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L. N. Marino
Niagra Mohawk Power Corp.

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400 Stanwix Street
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FOREWORD

The ISA Power Industry Division's primary purpose is to serve all professional personnel whose interests are in instrumentation, control and automation of power generation and distribution. The Division's Annual Power Instrumentation Symposium provides a forum where utility, consultant and supplier personnel can mutually dialogue on technical successes and failures, and new approaches and solutions which will result in advancing the state of the art of the power industry.

The theme of this 18th Annual Power Instrumentation Symposium is "Power Instrumentation - Challenge and Change." In a time of economic uncertainty, fuel shortages, and with concern over control of our environment, the Power Industry faces a multitude of challenges. Two definite changes or trends within the industry are the moves to a greater dependence on fossil coal fired and nuclear generating units. Hence, the technical papers programmed and presented at this meeting focus primarily on these two technologies. It further seems appropriate that this Symposium is being held in Houston where many industry representatives from the Southwest are initially involved with coal and nuclear plants.

The opening session dealing with viscosity measurement and control was programmed specifically in response to an interest survey of past symposium attendees. The remainder of the sessions deal with coal and/or nuclear applications. Control and safety system design for coal fired units can only be understood if one has knowledge of the process, hence Part I of Session II lays the foundation for Part II and Session IV. Session III dealing with Nuclear standards will provide design insight for future applications. Session V covers actual plant operating experiences and the knowledge gained in these plants may very well serve as an aid in solving problems in new plants.

It is a credit to the program chairman, session developers, and authors that they have been able to assemble such a meaningful technical program. While many problems remain unsolved, technical dialogue and engineering skills, similar to those demonstrated in these proceedings, will aid in their resolution.

By making this program available to you, we hope it encourages each of you to lend your talents to the Power Division. Together we can strive to serve the industry in a coordinated professional way, rising to the challenges and changes to meet the needs of tomorrow.

Paul L. Kenny
General Chairman

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EVALUATION OF PROCESS VISCOMETERS FOR FUEL OIL VISCOSITY MEASUREMENT AND CONTROL

Mr. T.P. Davis
Mechanical Engineer
Power Plant Engineering
Tampa Electric Company

Mr. E.M. Good
Chief Instrumentation & Control Engineer
Generation Engineering
Florida Power Corporation

Dr. C.A. Smith
Assistant Professor
College of Engineering
University of South Florida

ABSTRACT

Advantages of viscosity measurement and control as applied to power plant fuel oil supply systems are discussed, followed by background material including the definition of viscosity and the associated units of measurement. A typical viscosity control system application and the associated design requirements in a large oil-fired power plant are described, and finally, a review is performed of some of the most common types of process viscometers currently available on the market.

ADVANTAGES OF VISCOSITY MEASUREMENT AND CONTROL

Because of the ever-increasing costs of residual fuel for oil-fired power plants, the wide variations in the characteristics of crude stocks available to the refineries, and the more stringent pollution requirements imposed upon the power industry in recent years, it has become necessary to further refine the methods of combustion control. A means of establishing improved combustion is direct measurement and control of the fuel oil supply viscosity. Although not entirely new to the power industry, this method has only recently received adequate attention in this country.

The basic objective of combustion control is to establish and maintain the optimum air-fuel ratio for combustion at any given set of conditions in the boiler. This is accomplished by a complex control scheme which compares varying conditions of steam and feedwater temperatures, pressures, flows, air and fuel flow rates to control the amount of combustion air mixed with the atomized fuel. Fuel oil is atomized at the burners prior to ignition in the furnace and it is in this atomization process that viscosity control plays its role.

For each burner type there is an optimum viscosity range in which the fuel oil will be atomized to as near ideal combustion conditions as possible. The purpose of viscosity control then is to maintain this proper range of oil viscosity at the burners thereby insuring proper atomization and uniform flow of the fuel. With the fuel viscosity held constant, it is far easier to maintain tight combustion control. With tighter combustion control, a fuel savings will be realized as well as a reduction in pollution levels. Other advantages of viscosity control include maintaining close to "design" conditions for the burner pumps and prevention of premature fouling of fuel oil heaters due to overheating the fuel. In addition, some oil flow metering systems require constant viscosity for accuracy.

In the past, oil shipments were regular and consistent, and the viscosity-temperature curves derived from batch-testing fuel oil shipments were reliable. The optimum viscosity for combustion was determined largely by trial and error, through temperature variation of the heated residual oil. Once determined, the optimum temperature of the fuel oil served as the setpoint in a temperature control loop. By maintaining the optimum temperature for combustion, the viscosity was held within allowable limits.

However, with the variety of oil grades available today, especially with the low-sulphur oils, it is difficult to maintain viscosity within a narrow allowable range using a fixed temperature setpoint. This is especially true with multi-viscosity mixtures, where the oil characteristics are very difficult to predict. Another characteristic of residual fuel oils which necessitates the use of viscosity control is that viscosity is a sensitive function of temperature. A small change in temperature or oil consistency will cause a large variation in oil viscosity at the burners. See Figure 1 for typical viscosity-temperature curves for fuel oils.

Therefore, to maintain constant fuel oil supply conditions, viscosity, not temperature, is the parameter to measure and control. The added instrumentation and hardware is easily justified in view of the higher fuel prices of today and the fuel consumption of larger oil-fired plants. Before considering the viscosity control system description and hardware evaluation, background material including the basic definition of viscosity and the associated units of measurement shall be briefly defined.

BACKGROUND

Viscosity is defined as "that property of a fluid by virtue of which it offers resistance to shear." For a given rate of angular deformation of a Newtonian or "ideal" fluid, the shear stress is proportional to the viscosity or "fluidity" of the fluid. This proportionality varies for non-Newtonian fluids. Viscosity is independent of pressure for most practical purposes and is considered generally, a function of temperature only. The viscosity of gases increases with temperature whereas for liquids, including fuel oils, the viscosity decreases with increasing temperature. This is due to the fact that cohesive forces which decrease with increasing temperature are much higher in liquids and contribute much more than intermolecular forces to the overall viscosity. The units of viscosity and general form of the viscous force equation may be obtained by considering the case of a Newtonian fluid between two parallel planes of equal area shown in Figure 2.

The force F required to move the upper plate of area A at velocity V at a distance D above the bottom plate is given by the following equation:

$$F = \eta\left(\frac{AV}{D}\right) \qquad (1)$$

where η is referred to as the "dynamic" or "absolute" viscosity. For a given force of 1 dyne, area of $1cm^2$, velocity is 1 cm/sec., and distance of 1 cm, the constant η or dynamic viscosity is defined to be 1 Poise, or 100 Centipoise. Another viscosity constant γ , "kinematic" viscosity, is related to dynamic viscosity by the following equation:

$$\gamma = \eta/\rho \qquad (2)$$

where η is the dynamic viscosity in Poise, ρ is the density of the fluid in gm/cm^3, and γ is defined in units of Stokes, or 100 Centistokes.

A definition of viscosity for a fluid flowing in an enclosed tube in more practicable terms is given by the Poiseuille equation:

$$\eta = \frac{\pi \Delta P R^4}{8 Q L} \qquad (3)$$

where viscosity is described as the ratio of shearing stress to rate of shear as previously defined, and ΔP is the pressure drop in $dynes/cm^2$, radius R in cm, volumetric flow rate Q in cm^3/sec, length L in cm, and dynamic viscosity η in Poise. Note that under constant, non-pulsating, laminar flow conditions that viscosity is directly proportional to the pressure drop in a given length L of an enclosed tube.

Viscosity may be defined in terms of other relative viscosity constants such as Seconds Redwood, Seconds Saybolt-Furol, Seconds Saybolt Universal and Grades Engler. In the United States, Seconds Saybolt Universal (SSU) is a commonly used viscosity constant for fuel oil viscosity measurement. The relative viscosities Saybolt Universal, Redwood and Engler may be determined from the kinematic viscosity γ using the following empirical equation:

$$\gamma = At - B/t \qquad (4)$$

The scale time is given as t in seconds, and the constants A and B are given below:

t	A	B
Seconds Saybolt Universal	0.0022	1.80
Seconds Redwood	0.0026	1.72
Second Engler	0.00147	3.74

A TYPICAL VISCOSITY CONTROL APPLICATION

In most residual oil-fired power plants, the optimum viscosity for combustion usually falls within the range of 90-150 SSU. Since residual fuel oils are usually highly viscous, almost solidified, at ambient temperature, the oil must be heated in fuel oil heaters to obtain the desired viscosity prior to reaching the burners.

In a viscosity control system the amount of heating in the fuel oil heaters is controlled, and the feedback signal from a process viscometer is compared with the setpoint signal to obtain a viscosity error signal. There are various control loop configurations ranging from the elementary single loop system to rather complex cascade configurations to reduce the thermal and oil flow deadtimes. Regardless of the control strategy, there are general design requirements that the process viscometer must meet.

First of all, the process viscometer must

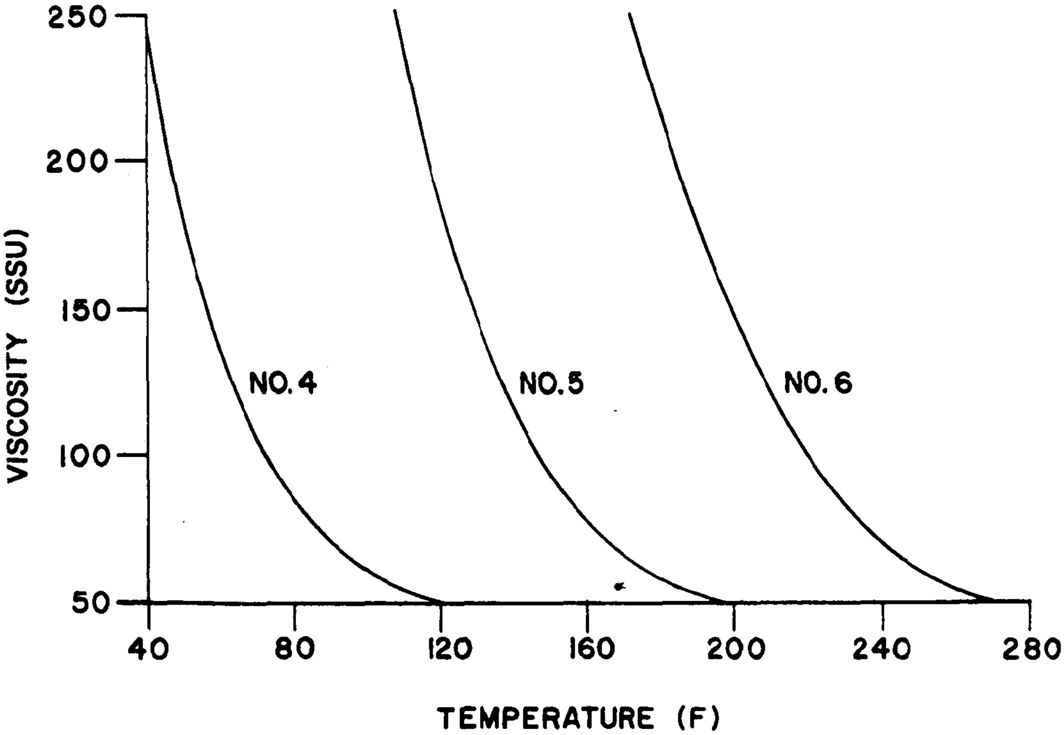

FIGURE 1 VISCOSITY/ TEMPERATURE TYPICAL FUEL OILS

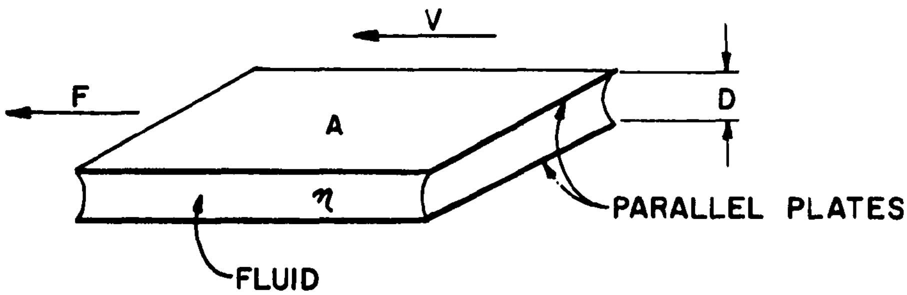

FIGURE 2 DERIVATION VISCOUS FORCE EQUATION

be reliable, low in maintenance, and be of rugged construction. In a large number of modern plants, a viscometer must be able to withstand pressure and temperature extremes in the neighborhood of 1200 psig and 350°F (84.36 Kg/cm^2, 176.7°C). The viscometer must be installed in a bypass line on the main fuel oil supply piping so that the instrument may be valved off for routine maintenance with the plant on line. Since high pressures and temperatures are involved, process connections should be welded, with no flanges in the process stream. The instrument housing should be weather-tight and of explosion-proof construction, if crude oil is to be burned. A continuous analog output signal is desired, preferably a current signal (4-20 mA, for example) that is compatible with modern electronic analog controls. Fast response is desired of the viscometer, and for this reason, a relatively high flow rate through the bypass line is required. A typical 515 MW oil-fired power plant will consume at full load an oil flow of approximately 600 GPM (37.85 l/sec.). It is desired that a flow of at least 50 GPM (3.15 l/sec.) be present in the viscometer bypass line at full load. Repeatability and accuracy over the entire span of approximately 0-200 SSU is, of course, a desirable factor. Let us now consider, in view of the design requirements, some of the various types of process viscometers available on the market.

VISCOMETER SURVEY

In this survey, five major types of viscometers popular in the process industry were chosen for investigation:

1. Falling Piston Type
2. Vibrating Probe Type
3. Rotational Type
4. Float Type
5. Capillary Type

Each of these types will now be considered in detail.

Falling Piston Type Viscometer

The measuring element consists of a piston, ring or slug that is periodically raised, either pneumatically or mechanically and is then allowed to fall under its own weight through the medium to be measured. The time of fall is recorded and transmitted as a viscosity signal. For the viscosity range required, full scale accuracy over full span is approximately 1%, repeatability is about 0.25% of full span and a sensitivity of 0.1% of full scale is realizable. Cycle times of once every 30 seconds may be obtained, but generally, cycle times are of the order of 2 minutes.

Vibrating Probe Type Viscometer

The principle of the measuring element is based upon the dampening effect of the measured medium as it passes over a vibrating probe. The measuring element consists of a frequency generator, vibrating spring rod, probe, and a magnetic pickup unit. A drive coil causes the probe to vibrate at 120 Hz and the amplitude of these vibrations is then measured with a pickup coil arrangement. The output signal is a continuous function of viscosity. Accuracy and repeatability are approximately 1% of full span.

Rotational Type Viscometer

This type viscometer utilizes a motor-spring-spindle arrangement by which viscosity is measured by the torque required to rotate a spindle in the measured medium. The angular displacement of the spring may be measured by a variable capacitance or by a potentiometer, and is then converted to a process signal proportional to viscosity. Accuracy is within 0.3% of full scale. Response time is less than 30 seconds. The standard spindle speed is 50 rpm and range changes are easily made by changing spindle size.

Float Type Viscometer

This type viscometer employs a variable area flow meter with the flow rate held constant. The position of the float, then, is a function of absolute viscosity and density or, in other words, kinematic viscosity.

Capillary Type Viscometer

The capillary type viscometer utilizes the Poiseuille law given previously in which the flow rate (Q) is a constant flow from a gear metering pump. The medium to be measured is pumped through a capillary tube under laminar flow conditions and the differential pressure developed across a given length of capillary defines the viscosity by the Poiseuille relationship. This differential pressure signal is then converted to an electronic signal using a differential pressure transmitter. Pressure and temperature ratings are acceptable and the instrument is not affected by flow or pressure transients since the housing of the instrument is simply a reservoir for the synchronous speed constant flow gear pump. This instrument has shown high reliability and is accurate and repeatable to within 1% of full span. The viscometer is not easily affected by contaminants since the unit is self-flushing and the gear pump will not permit particles of significant size to enter the capillary tube. Butt weld connections and a high pressure body are available. Only yearly maintenance is necessary and the unit is easily serviceable.

VISCOMETER SELECTION

There are several different types of viscometers on the market today. The ones mentioned in this paper do not represent the complete selection available, but only serve as examples of the many design types utilized to measure viscosity.

Each type has an application for which it is best suited. Since there are only a few types of viscometers which are capable of handling a particular process application, careful evaluation is required in viscometer selection.

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DESIGN AND COMPUTER SIMULATION OF A VISCOSITY-CONTROLLED FUEL OIL HEATING SYSTEM

Mr. T. P. Davis
Mechanical Engineer
Power Plant Engineering
Tampa Electric Company

Mr. E. M. Good
Chief Instrumentation
& Control Engineer
Generation Engineering
Florida Power Corporation

Dr. C. A. Smith
Assistant Professor
College of Engineering
University of South Florida

ABSTRACT

A typical residual fuel oil heating system and the design requirements of a power plant viscosity control loop are outlined. The selection and installation design of a process viscometer are also described. Finally, a computer simulation is derived and used to predict the dynamics and control loop performance of the actual viscosity-controlled fuel oil heating system.

SYSTEM DESCRIPTION

In residual oil-fired power plants, the optimum fuel oil viscosity for combustion usually falls within the range of 90-150 SSU. Since heavy fuel oils are highly viscous, almost solidified at ambient temperature, the oil must be heated in fuel oil heaters to obtain the desired viscosity prior to reaching the burners. Fuel oil viscosity is measured and used to control the amount of heat transferred to the oil.

The major elements in a viscosity-controlled fuel oil heating system consist of the fuel oil piping, fuel oil heater, viscometer and necessary control hardware. The system components and design covered in this paper are based upon a typical modern oil-fired power plant of approximately 515 MW capacity.

For this particular application, an assembly consisting of 80 hairpin concentric-pipe heat exchanger elements stacked vertically 4 units high and 20 units wide is utilized for fuel oil heating. The heat exchanger elements are connected in 20 parallel passes, each pass consisting of 4 heat exchanger elements. Each of the 20 parallel passes are connected together by oil and steam manifolds with shutoff valves for each parallel pass. The fuel oil heater is connected counterflow, with oil entering at the bottom and flowing upward through each parallel pass. Saturated steam entering at the top of the fuel oil heater assembly finally exits as subcooled condensate. Oil flows in the shell side over aluminum fins and is heated by the steam/condensate mixture flowing in the inner pipe. In a system of this size under full load conditions, oil flowing at approximately 600 GPM (37.85 l/sec.) is heated in the fuel oil heater from approximately 130°F (54.4°C) to as high as 280°F (137.8°C). Since the heat transfer and oil flow are relatively high in this large installation, steam flow to the heater cannot be controlled due to the risk of thermal shock to the heater under transient conditions. Instead, the rate of heat transfer is varied by controlling the condensate level in the fuel oil heater by means of a condensate drain control valve. This method of control is much slower than controlling steam flow, but eliminates the concern of thermal shock.

After leaving the fuel oil heater, the heated oil is then raised to high pressure by the fuel oil burner pumps before reaching the main fuel oil supply header at the boiler. It is at this point that the fuel oil viscosity is measured and converted into a continuous electrical signal, compatible with electronic analog controls. In a single-loop control system, this viscosity signal is used to develop an error signal based upon the desired viscosity. The error signal is then used to control the position of the fuel oil heater condensate drain valve. See figure 1. An increase in the viscosity signal above setpoint indicates that the fuel oil temperature is too low and the drain valve is opened further to increase heat transfer surface in the heater. The opposite response occurs for a low viscosity signal.

The detailed design of a viscosity-controlled fuel oil heating system will now be covered.

VISCOMETER SELECTION AND INSTALLATION

A power plant viscosity control installation requires that the process viscometer must be of rugged construction, reliable and require infrequent calibration and maintenance. Pressure and temperature extremes of 1200 psig (84.4 Kg/cm^2) and 350°F (176.7°C) are possible in this application. Welded connections are required, and the viscometer must be installed in a bypass line to allow control of the oil flow through the instrument. The flow through this bypass line should be at least 50 GPM (3.15 l/sec.) of the total 600 GPM (37.85 l/sec.) oil flow at full load to allow fast viscometer response. To facilitate maintenance, the bypass line should be supplied with shutoff valves and a drain valve. A continuous analog output current signal of 4-20 mA is desired in this particular application for use with electronic analog controls. The setpoint station and the electronic control hardware are located in the control room, quite a distance from the viscometer installation, thereby necessitating the use of an electrical process signal.

For this application, the capillary type viscometer was chosen since it has shown high reliability and meets the design requirements of this power plant installation. The instrument consists of a cast steel oil reservoir through which the bypass oil flows, a gear pump drawing suction from this reservoir and powered by a synchronous speed motor, and a capillary tube through which the oil is pumped in a constant, non-pulsating, laminar flow. The differential pressure developed between the reservoir pressure and the pressure required to maintain flow through the capillary tube is a proportional function of relative viscosity in SSU within 2% linearity over the measuring range of 0-230 SSU. The differential pressure signal is then converted to an electrical signal (4-20 mA) by a differential pressure transmitter located at the viscometer installation. Viscometer oil flow requirements are at least 5 GPM (0.32 l/sec.) minimum flow at low loads but no more than 100 GPM (6.31 l/sec.) maximum flow at full load.

A location was chosen for the bypass line installation as close to the burner headers as possible and inside the plant in an easily accessible area protecting the installation from weather. A maximum design flow of 50 GPM (3.15 l/sec.) was desired in the bypass line at full load conditions of 600 GPM (37.85 l/sec.) in the main fuel oil supply line. A minimum design flow of 5 GPM (0.32 l/sec.) was desired in the bypass line at low load conditions of 120 GPM (7.6 l/sec.) in the main fuel oil supply line.

With this design information and the physical properties of the fuel oil determined, the main fuel oil supply line orifice was then sized to provide the design flow conditions through the viscometer. The pressure drop in the main fuel oil line between the bypass line connections at full flow conditions was calculated. Likewise, the pressure drop across the bypass line was also determined, taking into account all piping, fittings and viscometer pressure drop at 50 GPM (3.15 l/sec.) maximum flow. The difference in pressure drop between the bypass line and the main fuel oil supply line determined the pressure drop required of the main fuel oil supply line orifice at full load conditions. The orifice was sized using a standard orifice sizing procedure.[1] Similar calculations were performed over the entire operating range of the installation, and the bypass flow characteristics were determined as shown in figure 2. Calculations with the bypass line valved off determined that the pressure drop across the main fuel oil supply orifice was not excessive and would not interfere with plant operation. As can be seen from the bypass flow curve, the flow through the viscometer is nearly linear with the main fuel oil supply flow and meets the design flow conditions established earlier. The bypass design was such that the installation could be field fabricated with minimal expense.

The bypass line and viscometer was electrically heat traced and insulated. Only two cable runs from the boiler controls cabinet to the viscometer installation were required for viscometer power and differential pressure transmitter connections. A small shutoff switch and thermal circuit breaker were installed at the viscometer to facilitate maintenance of the installation.

DERIVATION OF CONTROL LOOP MODEL EQUATIONS

The computer model used for evaluation of the viscosity control loop design consists of smaller mathematical models combined into a single digital computer program. Each of the mathematical models will now be considered.

Fuel Oil Viscosity-Temperature Model

From viscosity-temperature plots for typical fuel oil grades data was compiled and used to develop curve fit equations that were used in the digital computer program. A trial solution to the general curve shape was of the form:

$$SSU = a(T) + be^{c(T)} + d \tag{1}$$

where a, b, c and d are constants to be determined. T is the oil temperature in degrees Farenheit and SSU in the fuel oil viscosity in Saybolt Seconds Universal. By trial and error solution of the curve fit data, the following viscosity-temperature models were developed:

For No. 4 fuel oil:

$$(29 \le T \le 96) \quad SSU = 727 - 9(T+71) + 159 e^{.01(T+71)} \qquad (2)$$

$$(96 \le T \le 129) \quad SSU = 69 - 0.5536(T-96) \qquad (3)$$

For No. 5 fuel oil:

$$(100 \le T \le 167) \quad SSU = 727 - 9(T) + 159 e^{.01(T)} \qquad (4)$$

$$(167 \le T \le 200) \quad SSU = 69 - 0.5536(T-167) \qquad (5)$$

For No. 6 fuel oil:

$$(168 \le T \le 235) \quad SSU = 727 - 9(T-68) + 159 e^{.01(T-68)} \qquad (6)$$

$$(235 \le T \le 268) \quad SSU = 69 - 0.5536(T-235) \qquad (7)$$

These curve fit equations are reasonably accurate within the range normally encountered in power plant applications.

Fuel Oil Heating System Model

As mentioned previously, the fuel oil heater installation consists of 20 parallel passes of 4 concentric-pipe hairpin heat exchanger elements in series. The outer shell of each element consists of 3½ inch diameter schedule 40 steel pipe and the inner pipe side consists of 1½ inch diameter schedule 40 steel pipe. Bonded to the inner pipe are 16 aluminum fins, 3/4 inch (1.905 cm) in height, longitudinal and continuous over the entire length of the heat exchanger. The hairpin elements are approximately 22 feet (6.71 m) long, providing 176 total feet (53.64 m) of pipe per parallel path of 4 elements in series. At rated design conditions, No. 6 oil, at a flow rate of 600 GPM (37.85 l/sec.) flows through the shell side and is heated from 120°F (51.7°C) to approximately 240°F (115.5°C). Saturated steam entering at 366°F (185°C), 150 psig (10.5 Kg/cm^2), 22,000 lb/hr. (9979 Kg/hr.), is condensed and leaves the heater at 210°F (99°C). Total design heat transfer is 19,000,000 BTU/hr.

Since the fuel oil heating system model involved two-phase heat transfer, variable surface area, counterflow conditions, along with varying oil flow and temperature demands, an accurate dynamic model of the system would be impractical without simplifying assumptions in the development of model equations. In the derivation of the computer model for this system, care was taken to choose assumed conditions that would not adversely affect the validity of the model. Below are listed these major assumptions:

1. Divide the fuel oil heater into a steam section model and a condensate section model.
2. Assume the fuel oil heater to be 20 continuous, straight, concentric-pipe heat exchangers, the same length as four hairpin heat exchangers in series.
3. Assume that all condensate is formed in the steam section model. All condensate formed flows immediately to the condensate model, and heat transfer in the steam section is by flowing saturated steam.
4. Assume a distinct division line between the two models defined by the condensate level in the heater which is subject to constant change. The lengths of the respective models vary with condensate level, a function of the amount of steam condensed and the condensate drain valve position. The sum of the lengths of the two models is always the length of the heater.
5. The heater and piping is well insulated and heat traced. Assume no heat loss to the surroundings.
6. No fin thickness data was available for the heater modeled. Assume 18 BWG (0.049 inch thickness) aluminum fins bonded to the outside of the inner pipe with no bond resistance.
7. The following heat transfer film coefficients (BTU/hr. ft.^{2} °F) were calculated using typical values(2) for this type of heat exchanger:

a.	Inside steam heat transfer	1500
b.	Inside condensate heat transfer	250
c.	Outer oil heat transfer	17.9
d.	Outer oil heat transfer referred to inside	22.9
e.	Overall heat transfer	22.6

8. Assume heat transfer film coefficients and physical properties to be constant in both the steam and condensate section models.
9. Since the fuel oil heater is counterflow, the calculations follow the oil as it passes through the condensate section model and then into the steam section model. All calculations are based on equal increments of time. Each incremental heater length corresponding to a time increment varies in length depending upon the oil flow velocity. All calculations are time dependent as the oil passes through the heater.

With these assumptions, the following model equations were developed:

Inner Pipe Wall Heat Balance

A heat balance was performed for an incremental length section of the inner pipe wall. The heat added to the wall by the steam or condensate minus the heat transferred to the oil flowing over the wall and fins is equal to the heat retained within the wall:

$$h_i A_i (T_s - T_w) - h_f(16A_f + A_o)(T_w - T) = (A_{pw}) c_{pw} \frac{dT_w}{dt} \quad (8)$$

where:
- hi = inside heat transfer film coefficient (BTU/hr ft^2 °F)
- Ai = internal surface area of inner pipe per unit length (ft)
- Ts = steam temperature (°F)
- Tw = pipe wall temperature (°F)
- h_f = outside heat transfer film coefficient (BTU/hr ft^2 °F)
- $16A_f$ = surface area of 16 aluminum fins per unit length (ft)
- Ao = bare outer wall surface of inner pipe per unit length (ft)
- T = oil temperature over incremental length (°F)
- Ap_w = [Tube Cross-sectional Area][Density of Steel] + [Fin Cross-sectional Total Area][Density of Aluminum]
- Cpw = weighted average of Cp for steel and Cp for aluminum based on relative volumes per unit length (BTU/lb°F)
- t = time (hours)

Evaluating and rearranging the variables in the heat balance produced the pipe wall heat balance model equations for the steam and condensate sections:

$$\frac{dT_w}{dt} = \left(\frac{h_i A_i}{A_{pw} c_{pw}}\right)(T_s - T_w) - \left(\frac{h_f(16A_f + A_o)}{A_{pw} c_{pw}}\right)(T_w - T) \quad (9)$$

Fuel Oil Heat Balance

The following equations were used to describe heat transfer[2] from the aluminum fins and pipe surface to the flowing oil for a single parallel pass:

$$Q_F = (bPN_f \Omega + A_B)(h_f)(T_w - T) \quad (10)$$

$$\Omega = \frac{\tanh(mb)}{mb} \qquad m = \left(\frac{h_f P}{K_A a_x}\right)^{0.5} \quad (11)$$

where:
- Q_F = heat transfer from the fins to the oil (BTU/hr)
- b = fin height (ft)
- P = longitudinal perimeter of incremental fin ≈ 2ΔL
- N_f = number of fins = 16
- M = fin heat transfer parameter
- Ω = fin efficiency (dimensionless fraction)
- A_B = total bare surface
- K_A = thermal conductivity of aluminum (BTU/hr ft °F)
- a_x = longitudinal cross-sectional area of a single fin (ft^2)

Rearranging equation(10) yielded the following equation for heat transfer from the aluminum fins to the fuel oil, where ΔL is an incremental section length in feet:

$$Q_F = (2bN_f \Omega + A_o)(h_f)(T_w - T)(\Delta L) \quad (12)$$

The heat transferred from the fins to the flowing oil and the heat transferred by the oil flowing into the increment minus the heat transferred by the oil flowing out of the incremental section is equal to the heat retained in the incremental length of oil:

$$\begin{bmatrix}\text{Heat Flow}\\ \text{From Fins}\\ \text{To Oil}\end{bmatrix} + \begin{bmatrix}\text{Heat Flow Into}\\ \text{Incremental}\\ \text{Length}\end{bmatrix} - \begin{bmatrix}\text{Heat Flow Out of}\\ \text{Incremental}\\ \text{Length}\end{bmatrix} = \begin{bmatrix}\text{Heat Retained In}\\ \text{Incremental Length}\\ \text{Of Oil}\end{bmatrix}$$

$$Q_F + w_o c_{po}(T - T_r)\big|_{L+\Delta L} - w_o c_{po}(T - T_r)\big|_L = A \Delta L \rho_o c_{po} \frac{dT}{dt} \quad (13)$$

where:
- W_o = oil mass flow rate (lb/hr) in each single pass
- Cp_o = specific heat of oil (BTU/lb°F)
- Tr = arbitrary reference temperature (°F)
- A = flow area of oil (ft^2)
- ρ_o = density of oil (lb/ft^3)

Dividing by ΔL yields:

$$Q_F/\Delta L + (c_{po} w_o)\left(\frac{(T - T_r)\big|_{L+\Delta L} - (T - T_r)\big|_L}{\Delta L}\right) = A \rho_o c_{po} \frac{dT}{dt} \quad (14)$$

Taking the limitas (ΔL) approaches zero:

$$\lim_{\Delta L \to 0} (Q_F/\Delta L) + (c_{po} w_o)\left(\frac{\partial T}{\partial L}\right) = A \rho_o c_{po}\left(\frac{\partial T}{\partial t}\right) \quad (15)$$

Likewise, for the fin heat transfer equation:

$$\lim_{\Delta L \to 0} \left(Q_F / \Delta L \right) = (2bN_f \Omega + A_o)(h_f)(T_w - T) \qquad (16)$$

Substituting equation (16) into equation (15):

$$(2bN_f \Omega + A_o)(h_f)(T_w - T) + (C_{po} w_o)\left(\frac{\partial T}{\partial L}\right) = A_{\rho o} C_{po}\left(\frac{\partial T}{\partial t}\right) \qquad (17)$$

Rearranging equation (17):

$$\frac{\partial T}{\partial t} = \left(\frac{2bN_f \Omega + A_o}{A_{\rho o} C_{po}}\right)(h_f)(T_w - T) + \left(\frac{w_o}{A_{\rho o}}\right)\left(\frac{\partial T}{\partial L}\right) \qquad (18)$$

Condensate Production Equation

The condensate production rate in the steam section model was determined by a summation of the incremental condensate calculations over the length (L) of the steam section. The number of increments in the steam section is ($L/\Delta L$). Condensate produced in a given time interval may be described by the following summation:

$$\dot{m}_c = \sum_{i=1}^{N = L/\Delta L} \frac{Q_i}{\lambda} = \sum_{i=1}^{N = L/\Delta L} \left(\frac{h_i A_i \Delta L (T_s - T_w)}{\lambda} \right) (N_p) \qquad (19)$$

where: $\dot{M}_c$ = rate of condensate production (lb/hr)
L = length of steam section model (ft)
Qi = heat transfer from incremental length of steam section to pipe wall (BTU/hr)
λ = latent heat of vaporization of water (BTU/lb)
Np = number of parallel passes

Since the incremental length may be expressed as the product of oil velocity and incremental time period, the above equation may be expressed in terms of time (DT) in hours and oil flow (wo) in pounds per hour:

$$\Delta L = \left(\frac{w_o}{\rho_o A}\right)(DT) \qquad (20)$$

$$\dot{m}_c = \sum_{i=1}^{N = L/\Delta L} \left(\frac{h_i A_i w_o DT (T_s - T_w)}{\rho_o A \lambda} \right) (N_p) \qquad (21)$$

Condensate Mass Balance Equation

For this model derivation, it was assumed that the condensate section reacts as a "tank" of condensate. The level in the "tank" is a function of the condensate production rate in the steam section and the condensate drain valve position. Applying a mass balance to the condensate section:

$$\begin{bmatrix}\text{Condensate}\\ \text{Production}\\ \text{Rate}\end{bmatrix} - \begin{bmatrix}\text{Flow Through}\\ \text{Condensate}\\ \text{Drain Valve}\end{bmatrix} = \begin{bmatrix}\text{Accumulation}\\ \text{Rate In}\\ \text{Heater}\end{bmatrix}$$

$$\dot{m}_c - \dot{m}_o = \rho A_T \frac{dh}{dt} \qquad (22)$$

where: ρ = density of condensate (lb/ft^3)
A_T = cross-sectional area of "tank" (ft^2
$\frac{dh}{dt}$ = rate of level change in "tank" (ft/hr)
h = "tank" level (ft)
$\dot{M}_o$ = mass flow rate of condensate through valve (lb/hr)

The mass flow rate through the condensate drain valve was determined as follows:

$$C_V = Q\sqrt{\frac{SPG}{\Delta P}} \qquad (23)$$

where: Q = 0.002 $\dot{m}_o$ (lb/hr) for condensate

and: C_V = drain valve coefficient of discharge
Q = flow rate through valve (GPM)
SPG = specific gravity of condensate = 1.0
ΔP = total pressure drop across valve (psi)

Rearranging equation (23):

$$\dot{m}_o = 500\, C_V \sqrt{\Delta P} \qquad (24)$$

The steam section length (L) is equal to the heater length L_H minus the "tank" height (h):

$$h = L_H - L \qquad (25)$$

$$\frac{dh}{dt} = -\frac{dL}{dt} \qquad (26)$$

The accumulation rate in the heater may be expressed as follows:

$$\rho A_T \frac{dh}{dt} = -\rho A_T \frac{dL}{dt} \qquad (27)$$

The condensate mass balance equation may now be expressed as follows:

$$\dot{m}_c - \dot{m}_o = -\rho A_T \frac{dL}{dt}$$

The condensate mass balance equation may now be stated in terms of the overall heater model:

$$\dot{m}_c = 500\, C_V \sqrt{\Delta P} - \rho A_T \frac{dL}{dt} \qquad (28)$$

The condensate drain valve coefficient (C_V) was expressed as a function of drain valve position for a typical 1½ inch, equal percentage flow characteristic valve.

Condensate Heat Balance

In the condensate section model, the heat transfer to the pipe wall and the heat transferred into the incremental section minus the heat transferred out of the incremental section is equal to the heat accumulated in the condensate section of incremental length:

$$\begin{bmatrix}\text{Heat Transfer}\\ \text{To Pipe Wall}\end{bmatrix} + \begin{bmatrix}\text{Heat Transfer}\\ \text{Into Increment}\end{bmatrix} - \begin{bmatrix}\text{Heat Transfer}\\ \text{Out Of}\\ \text{Increment}\end{bmatrix} = \begin{bmatrix}\text{Heat Accumulated}\\ \text{In Condensate}\end{bmatrix}$$

$$h_i A_i (T_c - T_w) + \dot{m}_o c_p (T_c - T_R)\big|_{L+\Delta L} - \dot{m}_o c_p (T_c - T_R)\big|_{L} = A_{COND}\, \Delta L\, \rho\, c_p \frac{dT_c}{dt} \qquad (29)$$

where: T_c = condensate temperature (°F)
C_p = specific heat of condensate (BTU/lb)
A_{COND} = cross-sectional flow area of condensate (ft^2)
ρ = density of condensate (lb/ft^3)

Rearranging and taking the limit as (ΔL) approaches zero:

$$h_i A_i (T_c - T_w) + \dot{m}_o c_p \left(\frac{\partial T_c}{\partial L}\right) = A_{COND}\, \rho\, c_p \left(\frac{\partial T_c}{\partial t}\right) \qquad (30)$$

$$\frac{\partial T_c}{\partial t} = \left(\frac{h_i A_i}{A_{COND}\, \rho\, c_p}\right)(T_c - T_w) + \left(\frac{\dot{m}_o}{A_{COND}\, \rho}\right)\left(\frac{\partial T_c}{\partial L}\right) \qquad (31)$$

Fuel Oil Piping Time Lag

Since the viscometer in the fuel oil heating system installation is located quite a distance from the fuel oil heater outlet, the piping deadtime was modeled into the control loop as a linear function of oil velocity:

$$\text{TIME LAG (SEC.)} = \frac{(\text{PIPING LENGTH (ft)})}{\begin{pmatrix}\text{OIL MASS}\\ \text{FLOW RATE}\\ (\text{lb/sec.})\end{pmatrix} \begin{pmatrix}\text{OIL}\\ \text{DENSITY}\\ (\text{lb/ft}^3)\end{pmatrix}^{-1} \begin{pmatrix}\text{PIPE}\\ \text{FLOW AREA}\\ (\text{ft}^2)\end{pmatrix}^{-1}} \qquad (32)$$

Controller Model

In the single loop control system, the viscometer process signal was compared with a setpoint signal to obtain an error signal. This error signal was then converted into a signal in units of SSU which was then used in the controller model to determine the change required in condensate drain valve position. The following three-mode controller equation was utilized in the computer model:

$$m(t) = K_c E(t) + \frac{K_c}{T_i}\int E(t)\,dt + K_c T_d \frac{dE(t)}{dt} \qquad (33)$$

where: $M(t)$ = controller output signal expressed in % valve position as a function of time
Kc = controller gain (% VP/SSU)
$1/Ti$ = reset rate (repeats/minute)
Td = rate (minutes)
$E(t)$ = error signal expressed in SSU error as a function of time

Computer Simulation Using Model Equations

From the control loop model equations, a computer program was assembled utilizing the FORTRAN computer language. Solution of these model equations was performed by the fourth-order Runge-Kutta approximation technique for integration. This method lends itself well to digital computer applications.

Upon entering the heater, each segment of oil was integrated according to the incremental time base and changing lengths of the condensate and steam section models. At each time increment, the length of each oil segment was subject to change depending upon

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