Design of a finned radiator assembly

Design

The initial parameters were the operating requirements of the Diesel-Engine Generator Set 1500-EXCITED are as follows: Coolant capacity - The coolant chosen for our radiator is ethylene glycol (50/50 % by volume) Its maximum operating temperature of OFF Alarm flow rate - Since the generator Is stationary as opposed to that used In an automobile application, a fan will be needed to provide the necessary flow rate. The required air flow rate specified by the engine is 9. 383 Is in order to dissipate the heat generated Coolant flow rate - The coolant flow rate is 17. 14 keg/s through the radiator The initial coolant temperature is assumed to be OFF, which is slightly low the operating temperature of the engine. The initial coolant temperature Is taken as the ethylene glycol entering the radiator Immediately after leaving the engine. Pressure drop allowance - The The total heat rejected to the coolant is skew The outlet temperature of the coolant leaving the radiator was calculated to be DOFF Assumptions In order to design a radiator for a specific operating condition, we assume the Florida last year.

This is accounting for the worst case scenario. We also assume that the ambient air density is constant throughout operation. We also assumed that here was no significant fouling on the inside of the tubes, so the heat transfer coefficient of the materials remains constant. Design Methodology Identification of Problem The heat rejection requirements were specified by the diesel-engine generator's operating conditions. Our objective was to reduce construct a radiator that would provide better performance while functioning at a higher efficiency by reducing size and cost.

Selection of an Appropriate Heat Exchanger Classification The conventional and most effective form for a radiator in this application is the late-fin and tube heat exchanger operating in cross-flow conditions. Rather than attempting to create a more innovative heat exchanger assembly while sacrificing simplicity, the present work aimed to improve and maximize the performance of the current heat exchanger type. Material Selection The materials of each specific component were chosen based primarily on thermal conductivity as compared to price.

The cost analysis of each material can be observed in Table 1 shown below. Tablet . Cost analysis of the selected materials Component Description Unit Size Weight (lbs) Price per Unit Size Reel. Weight Reel. Cost Fins ASTM 8370 110 copper Sheet, 0. 08" thickness 90. 0239604 $960. 11 75. 35208664 $803. 63 Tubes Seamless Brass 270 Tubing, 0. 03" thickness 96" x 0. 5" DO 15. 7152 $16. 04 3771 . 648 $3, 849. 60 Cover Plates ASTM 836 260 Brass Sheet, 1/16" thickness 66. 5280198 $709. 52 60. 29101795 $643. 00 3907. 291 105 $5, 296. 24 Table 2.

The estimated manufacturing cost Manufacturing Hours $/her Cost Design Time $240. 00 Machining Time 2 12 $24. 00 Assembly Time 10 15 $150. 00 Miscellaneous $100. 00 $514. 00 $530. 24 The working fluid selected for the radiator was ethylene glycol (50% by mass), due o its prominent use in convective heat transfer applications, particularly in automobiles. Selection of Provisional Dimensional Parameters In order to design an ideal heat exchanger with minimal cost and size, preliminary measurements were chosen and initial calculations were run to provide a scope of the required parameters.

The radiator core's dimensions were determined by comparing the conventional radiator specified by the Diesel-Engine Generator. Its size was slightly reduced, since an increase in heat transfer in the tubes was anticipated in our design. The initial dimensions considered are listed in Table 3 low. Table 3. The initial dimensions chosen to gauge heat rejection rates Width (m) 2. 2098 Height (m) 2. 4384 Depth (m) 0. 12065 Number of Fins 1140 XSL (m) 0. 0254 Ext (m) 0. 0001 DO (m) 0. 127 The use of previously derived correlations for the inner-finned tubes was the primary means of determining heat rejection requirements. The pitch for the ribs was calculated using, direction of the tube. The Reynolds number was derived by, where the mass flow rate is that of the ethylene glycol. The friction factor for flow inside the tubes can be determined using the following equation. This correlation is valid for turbulent flows within tubes, and relates the pressure loss due to friction in the tubes to wall shear stress.

This pressure drop can be determined using the preceding equation. This pressure drop in turn governs the pumping power required for the coolant mass flow rate. The Mussels number required to find the heat transfer coefficient needed to determine heat transferred from the engine to the coolant to the air is given by the following correlation for turbulent flow. The local heat transfer coefficient for finned tube as compared to the coefficient for lane tubes is illustrated by the following equation.

The friction factor for the finned tubes can be determined using the original friction factor through the following correlation. (7) Design Optimization For the preliminary design of our radiator, some optimization was performed for the heat transfer in the coolant tubes, fins, inner ribs, as well as other surface areas. Local heat transfer coefficient was calculated and compared for various cases of inside tube diameters, rib angles, and rib heights, as well as the number of ribs. These results can be observed in figures 4 and 5.

Figure 5.

Local heat transfer coefficient as a function of rib angle The pressure drops were also calculated for differenttubediameters as well as rib angles.