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# Therminol FR-2 heat transfer systems

## Engineering design data are given for the design of high temperature systems using chlorinated biphenyls.

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Therminol FR-2 is a mixture of chlorinated biphenyls used as a heat transfer fluid to a maximum bulk temperature of 600°F because of its excellent thermal stability, fire-resistance, and high boiling point. The fluid was previously designated Aroclor 1248. Physical properties of this heat transfer fluid are given in Table 1.

### Thermal stability

When Therminol FR-2 is heated above 600°F, the fluid forms polymers of chlorinated biphenyl with the liberation of gaseous HCl. The higher molecular weight polymers are moderately soluble in the fluid, and the only measure of thermal degradation is the increased viscosity. Figure 1 shows the results of static thermal stability tests carried ont under isothermal conditions.

The polymer formation was found to be first order following Equation 1 until the viscosity increased from 20 centipoises to 30 centipoises measured at 130°F.

$$log(V_o/V_t) = 0.434\, K_t t \qquad (1)$$

Using Equation 1 together with the experimental rate constants, the intermediate viscosity curves for other temperatures were plotted on Figure 1. When the calculated values deviated from the experimental curves, intermediate values determined from special cross-plots were used to smooth the curves.

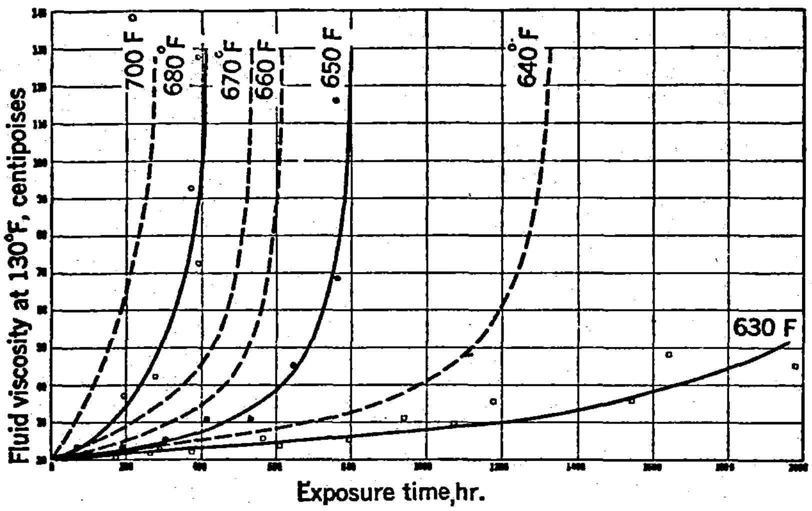



**Figure 1. Effect of temperature and time on the viscosity of Therminol FR-2.**

When fluid bulk or film temperatures exceed 630°F, the fluid viscosity increases rapidly with temperature. In a properly designed commercial heater operating at a bulk temperature of 600°F, with film temperatures below 630-640°F, the over-all rate of polymer formation is extremely low. This conclusion is substantiated from field results of several operating heaters.

## Heater study

A field study of several Therminol heaters was made to establish engineering design data such as allowable heat fluxes, *maximum film temperatures*, fluid stability rates, etc. The test heaters included ones where tube failures occurred shortly after startup and heaters which had been in successful operation for a number of years.

The Lobo and Evans (*1*) method seemed most applicable to evaluate heater performance since it had already been used to test many heaters. The method was based on the Stephan-Boltzmann equation and previously published data by Hottel and co-workers (*2, 3*). More rigorous methods published prior to 1958 appeared too rigorous for easy application and the rough data available did not justify a more rigorous approach. A method was needed to distinguish between a heater which would operate satisfactorily and one which would be a problem, even though the method was empirical.

Equation 2 is the Stephan-Boltzmann equation used by Lobo and Evans with an added convection term.

$$Q_A = 0.173\,\phi\,[(T_G/100)^4 - (T_S/100)^4]\,\alpha A_{cp} + h_C A(T_G - T_S) \quad (2)$$

$\phi$ is dependent upon the flame emissivity, $\epsilon_F$, and the ratio of the effective refractory surface $A_R$ (total refractory surface $A_t$ *less* $\alpha A_{cp}$) and the equivalent cold plane surface $\alpha A_{cp}$. Lobo and Evans presented a $\phi$ factor plot in their article.

Flame emissivity $\epsilon_F$ has been defined by Hottel (*4*) as:

$$\epsilon_F = [\epsilon_G - \alpha_G(T_S/T_G{}^4]/[1 - (T_S/T_G)^4] \quad (3)$$

Values of the combustion gas emissivities and absorptivities can be obtained from radiation charts (*4*) for $CO_2$ and $H_2O$.

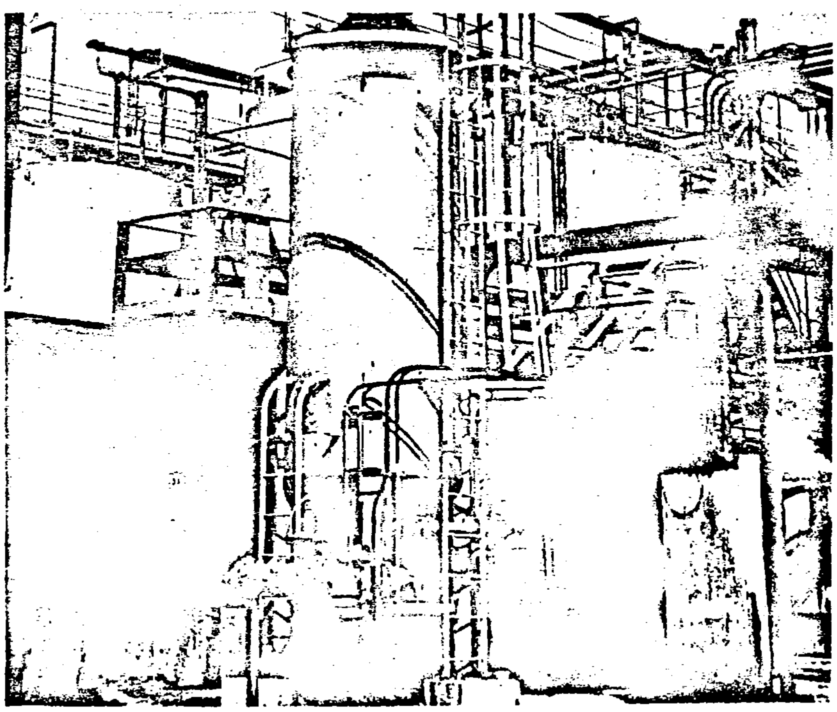
The equivalent cold plane surface $\alpha A_{cp}$ is the product of a plane $A_{cp}$ replacing the heater tube bank and an effectiveness factor which is the ratio of radiation reception by the actual tube surface to the reception by the continuous plane. A plot of $\alpha$ based on data by Hottel is presented in McAdams(*4*).

## Maximum heat flux

Equation 2 together with the overall heat balance, Equation 4, can be solved for the maximum heat flux $Q_a/\alpha A_{cp}$ by assuming values for $T_G$ until Equation 4 is satisfied. Assume 2% of the heat input as losses.

$$Q(\text{Input}) = Q_A(\text{Absorbed}) + Q_G(\text{Exhaust Gases}) + Q_L(\text{Heat Losses}) \quad (4)$$

In applying the previous equations to evaluate the field test heaters, a trial value for $Q$(input) was assumed. If the calculated absorption rate did not agree with the actual rate within 1% after Equations 1 through 4 were satisfied, a new value for $Q$(input) was used until the calculated and the actual duties were equal. This procedure yielded values for $T_G$ and $Q_a/\alpha A_{cp}$ maximum. Heater efficiency estimated by this procedure is incorrect due to the basic assumption that the radiating gas temperature $T_G$ is uniform throughout the heater. The final value of $T_G$ must be adjusted by subtracting 200 to 250°F to $T_G$ for estimating the heat content of the combustion gases leaving the heater (*5*).



**A Therminol FR-2 high temperature heater installation.**

# High temperature heating media

Maximum film temperatures were estimated by calculating the outside fluid heat transfer coefficient $h_{io}$ using Equations 5 and 6 and solving Equaton 7 for the temperature of the film, $T_f$.

$$h_i D_i / k = 0.027\ (D_i G_i / \mu)^{0.8}\ (C\mu/k)^{1/3} (\mu/\mu_w)^{0.14} \quad (5)$$

$$h_{io} = h_i (A_i / A_o) \quad (6)$$

$$Q_A / \alpha A_{cp} = (Q/A)_{max} = h_{io} \Delta T = h_{io}(T_f - T_b) \quad (7)$$

Table 2 is a summary of the data obtained on several of the test heaters. The tubes in units 1 through 4 burned out shortly after startup. The calculated film temperatures in each case were approximately 700°F. Inspection of Figure 1 shows that the fluid decomposition rate at 700°F temperature is very rapid. High polymeric tars rapidly form which settle out and cause blockage of orifices, control valves, headers, etc., and eventual tube failures.

Units 5 through 17 represent heaters which have been in successful operation for the number of years designated and where the fluid is still the original charge. In each case the estimated film temperatures were 640°F or less which correlates very well with the thermal stability data on Figure 1.

## Cylindrical heaters

Lobo and Evans' method was based on the well-stirred furnace chamber model. Kern (6) recommended that in applying the method to cylindrical heaters that the unit be zoned with a maximum zone length of 1.5 coil diameters.

It can be shown that calculated flux rates are much higher in the lower half of a cylindrical heater if the unit is zoned rather than treating it as an isothermal single zone. For this reason each of the cylindrical heaters in the study were analyzed with both single zoned and multi-zoned calculations to determine the film temperatures. The majority of the film temperatures calculated using a lower zone $L/D$ of 2 agreed with the fluid life data and with comparable results obtained in box heaters. When film temperatures approached or exceeded 700°F, tube burn-out or severe fluid damage occurred. Excellent fluid life was experienced in heaters where the film temperatures were below 640°F.

The design of a liquid heating system can be broken down into two major parts. These are: 1. heater design, and 2. system design.

## Heater design

There have been two types of heaters used for Therminol FR-2 service. These are the floor-fired cylindrical heater and the box heater. Each is of the forced-circulation type; natural circulation heaters should not be used because of the non-boiling characteristics of the fluid. Forced circulation provides positive fluid circulation for controlling the film temperature of the fluid near the coil surface.

A single coil heater is preferred to the multi-pass heater from the standpoint of positive circulation. However, by proper design of the multi-tube units to assure uniform distribution of liquid flow, excellent performance can be obtained. In many cases pressure drop across the unit must be low and multipass tube arrangements even in cylindrical heaters must be used.

The use of a convection section increases the heater efficiency from 60-65% to 75-78%, depending upon the design. Although the initial cost of the heater is significantly higher with the added convection section, usually the fuel savings yields a short write-off period. In many cases where a convection section is being used, it is advantageous to introduce the cold fluid into the radiant coil rather than to use the conventional countercurrent flow pattern. The $\Delta T$ across the coil is usually 60°F or less and with a maximum fluid limit of 600°F, little efficiency will be lost due to the reduced temperature approach. However, the effect is significant in the radiant section, because lower bulk temperatures permit higher heat fluxes to maintain the maximum film temperature. Where excessive decomposition is occurring in existing heaters, reversing the fluid in this manner would be most beneficial with very little cost involved.

The Lobo-Evans' method (1) already discussed, can be used for rating heater designs. The maximum allowable film temperature recommended is 640°F based on the maximum flux rate ($Q/\alpha A_{cp}$) determined by the method. Cylindrical heaters should be treated with a 2-zone cal culation (6) with the length of the lower zone equal to $2D$. The zone calculation results in slightly higher coil areas to allow for uneven temperature distribution in the heater and is not representative of actual thermal conditions. Box heaters can be treated with a single-zone calculation.

## Burner selection

The type of burner to be used should be carefully selected by the designer. It has been found that the low pressure combination oil-gas forced draft unit is the most satisfactory unit both from design and operational considerations. The initial cost is many more times that of the high pressure, natural induced type, but the advantages of better temperature control due to its higher turn-down ratio can readily be seen, especially when the heat load fluctuates widely. Most forced drafts units operate at turn-down ratios of 9:1 compared to 4:1 for the induced type. Since each application differs, it should be remembered that the burner is an important component of the system and should be carefully selected for the particular application.

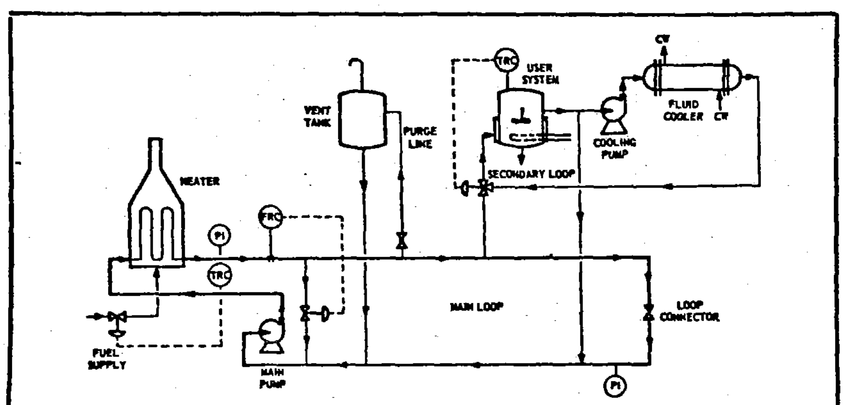



Figure 2. Typical system for heating and cooling cycles with Therminol FR-2.

## System design

Figure 2 illustrates a typical heating system. Major items are the heater, circulating pump, vent tank, primary and secondary loops, and control instrumentation.

The circulating pumps should be high temperature continuous duty steel centrifugal-type pumps. The pump should be installed near the heater at floor level for easy maintenance accessibility. Either indoor or outside installations can be used.

The vent tank can be sized either to allow for system expansion or to contain the entire charge of fluid. It can be fabricated from mild steel and should be insulated to prevent condensation of water vapors and HCl above the liquid level. A 1-in. surge line should be installed connecting the high temperature—high pressure side of the main circulating loop to the vent tank as shown in Figure 2. Hot fluid should be recirculated to the vent tank (above liquid level) continuously or intermittently to remove traces of water and HCl. Caution should be exercised never to use Therminol intermittently with steam or cooling water in the same jacket or coil.

The main circulating loop supplies hot fluid to the users and returns the cold fluid to the heater. It is not recommended even in small installations that the users be placed in series with the loop. Always install the user in parallel or provide a by-pass around the user.

Direct fired heaters are usually provided with instrumentation to obtain: 1. adequate operational control and 2. safety controls which will satisfy Factory Insurance Association (FIA) approval for direct-fired heaters.

Figure 2 shows the fluid temperature-controller output regulating the fuel supply and the flow by-pass controller which maintains the design flow rate through the heater. It is strongly recommended that the by-pass control valve be installed close to the furnace to minimize time lags and excessive pressure drop in the loop piping. The recirculation rate through the heater does not necessarily have to be the same as through the piping system; in most cases, it is larger. These two controllers are the most important portion of the instrument package to assure trouble-free heater performance.

A booklet on good practices has been published by the FIA (7). Approval by FIA is usually obtained during the design stage of any direct-fired heater installation.

**Table 1.** Physical properties of Therminol FR-2.

| Temp., °F | Sp. ht., Btu/lb.-°F | Thermal Conductivity, Btu/hr.-°F-ft. | Density, lb./gal. | Viscosity, cp. |
|---|---|---|---|---|
| 100 | 0.281 | 0.0585 | 12.00 | 62.4 |
| 150 | 0.289 | 0.0580 | 11.75 | 11.50 |
| 200 | 0.297 | 0.0579 | 11.53 | 4.95 |
| 250 | 0.305 | 0.0575 | 11.29 | 2.87 |
| 300 | 0.313 | 0.0570 | 11.06 | 1.87 |
| 350 | 0.321 | 0.0566 | 10.83 | 1.32 |
| 400 | 0.329 | 0.0560 | 10.60 | 0.99 |
| 450 | 0.337 | 0.0555 | 10.37 | 0.78 |
| 500 | 0.333 | 0.0549 | 10.14 | 0.63 |
| 550 | 0.341 | 0.0542 | 9.90 | 0.53 |
| 600 | 0.349 | 0.0536 | 9.67 | 0.45 |

**Table 2.** Therminol FR-2 field heater study.

| Unit | Capacity, Btu/hr. | Bulk temp. | Flux Btu/hr.-sq. ft. Max. | Flux Btu/hr.-sq. ft. Avg. | Max. film temp. °F | Fluid velocity ft./sec. | Fluid life to date, yr. |
|---|---|---|---|---|---|---|---|
| 1 | 6,000,000 | 600 | 19,200 | 9,950 | 696 | 3.9 | B.O. |
| 2 | 20,000,000 | 600 | 22,400 | 10,000 | 705 | 3.9 | B.O. |
| 3 | 5,000,000 | 430 | 21,400 | 13,700 | 694 | 1.6 | B.O. |
| 4 | 3,000,000 | 600 | 18,350 | 7,550 | 726 | 5.0 | B.O. |
| 5 | 6,000,000 | 600 | 19,200 | 9,950 | 640 | 7.8 | 4 |
| 6 | 2,000,000 | 600 | 48,800 | 20,000 | 675 | 8.6 | 7 |
| 7 | 1,000,000 | 530 | 7,400 | 3,600 | 593 | 2.5 | 11 |
| 8 | 3,000,000 | 550 | 12,900 | 6,700 | 627 | 3.8 | 8 |
| 9 | 6,000,000 | 585 | 17,850 | 9,300 | 631 | 10.3 | 4 |
| 10 | 10,000,000 | 475 | 23,300 | 13,200 | 573 | 3.2 | 4 |
| 11 | 5,000,000 | 550 | 21,400 | 11,600 | 631 | 4.0 | 7 |
| 12 | 3,000,000 | 550 | 18,350 | 7,550 | 640 | 6.7 | 8 |
| 13 | 1,500,000 | 600 | 10,000 | 6,400 | 635 | 5.8 | 6 |
| 14 | 3,000,000 | 570 | 19,150 | 9,950 | 640 | 8.0 | 5 |
| 15 | 3,000,000 | 535 | 27,400 | 17,800 | 595 | 12 | 2 |
| 16 | 5,000,000 | 600 | 19,600 | | 656 | 9.7 | 2 |
| 17 | 9,000,000 | 590 | 12,500 | | 646 | 6.0 | 1 |

*B. O. Burn out.*

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## Notation

$A$—area, sq. ft.
$C$—fluid specific heat, Btu/lb.-°F.
$D$—diameter, ft.
$G$—mass flow rate, lb./hr.-sq-ft.
$h_1$—rate constant.
$h_o$—convective heat transfer coefficient, Btu/hr.-°F-sq.-ft.
$h_i$—tube-side fluid heat transfer coefficient, Btu/hr.-°F-sq.-ft.
$h_{io}$—tube-side fluid heat transfer coefficient based on out-side tube area, Btu/hr.-°F-sq.-ft.
$k$—fluid thermal conductivity, Btu/hr.-°F-ft.
$t$—time, hr.
$T_G$—temperature of the combustion gases, °R.
$T_S$—tube wall temperature, °R.
$V_o$—initial fluid viscosity, cp.
$V_t$—fluid viscosity at time ($t$), cp.

Greek letters

$\epsilon_G$—gas emissivity.
$\mu$—fluid viscosity, lb./hr.-ft.
$\phi$—over-all furnace emissivity.

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