{API}) \\ -(0.3444 \cdot T - 11.022/1000)\end{array}}}$$

where C_v is the specific heat (BTU lb^{-1} R^{-1}), API the API crude gravity (°API) and T the temperature (°F).

ENERGY BALANCE FOR AN ESP

Overall efficiency of an ESP system is evaluated by calculating the ratio of the produced hydraulic horsepower and the supplied electric power. Theoretically, the efficiency of an ESP pumping water can reach as high as 50%. Field test results in the Daqing Oil Field reported by Zhen-Lhua and Zhu (1996) show that the actual system efficiency is often much lower, varying from 15 to 40% for shallow oil wells. Energy losses can be subdivided into the surface and downhole losses. The surface energy losses are usually 2–3% for constant speed pumps and 5–15% if a variable frequency drive (VFD) system is fitted.

The surface energy losses have not be considered any further since they do not affect the downhole performance of the pump—only downhole energy losses can be converted into heat that will reduce the viscosity of the fluid flowing through the well.

The efficiency of the ESP system is defined as:

$$\frac{\text{Hydraulic head produced}}{\text{Potential head from electrical input}}$$

The main components of the ESP system where energy losses occur during operation are the electric cable, submersible electric motor, multi-stage centrifugal pump and tubing string.

Cable Losses

Cable losses can be calculated from the working current. Evaluation of the portion of heat dissipated by the cable that contributes to an increase in temperature of the oil flowing in the tubing is difficult. This is because the cable is strapped at (ir)regular intervals to the tubing string and installed in the casing/tubing annulus. The temperature rise will depend on the completion method and the heat exchange processes occurring in the annulus. However, the effective heat generated by the down-hole cable is relatively small and will be neglected in the further analysis.

Electric Motor

Factors contributing to the efficiency of the electric motor, and hence the heat production, are as follows:

(1) the VFD generates a pseudo sine wave voltage waveform; the current harmonic generates increased motor losses (of the order of 4%);
(2) running at elevated frequencies also increases losses; resistive heating in the windings and all rotor losses remain constant, while inductive losses are roughly proportional to frequency;
(3) friction losses in the oil gap are approximately proportional to the square of the motor speed and hence contribute to an increase in the total (percentage) losses at higher speeds;
(4) the power factor, reflecting the difference in phase of the currents and voltages in the electric motor winding, may also affect the heat generated; at light shaft loads the power factor is very low, resulting in increased energy losses.

A constant electric motor efficiency of 90% was used for the remainder of the study, although a more detailed model could have been included based on published data.

The Centrifugal Pump

Each stage of the multistage centrifugal pump adds energy to the fluid in form of increased velocity and pressure. The impeller accelerates the fluid, increasing the kinetic energy, which is then converted into potential energy (pressure) in the diffuser. The fluid flow is described by the fundamentals of classical physics—conservation of mass, momentum and energy. The energy balance equation adiabatic flow can be written as:

$$\frac{P}{\gamma} + \frac{v^2}{2 \cdot g} + Z = \text{constant}$$

The symbols are defined in the Nomenclature. Note that: (P/γ) is the static pressure head, $(v/2{\cdot}g)$ is the velocity head and Z is the potential head (all having the units of length). The fluid static pressure rise across the pump is related to: (i) the change in the relative velocity of the fluid onto and off of the impeller blade; and (ii) the change in absolute velocity of the rotating impeller at the points where the streamlines are incident onto and leave the impeller blade.

An impeller operating at a given speed will generate the same amount of head irrespective of the fluid's specific gravity. The head produced by the pump varies with the quantity of fluid pumped, but not with the density. The discharge pressure, however, does vary with density. A given centrifugal pump develops the same head at a given capacity, irrespective of the fluid being pumped (this does not apply to gas—flow through the pump will cease if gas is present in significant quantities due to gas locking caused by the high compressibility).

The actual head developed by a pump is always less than the theoretical head due to energy losses.

Overall Pump Efficiency

The efficiency of a centrifugal pump is the ratio of fluid horsepower produced to shaft brake horsepower consumed. The fluid horsepower is the energy absorbed in the fluid leaving the pump, i.e. that part increasing the static and velocity pressure heads. The shaft brake horsepower is the total pump power required to supply the energy for pumping the fluid and to overcome all energy losses that occur in the pump. These losses include flow friction through the impeller and associated turbulent losses, the disk friction, the leakage of fluid from the outlet back to the eye of the impeller and mechanical friction losses in the bearings, and stuffing boxes. The relative importance of these various sources of energy loss as a fraction of optimum (design) pump rate are given in Figure 3.

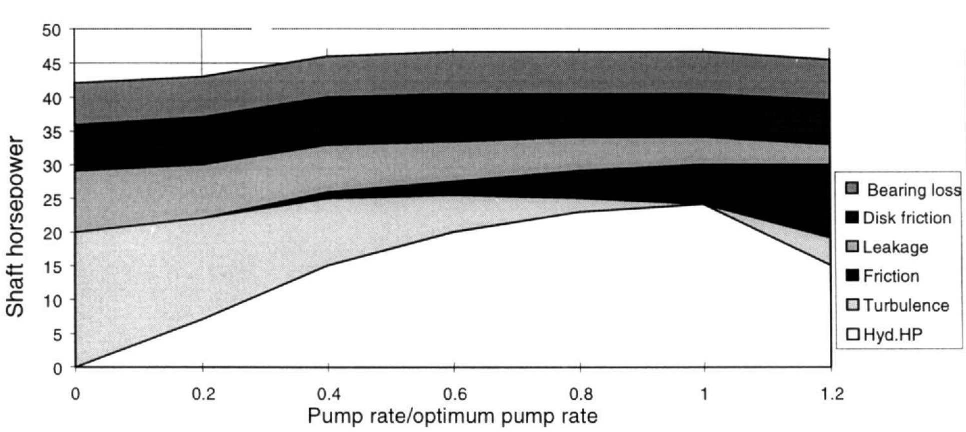

Figure 3. Efficiency in converting input mechanical shaft power to useful fluid power.

Eventually all energy losses which occur in the pump are converted into heat, increasing the temperature of the fluid being pumped.

Viscosity Effects

ESP performance is greatly reduced when pumping highly viscous oils when compared with the efficiency achieved when pumping low viscosity fluids. This efficiency drop appears as a reduced flow capacity, a decreased head and a lower overall pump efficiency.

Viscosity, like other physical properties of liquids, is affected by the fluid composition as well as by the physical conditions of pressure and temperature. An increase in temperature causes a decrease in viscosity, while an increase in pressure causes an increase in viscosity. Pumping viscous oil with an ESP under low efficiency conditions will result in significant energy losses from the pump system that appear as an increased liquid temperature and reduced liquid viscosity.

The American Hydraulic Institute have published correction factors, (Capacity-F, Head-F, Eff-F) to determine the performance of a conventional sized submersible pump handling a viscous liquid when its performance with water is known. These are available through Brown (1985). These correction factors data can be used as an approximate guide in the sizing of a pump and motor for a given application. They are as follows:

$$\text{Capacity-visc} = \frac{\text{Capacity-F} \cdot \text{capacity-water}}{100}$$

$$\text{Head-visc} = \frac{\text{Head-F} \cdot \text{head-water}}{100}$$

$$\text{Eff-visc} = \frac{\text{Eff-F} \cdot \text{efficiency-water}}{100}$$

The change in these pump correction factors as a function of fluid viscosity is illustrated in Figure 4. The kinematic viscosity is:

$$\kappa = \frac{\text{Absolute viscosity, } \mu}{\text{specific gravity, } \gamma}$$

Pump Mathematical Model

The mathematical relationships for the pump producing head and corresponding efficiency were derived as: head = function 1 (rate, speed, viscosity); efficiency = function 2 (rate, speed, viscosity). Ionel (1986) showed that the head–capacity relationship could be expressed as a second-order equation:

$$\text{Head} = A1 \cdot \left(\frac{F}{60}\right)^2 + B1 \cdot \frac{F}{60} \cdot Q + C1 \cdot Q^2$$

where F is the pump speed (frequency) while Q is the flow rate. Note that:

(1) the term $A1$ represents the theoretical head created by a pump; $A1$ can be derived from the pump head–capacity curve supplied by the manufacturer;

(2) the term $B1$ represents a linear, ideal pump relationship between the pumping head and the capacity; $B1$ can be derived from a linear regression analysis of the pump head–capacity curve;

(3) the term $C1$ represents all the energy losses occurring in a pump, including those described above; it is a function of viscosity;

(4) the pump efficiency (Eff) can now be expressed, as quoted by Ionel (1986), by:

$$\text{Eff} = \rho \cdot Q \cdot \frac{A1 \cdot (F/60)^2 + B1 \cdot (F/60) \cdot Q + C1 \cdot Q^2}{A2 \cdot (F/60)^2 \cdot Q + B2 \cdot F/60 \cdot Q^2 + D \cdot (F/60)^3}$$

(5) the determination of the performance characteristics of an actual pump requires the coefficients $A2$, $B2$, D; these are determined from at least three points on the pump performance curve based on measured data supplied by the manufacturer.

Results for single-stage pump mathematical model

The above methodology—which describes the performance of a single stage of an ESP–was implemented. The results are summarized in Figures 5–7. These curves illustrate how typical pump head–capacity and pump

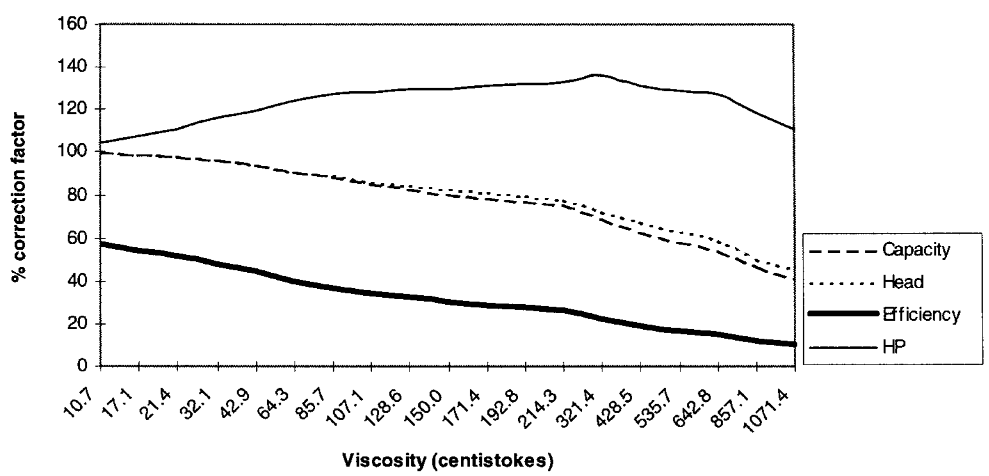

Figure 4. Pump correction factor as a function of fluid viscosity.

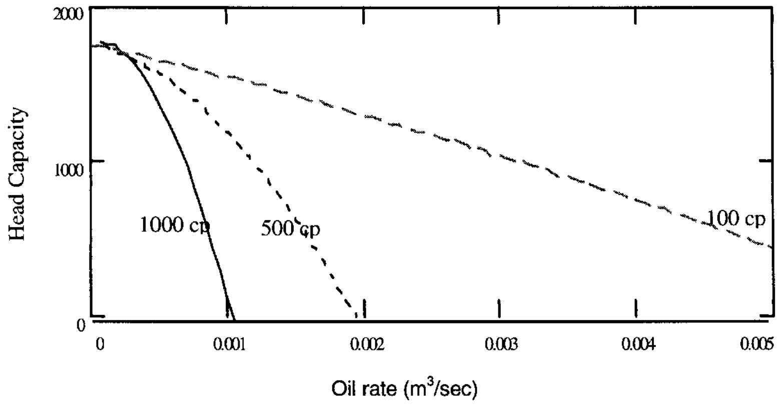

Figure 5. Typical head capacity as a function of pump rate and crude oil viscosity.

efficiency vary with the liquid viscosity (over the range of 1000–100 cp) and electrical supply frequency (over the range 80 Hz–40 Hz) as a function of pump rate.

COMPUTER MODELS

Two models were created—a stand-alone multi-stage ESP model called 'PumpCalc' and a complete ESP/well dynamic model. Commercial software is available to model the ESP performance. It is field proven, but is also a 'black box' whose internal calculation procedures cannot be altered by the user. The first stage of being able to evaluate the impact of including the enthalpy balance effect is to build a model of a 'conventional' pump.

The *PumpCalc* Multistage ESP Model

The down-hole pump and motor heats the produced oil, reducing its viscosity. This viscosity reduction changes the pump parameters (viz. head, pump efficiency etc.). The system thus exhibits non-linear behaviour. A seeded iteration procedure in which the first element of an array is specified followed by computing successive elements, based on the value of this first element, was chosen to solve this problem.

PumpCalc calculates the fluid temperature; pressure and specific heat are first specified at the pump inlet conditions. The fluid properties, electric motor and pump parameters in the form of correlations, mathematical equations or laboratory-determined data are also required. The value of the following variables are calculated for each pump stage:

(1) the liquid temperature in the each pump stage;
(2) the efficiency for each pump stage;
(3) the liquid temperature and viscosity at the discharge of each pump stage;
(4) the overall pump efficiency, produced head, required shaft horsepower and the electric power requirement.

Calibration of the *PumpCalc* model against available commercial software gave good agreement. *PumpCalc* could be developed further, e.g. for inclusion of gas effects.

The PumpCalc model assumptions

(1) The fluid properties (viscosity, pressure and temperature) are assumed to be constant within each stage;
(2) the fluid viscosity within each stage (at the current values of temperature and pressure) is calculated from the PVT correlations;
(3) the viscosity correction for the fluid properties (produced head, brake horse power requirement and overall efficiency) for each individual stage is made using the American Hydraulic Institute correction factors referred to earlier;
(4) all energy losses are converted into heat;

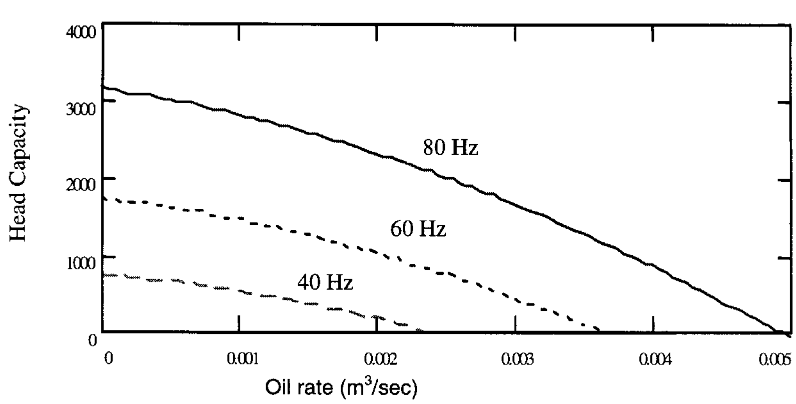

Figure 6. Typical head capacity as a function of electrical supply frequency.

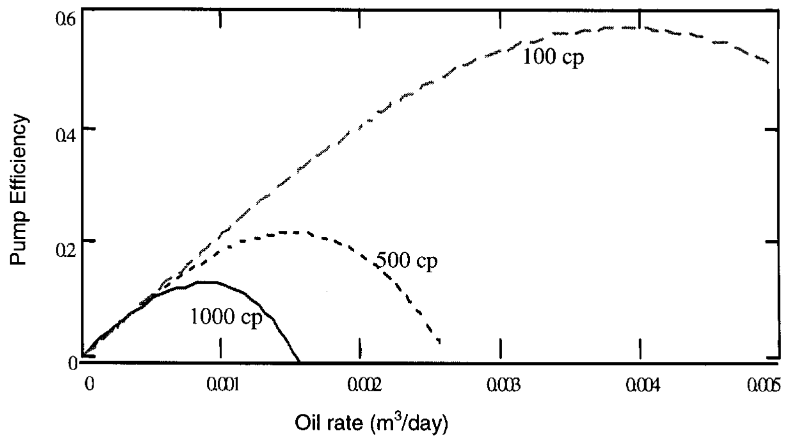

Figure 7. Typical pump efficiency as a function of viscosity and pump rate.

(5) all generated heat is transferred to the fluid passing through that stage; heat losses to the surroundings—casing, completion fluid, formation etc.—and the heat capacity of the pump itself are assumed to be zero;

(6) the fluid temperature rise (ΔT) in each stage is calculated from:

$$\Delta T = \frac{\Delta \mathrm{Heat}}{C_{\mathrm{v}} \cdot \mathrm{Mass\ flow}}$$

The calculation routine was created in the Mathcad 6 Plus for Windows environment. Figure 8 is the flow diagram illustrating the logic behind the computer program. Typical results from these calculations are presented in Figure 9. This illustrates the changes in fluid temperature and viscosity as it progresses through the various pump stages.

The ESP/Well Dynamic Model

Fluid inflow into the well has been assumed to be a constant temperature process. It was modelled as a straight-line, well drawdown pressure vs production rate relationship. This is independent of the pump performance, apart from the drawdown imposed on the well by the pump's operation.

By contrast, the fluid outflow through the tubing is strongly dependent on the viscosity, and hence on the temperature of the producing formation, i.e. on the well's vertical depth, production rate, heat losses through the tubing etc. This interaction between the fluid flow rate and the resulting pressure drop adds further non-linear elements to the system.

An integrated ESP/well computer model that includes all the mathematical models discussed above is required. Building an error-free, conventional (e.g. FORTRAN) computer model for such a complicated, interrelated system is a major challenge. *Stella*, marketed by High Performance Systems Inc. of the USA, allowed us to rapidly build an ESP/well dynamic, non-linear computer model that included all the hydraulic, heat and electrical elements illustrated in Figure 1 and discussed above.

The Stella ESP/well model

(1) The well was subdivided in eleven control volumes. Five control volumes were assigned for the tubing, four to the multistage pump, one for the electric motor, and one for the well bore.

(2) The mass of reservoir fluid and the heat content in each control volume were selected as the main model state variables.

(3) The main flow controls that will change the state variables are: (a) *the producing formation* supplies liquid into the well at a constant temperature; (b) *the electric pump* produces the pressure boost that drives the fluid flow up the tubing as well as introducing heat into the system (from the produced fluid, at reservoir temperature, and energy losses from the pump inefficiency); (c) *the viscosity effects* on the pump producing head and efficiency are modelled; and (d) *the tubing* is modelled as the recipient of fluid mass and heat energy from the system. The frictional pressure losses in the tubing are described by the conventional laminar flow equations. The heat exchange with surroundings is described by Shiu's empirical correlation for flowing wells quoted by Aquilera (1991).

(4) The assumptions of mass, energy, and momentum conservation provide the basis for the differential equations used in the model. This is achieved via balance sheets that are set up to account for inflow and outflow and any accumulation of matter and energy within each control volume.

(5) The program was rewritten in SI units.

COMPUTER MODELLING RESULTS

The results of the two models—the stand-alone, multi-stage ESP model and the complete ESP/well dynamic model—are discussed separately below.

Stand-alone ESP Model

An example set of sensitivity calculations was run to illustrate the variables controlling the overall pump efficiency, producing head, electrical power requirements. An oil flow rate of 4340 bpd and a 118-stage pump were chosen. The fluid being pumped (i.e. different crude oils)

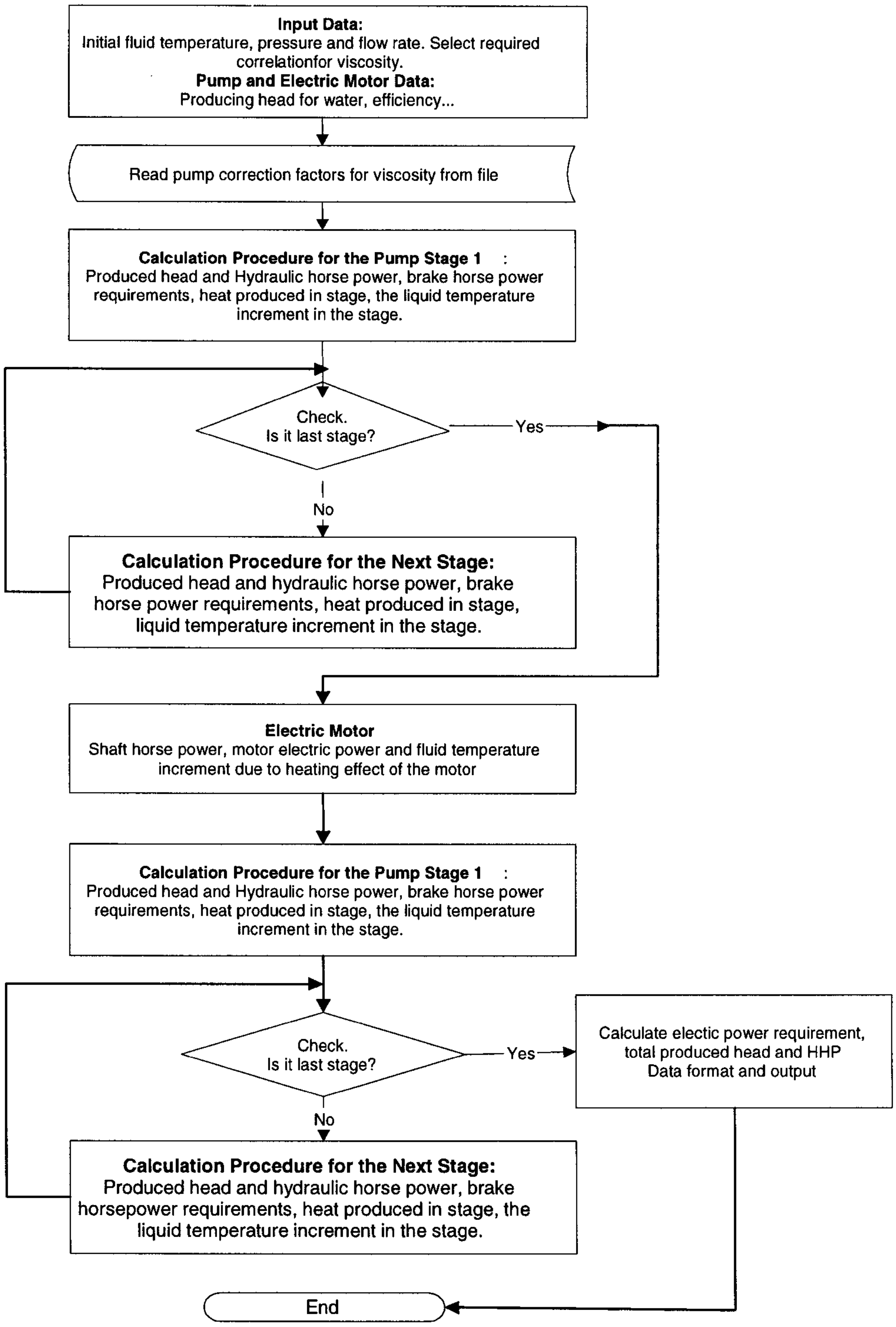

Figure 8. The calculation flow diagram for the program *PumpCalc*.

and the reservoir temperature were varied over a substantial range.

The results indicated that, for the arbitrary conditions chosen, the produced oil temperature increases by 5–10°C between the pump intake and discharge point, due to the heating effect from energy losses in both the motor and the pump. The viscosity of the fluid can be substantially reduced due to this temperature increase. As this occurs, the efficiency of each pump stage and the produced head per stage will increase.

The conventional pump system design procedure described by Brown (1985) and Golan and Wilson (1991) treats all the pump stages as being equivalent and does not take into account these phenomena. The accuracy of the predicted pump performance is substantially reduced if the properties of the fluid being lifted change substantially between the ESP inlet and discharge points. The degree of detail required, i.e. whether it is necessary to calculate the fluid properties for each stage or whether a number of stages can be lumped together, will depend on the conditions being studied.

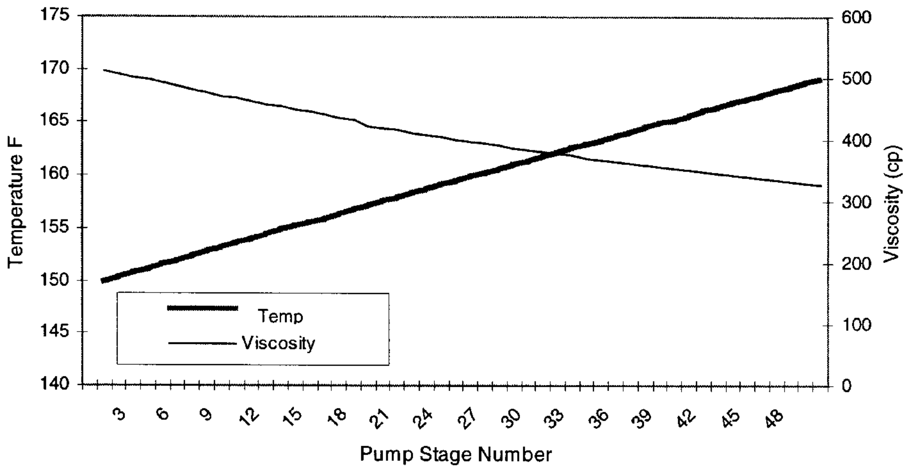

Figure 9. Temperature and viscosity along the pump stages.

The results

The value of correcting for these changes in fluid properties as the fluid progresses through the pump is illustrated by comparing the pump parameters (head, efficiency, horse power requirements) with constant and enthalpy balance corrected fluid properties. The impact of this correction was evaluated using the 'head ratio', where:

$$\text{head ratio} = \frac{\text{Uncorrected} - \text{head}}{\text{Corrected head}}$$

A second parameter that was found to be significant was called the derivative, which measures the change in oil viscosity per unit temperature change.

$$\text{Derivative} = \frac{\Delta\text{viscosity}}{\Delta\text{temperature}}$$

The results in Table 1 are from calculations in which the heavy oil type was fixed at 10°API gravity while the temperature was varied. Typical results are:

Temperature	Viscosity	Derivative	Head ratio
110°F	679 cp	−28 cp/°F	0.882

In Table 2 the heavy oil properties (based on API gravity) were varied while the temperature was fixed at 87.5°F.

Temperature	Viscosity	Derivative	Head ratio
87.5°F	502 cp	−24.4 cp/°F	0.893

Analysis of the results

Sensitivity calculations indicated that the importance of the full stage-by-stage description of the pump performance depends not on the magnitude of the liquid viscosity, as one would instinctively expect, but on the *magnitude of the change in viscosity with respect to the change in temperature.*

Tables 1 and 2 indicate that either calculation technique (with/without fluid properties changing per pump stage) gives essentially the same (within 10%) result providing the rate of change in the oil viscosity with temperature, when evaluated at average pump temperature, is less than 25 cp/°F. If an accuracy of 5% or more is required, this factor should be reduced to 8 cp/°F at average pump temperature. This conclusion is based on the arbitrary starting conditions described above.

This temperature effect can be important even for lighter, North Sea crude oils (21°API gravity) under unusual circumstances, e.g. pressure boosting with an ESP in a subsea pipeline where the crude oil is cold. This is because the derivative is greater than 25 cp/°F at temperatures below 45°F.

The effects described here will be reduced when the well begins to produce water. This is due to the lower value of the derivative factor for water. Also, a water external, low-viscosity emulsion will be formed at high water cuts. The pump behaviour will then become similar to the pumping pure water.

Table 1. Results for the head ratio when producing 10° API crude oil at varying temperatures.

Temperature at reservoir conditions (°F)	Viscosity at reservoir conditions (cp)	Viscosity at pump discharge (cp)	Head ratio	Derivative (cp °F^{-1})
80	3474	2126	0.25	−245
85	2481	1557	0.522	−159
90	1826	1211	0.677	−100
100	1063	772	0.818	−51.3
110	679	522	0.882	−28
120	457	369	0.921	−16
130	328	274	0.939	−10
150	189	164	0.99	−4.6

Table 2. Results for the head ratio when the crude oil's API gravity was varied at a fixed temperature.

Crude oil density (°API)	Viscosity at reservoir conditions (cp)	Viscosity at pump discharge (cp)	Head ratio	Derivative (cp °F^{-1})
10	2121	1370	0.61	−129.8
11	1537	1039	0.728	−88.2
12	1150	794	0.79	−64.0
13	867	613	0.827	−46.7
14	661	472	0.864	−34.4
15	502	373	0.893	−24.4
16	392	297	0.907	−18.5
18	246	192	0.932	−10.8
20	154	127	0.95	−6.3

The *Stella* ESP/Well Model

Calibration

The model was tested against production data from the extra heavy oil well MFB-159 made available to us by Torres (1997). It well's basic data was: (i) casing, 7 in; (ii) tubing, 3.5 in; (iii) depth, 3085–3192 ft; (iv) bottom hole pressure, 1160 psi; (v) well head pressure, 150 psi; (vi) productivity index, 3.88 bbl/day psi; (vii) oil, 9.3°API; (viii) bottom hole temperature, 137°F; (ix) crude oil viscosity, 2500 cp at 137°F.

The well had been steam stimulated followed by installation of an ESP (320 hp motor and 107 pump stages). The *Stella* simulation showed the instability observed in the field when the well was placed on production. This was noted in both the well production rate and the pump power requirements (Figure 10). Once steady state was achieved the field and *Stella* simulation results were comparable: Field results, 550 bopd @ 70 hp consumption; *Stella* simulation, 450 bopd @ 60 hp consumption.

However, the original MFB-159 design made with a commercial software program of similar functionality to *Pumpcalc*, called for a power requirement of 225 hp (320 hp motor installed in the well). The increased accuracy stemming from use of the complete model including the enthalpy balance is clear.

The full details of the simulation models and the field case history are available in a thesis by Kirvelis (1997).

Calculation of oil production

The impact of including the full pump description on the overall performance of the well was measured modelling two situations. Firstly, the part of the model that described the heating effect of the ESP pump was switched on. In the second experiment, the heating effect of the pump system was neglected, by switching off the appropriate algorithms.

The following ESP well model was studied: (i) middle perforations, 1600 m; (ii) tubing depth, 1500 m; (iii) casing diameter, 0.229 m; (iv) tubing diameter, 0.089 m; (v) well productivity index, $1.0 \times 10^{-9}\,m^3\,s^{-1}Pa^{-1}$; (vi) crude oil properties, 10°API; (vii) reservoir pressure, 10.34 MPa; (viii) reservoir temperature, 327 K.

The *Stella* model calculated the oil rate ($m^3\,s^{-1}$), friction losses in tubing (Pa), brake horsepower (kW), well bore temperature (K) and the pump efficiency (fraction). The solid curves in Figure 10 (RateE and FricE) represent, respectively, the oil rate and friction losses in tubing with the heating effect of the ESP pump switched on while the dotted curves (RateN and FricN) represent, respectively, the oil rate and friction losses in tubing with the heating effect of the ESP pump switched off.

Impact of the results

Inclusion of the heating effect of the ESP pump gave a 20% increase in the oil rate (from 0.93 to $1.11\,s^{-1}$) and the tubing friction losses were correspondingly reduced

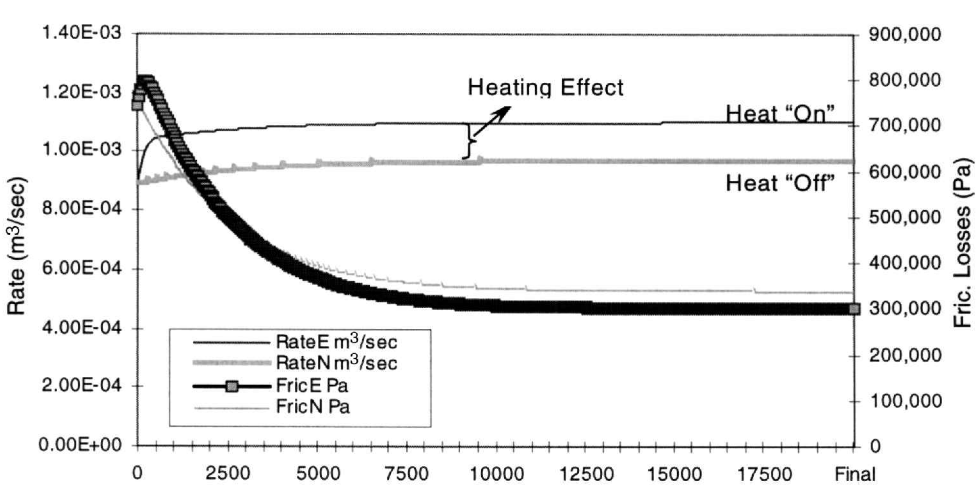

Figure 10. Oil rate and friction losses in tubing as a function of time.

from 350 to 300 kPa. Figure 10 also illustrates how the inclusion of the heating effect of the ESP pump gives an extended period of transient behaviour for the frictional pressure losses before it reached a steady state. The time scale required for this to occur is still underestimated due to the nature of the assumptions made when building the model.

FURTHER DEVELOPMENTS

The *Stella* ESP/well model can be extended to make it more realistic. It could include items such as the effect of highly compressible (gasy) fluids, a better heat transfer model describing phenomena such as fluid convection in the well annulus and cable heating effect, modelling of the thermal mass of the system etc. Matching the results to further real oil well production data would allow tuning of the correlations used and improvement of the modelling philosophy.

CONCLUSIONS

(1) A dynamic, non-linear the ESP/well model has been created. This model included the *PumpCalc* model that analysed the ESP behaviour in detail.
(2) Comparison of the pump model against available commercial software gave good agreement.
(3) Comparison of the Dynamic ESP/well model results with field data showed that the model could predict the unstable production behaviour observed in the field and that there was good numerical agreement during steady-state production.
(4) The heat dissipated by a down hole motor and pump was shown to have a significant impact on the overall production performance of wells producing extra heavy and heavy crude oil using ESP pressure boosting.
(5) The magnitude of the change in viscosity of the crude oil with respect to change in temperature was identified as being the key variable in determining whether the inclusion of the heat effects discussed here are essential when predicting pump performance.
(6) It was shown that the inclusion of this enthalpy balance routine is also required when producing lighter crude oils (e.g. North Sea crude with a 21°API gravity) if they are being pumped under unusual circumstances, e.g. pressure boosting with an ESP in a subsea pipeline where the crude is cold.

NOMENCLATURE

API	API crude gravity
F	pump speed (frequency)
H	head
Q	flow rate
P	pressure
T	temperature
Z	potential head
Eff	efficiency
g	acceleration due to gravity

Greek symbols

κ	kinematic viscosity
γ	fluid density
v	velocity
μ	absolute viscosity

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APPENDIX 1

Modified Egbogah–Jacks correlation

$$\mu_o = 10^{10^{[2.06492-0.0179*^{o}\mathrm{API}-0.70226*\log(T)]}} - 1$$

Modified Kartoatmodjo correlation

$$\mu_{ob} = 0.0132 + 0.9821 * F - 0.005215 * F^2$$
$$F = \left(0.2038 + 0.8591 * 10^{(-0.000845*R_s)}\right) * \mu_{od}^{\left(0.3855+0.5664*10^{(0.00081*R_s)}\right)}$$

Modified Labedi Correlation

$$\mu_o = \mu_{ob} - \left(1 - \frac{P}{P_b}\right)\left(\frac{10^{-3.8085}\mu_{od}^{1.4131}P_b^{0.6957}}{10^{0.00288^{o}\mathrm{API}}}\right)$$

Standing correlation

$$P_b = 18.2\left[\left(\frac{R_s}{S_g}\right)^{0.83} * 10^a - 1.4\right]$$

where $a = 0.00019T - 0.0125*(°\mathrm{API})$.

ADDRESS

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