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Optimization of a radiator for a MPFL system in a GEO satellite

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Abstract. One of the components that used in the satellite thermal control subsystem is the Mechanically Pumped Fluid Loop (MPFL) system; this system mostly used in geosynchronous orbit (GEO) satellites, and can transfer heat from a hot point to a cold point using the fluid which circulated in a closed loop. Heat radiates to the deep space at the cold plate to cool down the fluid temperature. In this research, the radiative heatexchanger (RHX) for a MPFL system is optimized. The genetic algorithm has been used for minimizing the total mass and pressure drop by considering a constant transferred heat rate at the heat exchanger. The optimization has been done in two cases. In case I, two parameters are considered as a goal function, so optimization is performed using NSGA-II method. Results of optimization are shown in the pareto diagram. In case II, the diameter of pipe is considered constant, so the optimized value for distances of the parallel pipes is obtained by using the genetic algorithm, in which the system has the least total mass. Results show that in the RHX, by increasing the pipe diameter, pressure drop decreases and total mass increases. Also by considering a constant value for pipe diameter, an optimum distance between pipes and pipe length are obtained in which the system has a minimum mass.

Keywords: mechanically pumped fluid loop (MPFL); geosynchronous orbit satellite (GEO); optimization; genetic algorithm (GA)

1. Introduction

In geosynchronous satellites, generated heat from the payload subsystems or other subsystems should be transferred to the cold point to prevent the satellite components from damaging. For controlling the satellites components temperatures and transferring the generated heat to a cold place, a thermal component should use which has a long lifetime and can transfer a considerable amount of heat to the radiator. One devices which are used for that purpose is mechanically pumped fluid loop (MPFL) system. This system consists of a closed loop path, which filled up with a fluid. The pump in the system produce a positive pressure to circulate the fluid in the route and causes to heat transfers from the hot point to the cold point.

The one-phase or two-phase fluid can be used in this system. A simple schematic of the MPFL system is shown in Fig. 1. As seen, this system consists of a pump, an accumulator, a fluid control valve, a payload heat exchanger (PHX) and a radiative heat exchanger (RHX). Fluid insides the

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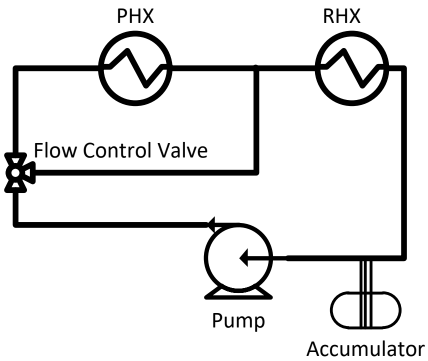

Fig. 1 Schematic of a MPFL system

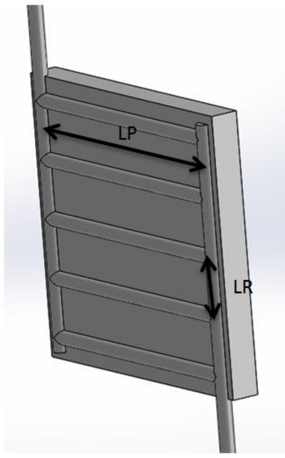

Fig. 2 Radiative Heat Exchanger (RHX)

pipe transfers the heat from the PHX to the RHX. Amount of fluid which passes through the PHX is controlled by the fluid control valve, so extra flow turns back to the RHX using the bypass rout. An accumulator is connected to the system for keeping the pressure constant by enlarging or contracting regards to the fluid volume change.

In this research, a radiative heatexchanger is designed and optimized. In Fig. 2, the configuration of the pipes in this system is demonstrated. As seen in the figure, inlet flow is divided into the equal parallel paths and connected together after passing through the heatexchanger surface and exits from the RHX. Parallel pipes are connected firmly to the radiator. Fluid is cooled down by dissipating heat to the deep space.

An MPFL system which used in a satellite, should has a minimum total mass and pressure drop for a constant amount of the transferred heat.

Heat exchangers have been surveyed by many researchers up to now. Most of the researches have been done for non-radiative heat exchangers. Linnhoff and Ahmad (1990) optimized a heat exchanger by surveying a fluid in a network of heat exchangers using simple analytical relations. The parameters for which the optimization is done are the capital investment, the pumping electricity cost and the heat transfer rate in the radiators. Naumann (2004) considered fins that radiates to the deep space and optimized the plate thickness and width using semi-analytical

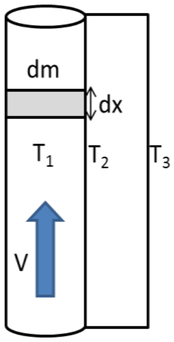

Fig. 3 Differential element of the pipe

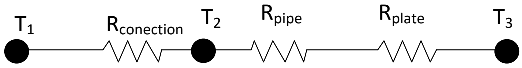

Fig. 4 Thermal model of the pipe

relations to obtain a maximum value for transferred heat rate per mass in a special heat exchanger. Arslanturk (2006) optimized a high temperature radiative heat exchanger using mathematical series method and considered the parameters as a function of temperature. This heat exchanger is used in a power plants and analytical method is used to optimize it. In another research done by Xie *et al.* (2008) a CHE type heat exchanger has been surveyed and optimized using the genetic algorithm. The goal functions which were optimized are heat exchanger volume and annual electric cost for pump by optimizing the geometry parameters. Epsilon-NTU method has been used in this research. Sanaye *et al.* (2010) surveyed a finned heat exchanger and flow in the perpendicular direction. Geometrical parameters of the fin are optimized using the genetic algorithm (NSGA-II) by semi-practical relations and epsilon-NTU method to obtain the optimized heat transfer rate. The optimum model has the least pressure drop and the maximum efficiency. The obtained results are shown in a pareto diagram. Rao (2010) considered the combination of the cost, heat transfer rate and entropy to optimize the fin-plate heat exchanger by the PSO algorithm; he also compared the results of that method with the genetic algorithm. PSO optimization method has been used to optimize a shell and tube heat exchanger by Patel (2010) which two types of configuration are considered for the heat exchanger. In each configuration, three geometry parameters are considered to optimize the total cost, which consists of the capitalizing cost, electric annual cost and maintenance cost. Najafi (2011) optimized the cost and heat transfer rate by NSGA-II method to obtain the geometry parameters. In recent research by Kumar Rai (2015), amount of heat transfer rate per heatexchanger mas is optimized in different conditions using analytical methods. Zhao *et al.* (2012, 2016) surveys the effects of the permeability and heat transfer of methane hydrate dissociation by depressurization using the numerical and experimental methods.

2. Modeling

For modeling the heat transfer in the heat exchanger, one of the parallel pipes is considered and heat transfer rate is obtained. As shown in Fig.3, heat transfers from the fluid (T1) to the other side of the pipe (T2) and outer side of the radiator (T3). Heat from (T3) radiates to the deep space. Simple schematic of the heat transfer model is shown in Fig. 4. For modeling the heat transfer, a differential element of the pipe is considered as shown in the Fig. 3.

Amount of the heat transferred from the shown finite volume is

$$\dot{m}C_p dT_f = d\dot{q} \tag{1}$$

In which $d\dot{q}$, dT_f, C_p and $\dot{m}$ are differential heat transfer rate, differential temperature increment, specific heat specific capacity and mass flow rate in the pipe; The amount of heat transfer rate in the element is

$$d\dot{q} = \varepsilon\sigma dA T_{fin}^4 \tag{2}$$

Where T_{fin}, dA, σ, ε are fin temperature, differential radiator surface area, Stephen Boltzmann constant and surface emissivity respectively.

By considering the radiator effective surface $L_R dx N_s$ (N_s = number of radiative surface) and substituting Eq. (2) into the Eq. (1), we have

$$\dot{m}C_p dT_f = \varepsilon\sigma L_R dx N_s T_{fin}^4 \tag{3}$$

As shown in Fig. 4, the total thermal resistance is

$$d\dot{q} = \frac{T_1 - T_3}{R_{tot}} \tag{4}$$

where R_{tot} is the total heat resistance between the fluid and the outer surface. By considering Fig. 4 we have

$$R_{tot} = R_{convection} + R_{pipe} + R_{plate} = \frac{1}{h\Gamma dz} + \frac{\ln\left(\frac{D_0}{D_i}\right)}{2\pi dzk} + \frac{t}{k_s L_P dz} \tag{5}$$

D_o and D_i are the inner and outer diameters of the pipe respectively. Assume that pipe is completely filled with fluid and the entire perimeter is wet, so $\Gamma = \pi D_i$ and heat transfer coefficient is obtain from the relations mentioned by Incropera.

As the Reynolds number is $\mathrm{Re} = \frac{\rho v D}{\mu} < 2300$, the flow regime inside the pipe is laminar, so the heat transfer coefficient in the pipe can be obtained by relation

$$Nu = 1.86\left[\mathrm{Re}\,Pr\frac{D_i}{L_R}\right]^{\frac{1}{3}} \text{ if } \mathrm{Re}\,Pr\frac{D_i}{L_R} > 2, \mathrm{Pr} > 0.5$$

$$Nu = 3.66 \qquad \text{if } \mathrm{Re}\,Pr\frac{D_i}{L_R} < 2$$

Reynolds number (Re), Prandtel number (Pr) and heat transfer coefficient (h) are defined as

$$\mathrm{Re}=\frac{4\dot{m}}{\pi\mu_l D_i};\quad Pr=\frac{\mu_l C_{p,l}}{k_l};\quad h=\frac{Nuk_l}{D_i}$$

Variable μ_l, $C_{p,l}$, k_l and D_i are dynamic viscosity, specific heat capacity, conduction coefficient, pipe inner diameter and mass flow rate respectively.

Conduction coefficient of the heatexchanger surface is considered k_s=205 W/mK.

The number of active radiator surface, N_s can have value 1 or 2. By considering the differential area for the radiator and substituting it into Eq. (4), we have

$$\frac{T_1-T_3}{R}=\dot{q}=N_s\varepsilon\sigma dAT_3^4 \tag{6}$$

Substituting into Eq. (5) for i[th] element, we have

$$T_{3,i}=T_{1,i}-\left(\frac{t}{kL_p dz}+\frac{1}{h\Gamma dz}+\frac{\ln\left(\frac{D_0}{D_i}\right)}{2\pi dzk}\right)N_s\varepsilon\sigma dzL_pT_{3,i}^4 \tag{7}$$

To obtain the temperature distribution along the pipe in steady state condition, according to Eqs. (2) and (6) we have

$$T_{1,i+1}=T_{1,i}-\frac{N_s\varepsilon\sigma dzL_PT_{3,i}^4}{\dot{m}c_p} \tag{8}$$

and initial condition is $T_1=T_{in}$.

To find the temperature distribution along the pipe and amount of heat transfer, temperature T3 is obtained for ith element using Eq. (7) and numerical calculation, subsequently, T1 for (i+1)th element is obtained by Eq. (8), and this calculations are repeated until he last element of the pipe.

3. Assumptions and simplifications

In this problem, thermo-physical properties of the fluid and the surfaces are considered to be constant. Conduction heat transfers along the pipe is neglected, also it assumed that, there is not any heat radiation happened to/from the surface except the solar radiation.

The heat exchanger is optimized, for minimizing the total system mass and pressure drop.

The total mass is the sum of the masses of the pipes, fluid and the radiator. For Nt number of parallel pipe we have

$$M_{tot}=M_l+M_t+M_P=N_t\left(m_l+m_t+m_P\right) \tag{9}$$

For each term, we have

$$m_l=\rho_l\left(\frac{\pi}{4}D_{t,i}^2L_R\right)\text{for fluid mass} \tag{10}$$

$$m_t = \rho_l \left[\frac{\pi}{4} \left(D_{t,o}^2 - D_{t,i}^2 \right) L_R \right] \text{ for pipe mass} \tag{11}$$

$$m_P = \rho \left(2 L_R L_p t \right) \text{ for radiator mass} \tag{12}$$

As shown in Fig. 2, the routs from the input to the output of the heatexchanger pass from each of the parallel pipes has the same length, so the pipes have equal flow causes even distribution of flow between pipes.

In the heat exchanger, power loss due from the pressure drop should be minimized. As the fluid regime inside the pipes is laminar, pressure drop in each pipe is

$$\Delta P = 128 \mu \frac{L_R}{D^4} Q \tag{13}$$

As a designing restraint, mass flow rate considered to be constant. The power loss at the pipe is

$$P_{pump} = \dot{m} \Delta P = \dot{m} \times 128 \mu \frac{L_R}{D^4} Q = \frac{\dot{m}^2}{\rho} \times 128 \mu \frac{L_R}{D^4} \tag{14}$$

It assumed that fluid in the heatexchanger remains in liquid state, so the pressure change does not have noticeable effect on amount of the heat transfer rate.

3.1 Optimization

The amount of transferred heat, in the RHX is a function of the pipe length (LP), distance between parallel pipes (LR) and the pipe diameter (Di) (As shown in Fig. 2). To reach an optimum design, the total mass and pressure drop should be minimized. Heat transferred is considered equal to 600 watt and optimization for geometry is applied for that assumption.

Two cases are considered for optimization. In case I, two optimization goal functions are considered and total mass of the heatexchanger and pressure drop are minimized. In case II, by

Table 1 parameters that used in process of optimization the RHX

Number of pipes Nt	20
Total amount of the heat transfer	600 watt
Surface thickness	0.6 mm
Stephen Boltzmann constant	5.67×10^{-8} $W/(m^2 T^4)$
Specific heat capacity of aluminum	1800 J/kgK
Water density	1000 kg/m^3
Aluminum density	2840 kg/m^3
Dynamic viscosity of water	1.3064×10^{-3} $Pa.s$
Heat conduction coefficient of water	0.106 W/mk
Heat conduction coefficient of aluminum	170 W/mk
Emissivity (ε)	0.8
Mass flow rate	0.002 kg/s

Table 2 Parameters that used in NSGA-II method for optimization

Initial population	500
Cross over displacement	0.8
Selection function	Tournament
Number of generations	120 (Ns=2) 109 (Ns=1)

Table 3 Accepted range of parameters for optimization

$0.01<LP<0.5$	Length of each pipe
$0.001<Di<0.1$	Inner diameter of the pipe (m)

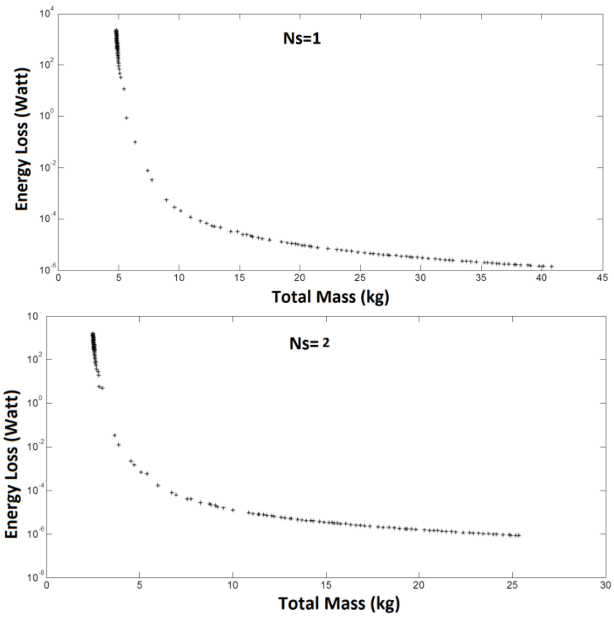

Fig. 5 Result of the optimization shown in pareto diagrams in Case I for Ns=1,2

considering a fixed diameter for pipes, an optimum value for heatexchanger mass is obtained. Parameters that used in process of optimization the RHX are shown in Table 1.

3.1.1 Optimization in case I

In this section, optimization is performed by considering two goal functions.

Total mass and pressure drop are considered as a goal functions which should be minimized using NSGA-II method. Parameters that used in the optimization are shown in Table 2. Optimization is performed for two conditions: Ns=1 and Ns=2.

Design parameters for optimization are pipe length (LP) and initial diameter (Di). Distance between the pipes (LR) is calculated based on the heat transfer rate. The accepted ranges of parameters for optimization are shown in Table 3.

3.1.2 Optimization in case II

Table 4 Results of optimization in case II

	Ns=1	Ns=2
LP (m)	0.1307	0.1008
LR (m)	0.6109	0.425
Optimum value of total mass (kg)	6.3656	3.561
Wasted energy in the pipe (W)	0.409	0.028

In this section, optimization is performed by considering one parameter as a goal function while the pipe diameter is considered 10 mm. Optimization is done using the genetic algorithm, for having the least total mass and obtains the optimum value for length LR.

4. Results and discussion

4.1 Case I: Optimization for two goal functions

In this condition, two parameters of mass and pressure drop should be optimized. Pareto diagrams for the results are shown in Fig. 5 (Note that plots vertical axes are shown in logarithmic scale). Two parameters are in a contrary, so in the optimum conditions, by decreasing one parameter (total mass or pressure drop), the other parameter increases and vice versa. In this diagram, each node in diagram shows an optimum value for the vertical parameter (energy lost due from pressure drop) by assuming a fixed value for the horizontal parameter (total mass).

4.2 Case II: Optimization for one goal function

In this condition, geometry is optimized by considering the constant value for pipe diameter, for that purpose consider that Di=10 mm and the optimum value for LP and LR are calculated using the genetic algorithm. The optimum value for optimization is shown in Table 4.

5. Conclusions

In this research, heat transfer in a radiative heat exchanger (RHX) is modeled in one-dimensional condition and geometric parameters are optimized using the genetic algorithm. Amount of transferred heat and wasted energy due from the pressure drop inside the pipe is considered as a function of the geometrical parameters. By increasing the diameter, heat transfer, energy loss due from pressure drop and total mass are decreased. The optimum condition is a compromise between the minimum mass and pressure drop for the constant value of the transferred heat. By considering the fixed diameter of heat exchanger pipe, the optimum value for the geometry is obtained that provides the minimum mass for the heatexchanger.

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