ESTIMATION OF THE RECOVERABLE HEAT ENERGY FROM THE ENGINE EXHAUST GASES

1. INTRODUCTION AND SUMMARY

In order to maximize the efficiency and cost effectiveness of the Reaction Engine Powered Generating System (REPGS), it is desirable to extract as much heat energy as possible from the engine exhaust gases. For this purpose, heating of water is considered in three different stages, viz:

a) water jacket across the exhaust deflector plate;
b) water jacket around the exhaust pipes; and
c) a gas-to-liquid heat exchanger.

About 1000 gpm of water can be heated from ambient (90^{o}F) to near boiling temperatures. The corresponding heat recovery is about 60 x 10^6 to 90 x 10^6 BTU/hr, which is 10 to 15% of the total heat or work* potential of the exhaust gases.

2. SYSTEM PARAMETERS

The basis and calculated values of the heat transfer rates etc. are described in the following section. However for ready reference, first the assumed base system parameters are summarized below.

No. of engines = 4
Exhaust gas flow rate/engine = 250 lb/s
Exhaust gas temperature = 850^{o}F
Exhaust gas density = 0.0299 lb/ft^3
Exhaust gas velocity = 1190 ft/s

Exhaust deflector plate height = 75 ft.
Exhaust deflector depth = 40 ft.
Exhaust deflector plate area = 8282 ft^2

Ambient water temperature = 90^{o}F

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*Conversion factor for thermal to mechanical energy; 1 BTU $\equiv$ 778 ft lbf

3. RATE OF HEAT TRANSFER ACROSS THE EXHAUST DEFLECTOR PLATE

A water jacket can be formed on the convex side of the deflector plate, to preheat the water. The rate of heat transfer will depend upon the following factors.

i) Film heat transfer coefficient of gas, h_g, which is a function of local flow and fluid properties such as Reynolds and Prandtl numbers.

For a confined gas passage, the heat transfer coefficient can be estimated as:

$$hg = 0.023 \, Re^{-0.8} \, Pr^{0.667} \frac{k}{D_e}$$

ii) Heat transfer coefficient of metal, h_m, which depends on its thermal conductivity, k, and thickness, t:

$$h_m = \frac{k}{t} .$$

iii) Film heat transfer coefficient of liquid (water), h_ℓ, which is a function of local Reynolds and Prandtl numbers, similar to the one given above for gas film.

iv) Fouling factors for the metal surface particularly on the liquid side, which depends on corrosion, etc.

v) Temperature differences between the gase and liquid.

vi) Overall flow arrangement, for example parallel, counter or cross-flow of two streams.

For the purpose of steady state analysis, the first four factors can be usefully combined into one quantity, ie. "overall heat transfer coefficient", U, defined as follows:

$$\frac{1}{U} = \frac{1}{h_g} + \frac{1}{h_\ell} + \frac{1}{h_m} + \frac{1}{h_f}$$

Since U is the harmonic mean, its value is smaller than and closest to the smallest of the individual heat transfer coefficients. For this reason, the fouling factor on the liquid side and the gas side film heat transfer coefficient are expected to play the limiting role. From the study of References 1, 2 and 3, the best estimate for the overall heat transfer coefficient is:

$U = 5$ BTU/hr ft^2 ^oF .

The expected range of U values for various possible designs is 2 to 10 BTU/hr ft^2 ^oF. For a finalized conceptual design, it is possible to make more accurate predictions with the aid of the state of the art mathematical models, such as in Reference 4.

The last two factors (items v and vi) viz. temperature difference and flow arrangement, go hand in hand. The net effect of flow arrangement is accounted for by calculating the so called "Log Mean Temperature Difference" (LTMD), which is defined as:

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ell \; \dfrac{\Delta T_1}{\Delta T_2}}$$

where ΔT_m = LTDM, and ΔT_1 and ΔT_2 are the differences between the two fluids (gas and liquid) at the two ends of the jacket. It is well known that the counter-flow arrangement yields a relatively higher value of ΔT_m. The cross-flow effectiveness is somewhat intermediate. However when both ΔT_1 and ΔT_2 are significantly large.the LMTD value is not very sensitive to the flow arrangement. Therefore in the present application, there will not be much influence on the heat transfer due to the upward or downward direction of water flow.

In order to estimate the total heat transfer across the deflector plate, several trial calculations were performed, and the most suitable one considers that the gas temperature drops from 850^{O}F to 800^{O}F, and water temperature rises from 90 to 150^{O}F.

Then,

$$\Delta T_1 = 850 - 150 = 700^{o}F$$
$$\Delta T_2 = 800 - 90 = 710^{o}F$$

$$\Delta T_m = \frac{710 - 700}{\ell n \frac{710}{700}} \simeq 700^{o}F$$

Total heat transfer, $Q = UA\ \Delta T_m$

$$= 5 \times 8282 \times 700 \text{ BTU/hr}$$
$$= 28.98 \times 10^6 \text{ BTU/hr}$$
$$\simeq 30 \times 10^6 \text{ BTU/hr}$$

This heat extraction rate is about 5% of total heat potential (above ambient) of exhaust gases.

The corresponding water flow rate, $\dot{m}_w$, is:

$$\dot{m}_w = \frac{Q}{c_p\ (T_{hot} - T_{cold})}$$

$$= \frac{30 \times 10^6}{1 \times (150 - 90)} = 0.5 \times 10^6 \text{ lb/hr}$$

$$= \frac{0.5 \times 10^6}{62.5 \times 0.1337} = 5.98 \times 10^4 \text{ Galons/hr}$$

$$\simeq 997.3 \simeq \underline{\underline{1000 \text{ gpm}}}$$

In order to maintain a good heat transfer rate, it is desirable to have a reasonable flow velocity in the water jacket; otherwise, the liquid film heat transfer resistance as well as the fouling resistance increase. Therefore a design with smaller cross-section of jacket will be preferred.

4. RATE OF HEAT TRANSFER IN EXHAUST PIPES

From the general arrangement diagram of the REPGS, it can be seen that the exhaust gases are passed through 3 or 4 exhaust pipes. These pipes can be surrounded by water jackets carrying the same water which was heated across the deflector plate. From the heat transfer considerations, a counter-flow arrangement is preferred and is assumed feasible here. The trial calculations indicated that the water temperature could be raised from 150^{o}F to 210^{o}F. Since the water flow rate and temperature rise are same as in deflector jacket, the net heat transferred is also the same ie. 30 x 10^{6} BTU/hr or about 5% of the exhaust gas heat potential. However, due to the better control on liquid and gas velocities, a relatively higher overall heat transfer coefficient (U = 10 BTU/hr ft^{2} ^{o}F) is feasible here. Therefore, the required heat transfer surface area, A, is calculated as follows.

$$A = \frac{Q}{U.\ \Delta T_m} = \frac{30 \times 10^6}{10 \times 600}$$

$$= 500^{o} ft^2 .$$

From the general arrangement diagram of REPGS, 3 pipes of 8 foot diameter and 60 ft. length, seem quite suitable. These pipes have a surface area of:

$3 \times \pi \times 8 \times 60 = 4523\ ft^2$,

which is fairly close to the required value. The actual values can be finalized from the overall system considerations.

5. RATE OF HEAT TRANSFER IN A HEAT EXCHANGER

In order to further utilize the exhaust gase heat, it would be necessary to employ the following:

a) a well desinged gas-to-liquid heat exchanger;

b) a booster pump to force the exhaust gases through the heat exchanger, without raising the pressure of engine exhaust.

The cost effectiveness of above system would depend upon the utilization of hot water. It may also be possible to convert this pre-heated water into steam, by using direct fired heaters or boiler, and use steam for supplementary power generation via a low-pressure turbine. For such a system, the condensed water will be passed through the deflector jacket and exhaust pipe jackets.

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