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TRANSIENT ANALYSIS OF AIRCRAFT OIL SUPPLY SYSTEM WITH FUEL-OIL HEAT EXCHANGERS DURING ABRUPT CHANGE IN ENGINE OPERATING MODES

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ABSTRACT

During different airplane flight modes, various effects may appear that need to be analyzed for both the oil and the fuel system at steady-state and transient operating modes. The effects, which relate to the cold temperature, associated with fuel freeze or wax point, cause a malfunction in the fuel pumps, nozzles, and other areas of the fuel system. On the other hand, high fuel temperature also leads to negative effects - the most common failure of high-flow fuel systems is cavitation, or "vapor-lock." The combination of too much heat or too much inlet restriction can create this operating condition, where the liquid fuel literally boils inside the fuel pump. These effects are eliminated by the fuel/oil heat exchange system. In case of low fuel temperature, the fuel is used as a refrigerant to cool down hot oil coming from bearings. And in case of high fuel temperature, the oil serves as a coolant.

This paper considers the method of evaluating normal and critical aircraft engine operation modes of the oil supply system with a fuel-oil heat exchanger utilizing an unsteady-state thermal-fluid network approach. The analyses are done based on the aircraft engine example to evaluate fuel and oil systems parameters variation in time under different flight conditions – the amount of fuel in the tank, inertial thermal effects, and the response time of the system to the regulation of the heat exchanger. The article is focused on sudden switching from a high to low gas engine operating mode. Fuel consumption to the engine is reduced abruptly, but the heat transfer from the bearings to the oil is still high due to thermal inertia. In this situation, a large amount of heated fuel must be returned to the fuel tank. At a certain point in time, the temperature of the fuel can reach a critical value. At the same time bearing cooling becomes ineffective, which leads to overheating. The calculation of thermal management system was performed at nominal conditions to obtain the initial data for low power settings analysis. As results of analysis at the low power settings mode the oil temperature before fuel cooled oil cooler is reached above 138 ⁰C, which is high value. The failure of flow return valve is considered. The variations of oil temperature after the tank and increasing of fuel temperature at the tank in case of emergency situation are obtained. The influence of cooled fuel amount on the system thermal management is analyzed.

Keywords: oil system, secondary flows, heat transfer, transient analysis, fuel supply system, fuel cooled oil system, thermal-fluid network.

NOMENCLATURE

α	heat transfer coefficient
η	efficiency
λ	Darcy friction factor
ρ	density
τ	bypass time
ζ	resistance coefficient
A	cross-sectional area
B	correlation coefficient
C	specific heat capacity
D	correlation coefficient
D_h	hydraulic diameter
Eu	Euler number
F_{ext}	external forces
G	mass flow rate
$G_{f.bypass}$	fuel mass flow rate in a bypass line
$G_{f.eng}$	fuel mass flow rate through the engine
GP	geometry parameters
LHS	time derivatives
Nu	Nusselt number
$\Pr$	Prandlt number
Q	heat flow
Re	Reynolds number
S_q	heat source
T	temperature

ΔT_{ζ}	fuel heating in supply pipes
ΔT_{HEX}	fuel heating in heaters
$T_{\sup.\tan k}$	fuel temperature in supply tank
V	volume
W	power/friction power
dt	temperature difference
h_0	total enthalpy
l	length
m	fluid mass
t_f	current fuel temperature in supply tank
t_0	initial fuel temperature in supply tank
p	static pressure
Δp^*	total pressure drop
x	correlation coefficient
y	correlation coefficient
z	correlation coefficient

1. INTRODUCTION

The reliable operation of a lubrication system is critical for any type of aircraft engines, whatever type they are. A typical variant of the lubrication system is shown on (see Fig. 1). The system serves multiple functions to ensure the reliable operation of the engine. The lubrication of gears, gearboxes, and bearings prevents metal-to-metal contact between moving parts, supplying a squeeze film between contacting surface, thus reducing friction and power losses. Cooling function, which is especially important in the hot turbine area, where in addition to bearings self-generated heat significant amount of heat flux comes from rotor and stator components. Removing of the contaminants from the lubricant.

The removal of heat generated by friction and transmitted from heated parts increases the temperature of the oil in the engine, which is limited by the ultimate temperature at which thermal stability is still maintained. This temperature level within the specified range (70...90 ^{0}C) providing reliable engine operation [1].

Either air or fuel (in some cases both of them) can be used as a cooling refrigerant depending on the engine type, its power, and specific heating conditions. The oil cooling process takes place in fuel cooled heat exchanger transmitting excessive heat from oil to fuel or air. Typically, the heating of the fuel increases with the growth of the required heat removal quantity and decreases with the growth of fuel consumption. Thus, the usage of fuel as the refrigerant in turboprop engines is not suitable because the heat generation value from reduction gears and gearboxes is too large compared with other types of gas turbine engines. That is why in turboprop engines oil has to be cooled by air or should be considered usage of both - fuel and cooling systems. The hot air after oil cooling can be used for environmental control systems. On the other hand usage of fuel-oil heat exchange in turbojet engines provides the possibility to increase the thermal efficiency of the gas turbine engine (GTE) due to additional using of heat from the oil to increasing the temperature of the fuel before burner.

Design of oil and fuel thermal management system of both air and fuel cooling system has to take into account the most critical working regimes of aircraft - ground idle and top-off flight idle. While the engine is running at low power settings, fuel cooling systems are most tense, thus this regime must be accurately estimated. Providing acceptable work of fuel cooled oil system and permissible temperature of both fuel and oil is a highly important and complex task. This is evidenced by the experience of designing and calculating both subsonic and supersonic aircraft. In some cases, the additional cooling system has to be implemented. Thus, considering the history of XB-70 design, the usage of fuel as coolant was efficient for each flight mode except the low gas one. To prevent the system from overheating, the water was used as an additional coolant [2].

The effects of system short-time overheating can be analyzed using unsteady-state approaches. The article is focused on the calculation of fuel-oil systems and their assessment at low gas critical operating modes.

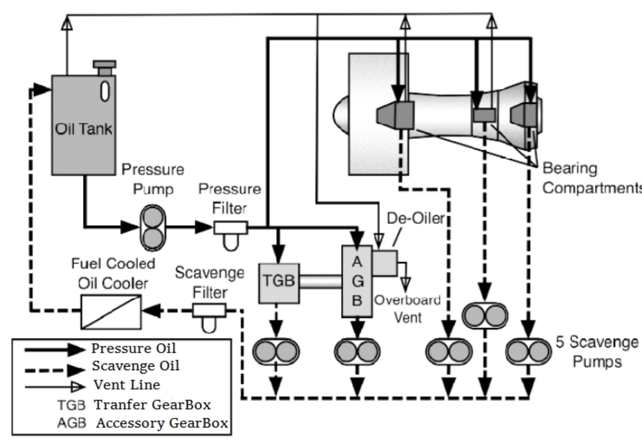


FIGURE 1: A TYPICAL LUBRICATION SYSTEM FOR AN ENGINE

2. THERMAL MANAGEMENT OF FUEL COOLED OIL SUPPLY SYSTEM

Regulation of fuel cooled oil cooler can be performed in several ways. One of them when it is necessary to pump fuel through a heat exchanger in a large amount compared with current fuel consumption of the engine. Thus, some portion of the fuel returns to the fuel supply tank with the help of a controlled fuel bypass behind the heat exchanger. In this case, the excess heat flow is returned to the fuel tank:

$$Q_{exc} = Q\frac{G_{f.bypass}}{G_{f.eng} + G_{f.bypass}} \quad . \tag{1}$$

Here Q - heat removal in heat exchanger, $G_{f.eng}$ - fuel mass flow rate through the engine, $G_{f.bypass}$ - fuel mass flow rate in a bypass line.

The heat balance for the fuel in the supply tank can be described as:

$$C_f m dt = (C_f G_{f.eng} t_0 + Q_{exc} - C_f G_{f.eng} t_f) d\tau \quad , \qquad (2)$$

where C_f - fuel heat capacity, m - fuel mass in supply tank, t_0 - initial fuel temperature in supply tank, t_f - current fuel temperature in supply tank, τ - bypass time, dt - temperature difference.

The accurate estimation of bypass time is crucial for subsonic ($M<1$) and supersonic ($M>1$) aircraft. Thus, during long flights at supersonic speeds, heat is supplied from the aircraft structure to the fuel in the tanks, and the fuel temperature, excluding influence of fuel heating in the section from fuel tanks to pumps, can reach 80 - 120 ºC [2]. It critically influences the thermal stability of the fuel. On the other hand, during long flights at subsonic the fuel temperature can reach negative values. Fuel temperature in the inlet of the engine can be estimated as:

$$T_{eng} = T_{\sup.\tan k} + \Delta T_{pump} + \Delta T_{HEX} + \Delta T_{\zeta} \quad , \qquad (3)$$

where $T_{\sup.\tan k}$ - fuel temperature in supply tank, ΔT_{pump} - fuel heating in the pumps, ΔT_{HEX} - fuel heating in heaters, ΔT_{ζ} - fuel heating in supply pipes. Calculation of each component in equation (3) requires a complex integrated approach in order to accurately solve the thermal task.

A lot of engines contain the fuel-cooled integrated drive generator (IDG), which transfers the additional heat to fuel. At the higher power settings, this leads only to a slightly higher oil temperature. But at power settings at idle the fuel flow is very low and the amount of heat from the IDG is not reduced[3]. The regulation of the hot fuel bypass can be provided by a flow return valve (FRV) where hot fuel is mixed with a cold one and return to the tank. However, this method of regulation imposes strict conditions on the regulation time. Bypass of hot fuel can be performed in case of short-term shortage of fuel cooling resource and increasing of its time become impractical. Increasing the fuel temperature in the tank leads to a number of negative effects. One of the common issues is cavitation in the pump or "vapor-lock" when the liquid fuel could literally boil inside the pump facility. Obliviously, hot fuel usage leads to ineffective cooling of the bearings and gears. Determination of optimal bypass time can predict almost all negative issues during system operation.

Addressing all above issues requires the approach, which can provide the solution of the possible problems at early stage. Usage of thermal-fluid network approach is proposed by authors to handle the thermal management systems in aircraft design, which allows considering complex full integration of fuel and oil systems and estimation of its interaction at each operation mode taking into account the equipment such as pumps, heat exchangers, valves, etc., while analyzing pressure losses and thermal process in the system. The system can be discretized on different levels of abstraction taking into account computational time and result accuracy. The thermal-fluid network approach is acceptable instrument for evaluation both steady-state and transient tasks that can deal with complex process in aviation field.

3. THERMAL-FLUID NETWORK APPROACH

Steady-state and transient analysis for this study is performed using thermal-fluid network approach implemented in AxSTREAM NET™ software developed by SoftInWay Inc [4].

The thermal-fluid network method is used to model fluid and heat flow utilizing one-dimensional abstraction. The main idea behind the method is to represent different sections of a fluid path and different parts of a solid structure as one and zero-dimensional components (see Fig. 2), which then are connected with each other in order to form a thermal-fluid network and to simulate fluid and heat flow through these components.

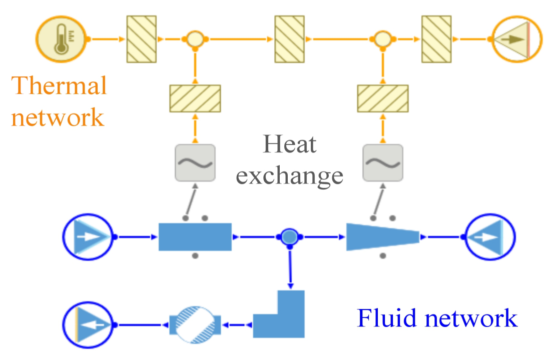


FIGURE 2: EXAMPLE OF THERMAL-FLUID NETWORK

Each component may consist of one or several elements, which can be branches and nodes, as shown in Fig. 3. Branches are used to describe different resistances to fluid and heat flow, while nodes are used to connect branches and set up boundary conditions.

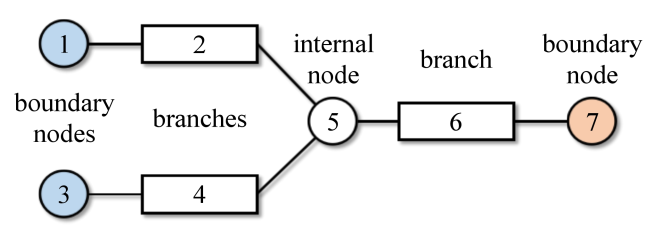


FIGURE 3: NODES AND BRANCHES

Fluid branches utilize momentum equation to calculate mass flow rate through a section of fluid path, which could be written in the next form:

$$\Delta p^* = 0.5\ \zeta\ \frac{G^2}{\rho\ A^2} + \frac{F_{ext}}{A} \tag{4}$$

This equation describes total pressure drop (Δp^*) between inlet and outlet of a branch due to fluid flow resistance and external forces (F_{ext}) for a given section of fluid path, where ζ – resistance coefficient; G – mass flow rate; ρ – density; A – cross-sectional area. Here external forces could be gravity or inertial forces due to rotation effect.

Conservation of mass and total enthalpy equations are used to model fluid flow separation and mixing in internal fluid nodes. They can be written as follows:

$$\frac{\partial}{\partial t}(m) = \underset{branches}{\quad} G_i \tag{5}$$

$$\frac{\partial}{\partial t}\left(m\ h_0 - p\ V\right) = \underset{branches}{\quad} G_i\ h_{0iupstream} + S_e \tag{6}$$

Equation (5) describes rate of fluid mass change in a node due to fluid flows through connected fluid branches, where m – fluid mass in a node; t – time; G_i – mass flow rates through connected fluid branches.

Equation (6) represents rate of energy change in time through the change of total enthalpy and pressure in a fluid node due to total enthalpy transfer from upstream nodes through connected fluid branches and energy source. Here m – fluid mass in a node; h_0 – total enthalpy at a node; p – static pressure at a node; V – volume of a node; t – time; G_i – mass flow rates through connected fluid branches; $h_{0i\ upstream}$ – upstream total enthalpies of connected fluid branches; S_e – energy source (could be heat flow because of convective and radiation heat exchange with solid structure).

Equations (5) and (6) can be complemented with an additional pressures equation in order to model lossless junction using the approach described in [10].

Thermal branches utilize an equation of heat flow due to temperature difference between different places in a solid structure and between a solid surface and a fluid flow, which can be written in the next form:

$$Q = \alpha\ \left(T_1 - T_2\right)\ A \tag{7}$$

In this equation: Q – heat flow through a thermal branch; α – heat transfer coefficient; T_1 and T_2 – temperatures at nodes connected by the thermal branch; A – cross-sectional area of the thermal branch.

The energy conservation equation is used for a thermal node and can be expressed as:

$$\frac{\partial}{\partial t}\left(m\ \ C\ \ T\right) = \underset{branches}{\quad} Q_i + S_q \tag{8}$$

This equation describes rate of energy change in time through the change of temperature (T) within the mass (m) of the substance (solid or fluid) with a specific heat capacity (C) due to the heat flows through connected branches (Q_i) and heat source (S_q).

Time derivatives in the left hand side (LHS) of conservation equations (5), (6) and (8) are approximated using the fully implicit scheme of the first order as follows:

$$\frac{LHS^{new} - LHS^{old}}{\Delta t} = RHS^{new} \tag{9}$$

which gives unconditionally stable solution for any time step (Δt).

Momentum equation (4) and mass conservation equation (5) for all fluid nodes and branches form one system of equations, while energy conservation equations (6 and 8) for fluid and thermal nodes and heat flow equation (7) for thermal branches form another equation system. These two systems are solved using Newton-Raphson method sequentially in iterations while both systems are converged with a given tolerance.

As a result of calculation, the solver computes fluid flow properties (such as pressures, densities, velocities) for fluid components and heat flow properties (such as heat fluxes and temperatures) for thermal components.

Despite implementation differences, all programs for thermal-fluid network modeling deal with the same governing equations, some of which were briefly presented in this section. This equations describe conservation of mass, momentum and energy, and supplemented by the equation of state allow to perform steady state and transient analysis of fluid and heat flows. The usage of these equations in a one-dimensional formulation allows to model complex thermal-fluid systems, and reduce computational time compared with 2D-3D CFD.

4. A CASE STUDY CHALLENGES

The accurate analysis of transient phenomena at fuel cooled oil system allows solving a number of tasks. For example, choice of supply tank construction and capacity is detected by acceptable fuel heating in the tank. The volume changes due to increasing temperature can be described as:

$$V_2 - V_1 = m\ \frac{1}{\rho_2} - \frac{1}{\rho_1} \tag{10}$$

.

Here ρ_1 - density at temperature t_1 and ρ_2 - density at temperature t_2.

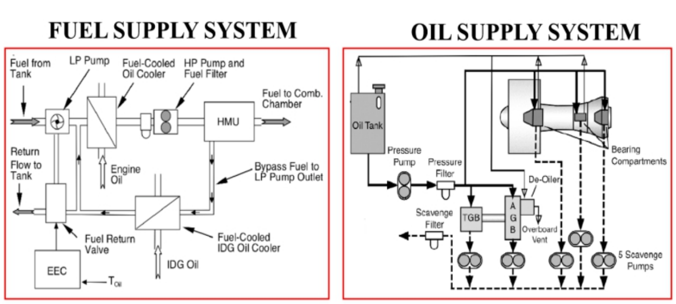
As was noted the heating of the fuel influences its thermal stability. As the main criteria of above is the amount of formed insoluble precipitation per unit mass of fuel, depending on its temperature, which influences the pressure drop in the system. Effects of precipitation leads to faster clogging of filters and additional corrosion of equipment even if fuel temperature reaches 100 - 110 ºC.

Hydraulic pressure drop in the sudden constructions and expansions of the system also increases the fuel temperature ΔT_ζ despite this value is small. However, for unsteady-state analysis including emergency scenario, it has to be taken into account.

The mentioned problems can critically manifest themselves at the low gas mode. During this mode, the amount of cooling fuel abruptly decreases, however, the heat exchange from the engine is still high due to thermal inertia. Moreover, considering emergency situations, when due to some issues the amount of fuel in the tank is lower than necessary, these factors also significantly influence the entire thermal management of the system. Prediction and evaluation of any emergency in transient flight conditions have to be established at the early stage of design.

5. A FUEL COOLED OIL SUPPLY MODEL SIMULATION

The fuel cooled oil supply system (see Fig. 4) contains common components in aircraft engines, which provides the possibility to accurately describe each element of the system and subsystem using thermal-fluid network approach. All components of the system interact with each other. The system contains an oil tank, pump (P), strainers, and bypass valves, jets, bearing components, 9 scavenge pumps (SP), scavenge filter, fuel-oil detailed heat exchanger model, pipes and fittings from the oil supply system side. The bearings units increase the oil temperature at the system depending on heat generation values and working regime. The fuel supply system is presented with a fuel tank, low-pressure fuel pump stage, high-pressure fuel pump stage, filter, IDG oil cooler, and FRV and fuel supply nozzles.



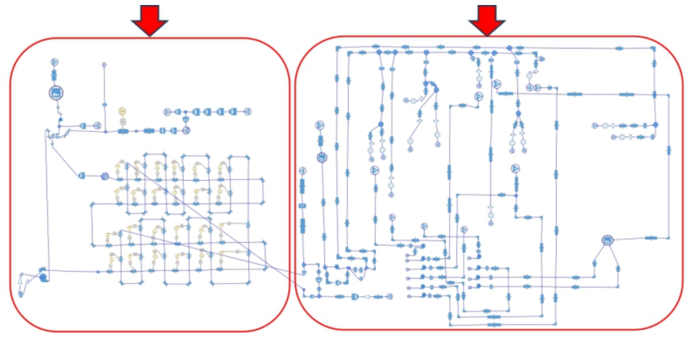


FIGURE 4: A FUEL COOLED OIL SUPPLY SYSTEM MODEL

Oil side. The oil supply path layout is presented in Fig. 5. From the oil tank with a volume of 0.029 m^3 oil goes through the pipes and fittings, supply pump, and strainer to bearing components. Estimation of friction resistance coefficient calculation for channels can be described as:

$$\zeta = \lambda \frac{l}{D_h} \tag{11}$$

Depending on channel type, length and cross-section, Darcy friction factor (λ) is a function from Reynolds number:

$$\lambda = f(\mathrm{Re}) \tag{12}$$

Each pump component contains assigned characteristic that allow automatically calculating the pressure drop and efficiency values during transient solution. The power of pump is calculated as:

$$W = \frac{G \ \Delta P^*}{\rho \ \eta} \tag{13}$$

Depending on mass flow rate value before the pump, the values of total pressure difference and efficiency are picked out from the characteristics table. The determination of intermediate parameters carried out by interpolation methods.

Adding of pumps characteristics is a complex task which requires the full set data for each type of them. The pump components data was simplified. The transient analysis assumed estimation of low power settings mode. Assuming that rotational speed is a constant value in considered time interval, the parameter dependencies from rotational speed can be neglected without losing of physical meaning. The efficiency values also were accepted as a constant value.

The capacities of supply and scavenge pumps are presented in Table 1 and the its layout is presented in Figure 6.

TABLE 1 - OIL SUPPLY AND DISCHARGE PUMPS CAPACITY

Pump	P	SP1	SP2	SP3	SP4	SP5
Capacity, kg/s	0.974	0.990	0.990	0.660	0.990	0.264

Pump	SP6	SP7	SP8	SP9	SP10	SP11
Capacity, kg/s	0.594	0.215	0.479	0.264	0.264	0.594

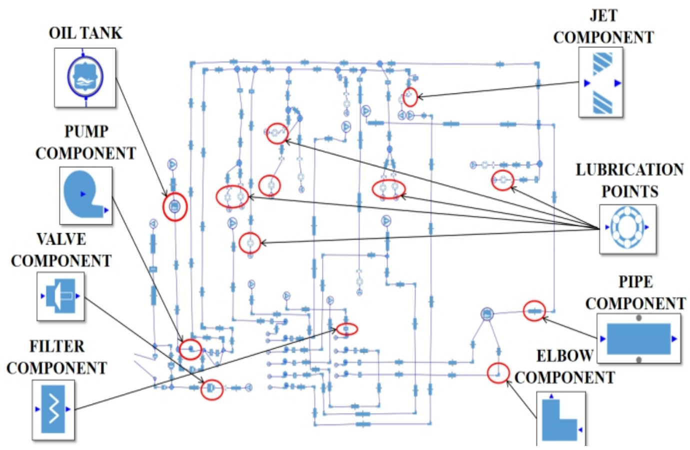


FIGURE 5: AN OIL SUPPLY PATH MODEL LAYOUT

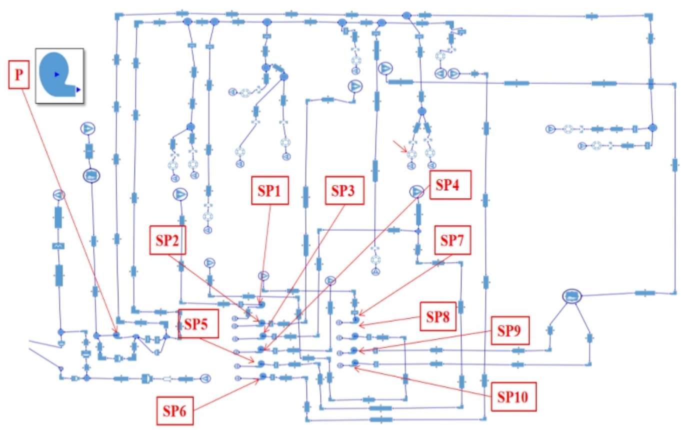


FIGURE 6: OIL PUMPS LAYOUT

The heat generated from each bearings and gears depends on the working regime of the engine and the appropriate load of them. The necessary value of the mass flow rate to each bearing is controlled by calibrated jets and depends on bearings friction power. Friction power is taken into account the bearings and gears configuration, axial and radial loads and its resistance coefficients. The resistance coefficient includes dimensionless correlations and calculated as [5]:

$$\zeta = f(B, D, \mathrm{Re}^x, Eu^y, \mathrm{Pr}^z), \tag{14}$$

where B, D, x, y, z - correlation coefficients. Euler criteria takes into account axial and radial loads on the bearings.

The friction power is calculated as a function from resistance coefficient, bearings geometry data and fluid density at the bearing temperature:

$$W = f(\zeta, GP, \rho), \tag{15}$$

where GP - geometry parameters.

The oil mass flow rate to cool down the bearings and gears to required temperature is calculated from thermal balance equation as:

$$G = \frac{W}{(h_{02} - h_{01})}. \tag{16}$$

The oil mass flow rate values to lubrication points are presented in Table 2 and and the its layout is presented in Figure 7:

TABLE 2 - OIL MASS FLOW RATE TO LUBRICATION POINTS

Lubrication point	1	2	3	4	5	6
Oil MFR, kg/s	0.020	0.041	0.025	0.063	0.036	0.041
Lubrication point	7	8	9	10	11	12
Oil MFR, kg/s	0.071	0.058	0.038	0.028	0.215	0.017

The hot oil passes to scavenge pumps and filters. Then, it is supplied to the IDG, which heats the oil. After IDG, the oil passes to fuel cooled oil cooler. The Nusselt number, which is recommended for heavy liquids and oils [6] to calculate heat transfer coefficient between oil and fuel in fuel cooled oil cooler, were used as:

$$Nu = 0.0118\ \mathrm{Re}^{0.9}\ \mathrm{Pr}^{0.3}, \tag{17}$$

The layout of the oil cooler is presented in Table 3 and Fig. 8.

TABLE 3 - FUEL COOLED OIL COOLER HEAT EXCHANGER LAYOUT

Parameter	Unit	Value
Number of tubes	pcs	840
Tubes wall thickness	mm	0.2
External tube diameter	mm	2
Tubes length	mm	318
Fuel side number of passes	pcs	4
Oil side number of passes	pcs	12

Fuel side. The fuel supply path layout is presented in Fig. 9. From the fuel tank fuel passes to the low-pressure stage of the fuel pump through pipes and fittings. Low-pressure cold fuel passes through the fuel cooled oil cooler and removes the heat from the oil. The heated fuel goes to the high-pressure pump stage and filter and passes to cool the IDG. The required controlled amount of fuel is supplied to the burner depends on the working regime. The excess of fuel can be returned to either the fuel cooled oil cooler if the FRV is closed or only some amount of hot fuel is returned to the oil cooler. The other bypasses to fuel tank through the FRV where hot fuel is mixed with low-pressure cold fuel to increase its temperature.

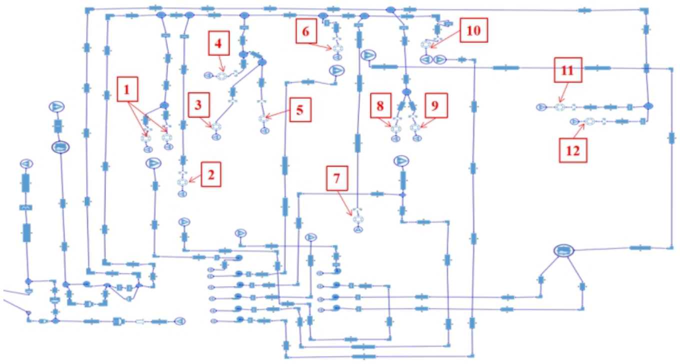


FIGURE 7: LUBRICATION POINTS LAYOUT

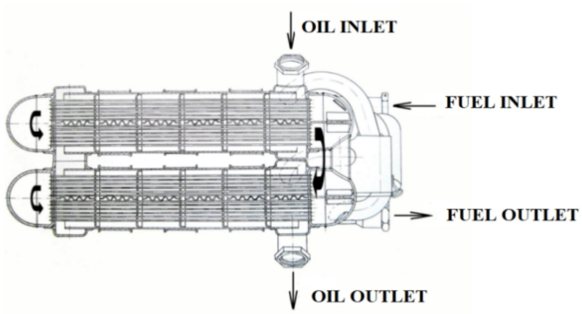


FIGURE 8: A FUEL COOLED OIL COOLER TMT 426TA

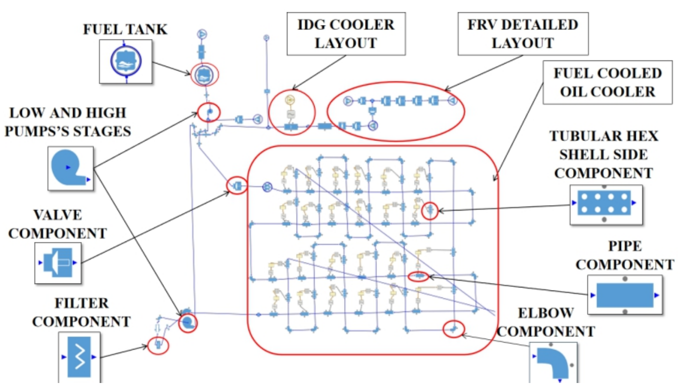


FIGURE 9: A FUEL SUPPLY PATH MODEL LAYOUT

Methodology. The described model includes inlet/outlet boundary conditions such as total and static pressures, mass flow rates and temperatures of fluids. The geometry data as channels cross sections, length, areas, etc. is set up as mandatory data for simulation. Each simulated model includes fluid flow resistance components that provide the possibility to estimate impact the fuel/oil properties in the system, calculate components pressure drop, pressure, temperature and flow rates distribution. Also, model includes thermal components to calculate heat exchange between different parts in the system.

A simplified fuel-oil system is considered, which summarizes data for passenger engines of mid-haul flights [7]. The initial boundary conditions for low gas settings performance were used from calculation of the nominal mode. As the first goal, the system heat task was evaluated depending on the fuel amount in the tank. The second aim was performing transient analysis of FRV failure while the amount of fuel in the supply tank is low. During the changing of the operating mode, the fuel mass flow rate decreases to 0.18 kg/s [8]. The excess fuel mass flow rate is 0.37 kg/s. The considered residual volumes in the fuel tank are 0.8; 0.6; 0.4 m^3. The bearings thermal inertia influence was added to the simulated model as a function of lubrication points heat generation changing in time and depending on shaft rotation speed. Constant decreasing of fuel in the tank are taken into account

Normally, the FRV component works depending on the fuel temperature after IDG. The hot fuel mixes with cold fuel taken after the low-pressure pump stage in proportion 2:3 - at low-flow mode, when the fuel temperature after IDG is less than 131 °C, the value of returned fuel is 0.138 kg/s and in a high-low mode, when the fuel temperature reaches 140 °C the flow rate is about 0.278 kg/s. It should be noticed that at normal conditions FRV has to be closed if the fuel temperature in the tank reaches 52.5 °C. If the fuel consumption in the tank is less than 300 kg, FRV also is shut off.

Boundary conditions. The off-design transient analysis was compared with results of thermal management system working at nominal conditions. Cruise mode under nominal conditions has been estimated. The initial and boundary conditions of system at nominal power settings are described in Table 4 and used for transient simulation. The entire scheme contains constant geometry data. The time interval of 1800 sec. was considered.

TABLE 4 - INITIAL AND BOUNDARY CONDITIONS AT NOMINAL MODE.

Fuel side	Units	Value
Working fluid	-	JetA1
Constant fuel pressure at the tank	bar	2.0
Initial fuel temperature at the tank	°C	30
Constant fuel mass flow rate to the burner	kg/s	0.345
Constant bypass fuel mass flow rate	kg/s	0.6
Initial pressure after first stage pump	bar	≈14.64
Constant pressure after lubrication point	bar	1
Oil side		
Working fluid	-	B3-V
Constant oil tank volume	m^3	0.029
Constant oil pressure at the tank	bar	1.1
Initial oil temperature at the tank	°C	40
Constant oil mass flow rate	kg/s	0.652
Initial oil pressure at the system	bar	≈4.0

As result of nominal mode transient calculation, by the end of estimated period of time, the maximum oil temperature before heat exchanger became close to constant value as 112.5 ºC (see Figure 10). The obtained results of cruise mode transient estimation were taken as initial conditions to further low power setting system behaviour evaluation.

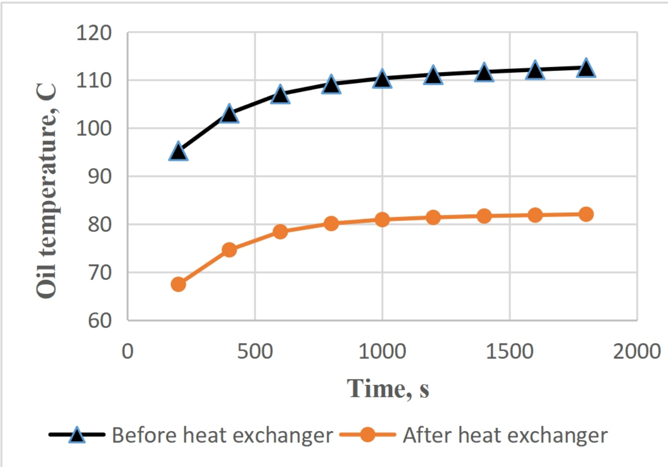


FIGURE 10: VARIATIONS OF OIL TEMPERATURE AT NOMINAL CONDITIONS

During transient analysis of the low setting power, the thermal-hydraulic parameters of the entire system have been estimated. The temperature variation in time at each critical point from the fuel side and oil side were obtained. The fuel temperature values were controlled after IDG, after FRV, after the nozzles, and in the fuel tank. The oil temperature was controlled before the fuel cooled oil cooler, after it, and in the oil tank.

6. SCENARIO AND RESULTS DISCUSSION

The estimation of any emergency situations must be performed at an early stage of design or verification. Considering of fuel lack scenario and failure of FRV can provide a vision of the system behavior and its weaknesses for possible remedial actions in the future.

As was mentioned, the initial conditions were taken from nominal mode calculation. The boundary conditions for low power settings were accepted without changing except of fuel mass flow rate to burner, which is equal 0.16 kg/s and bypass fuel mass flow rate is equal 0.37 kg/s. The friction power of the decreases in time according to the function of dependency from shaft rotation speed. The low power settings mode is analyzed during 300 sec. time interval. The analysis of the system has been performed for different fuel volumes in the tank: 0.8, 0.6. 0.4 m^3.

Thus, analysis of fuel amount influence on the heat utilization shows that the maximum oil temperature at low power settings during calculated time reaches up to 138 ºC (see Fig. 11), which is high value for the system. The analysis shows that at each 0.2 cub. meters of fuel amount decreasing at the tank, the oil temperature after engine increases on roughly 1 ºC. (see Figure 11). The maximum oil temperature after fuel cooled oil cooler is about 109 ºC (see Fig. 12). Obviously, the increasing of the time of working in low power settings is not acceptable in the considering case and will lead to bearings overheating. The oil temperatures after lubrication points distribution are presented in Table 5.

TABLE 5 - OIL TEMPERATURE AFTER LUBRICATION POINTS DISTRIBUTION

Lubrication point	1	2	3	4	5	6
Oil temperature, ºC	136.4	129.8	136.4	136.4	136.4	136.4
Lubrication point	7	8	9	10	11	12
Oil temperature, ºC	136.4	136.4	136.4	129.8	129.8	129.8

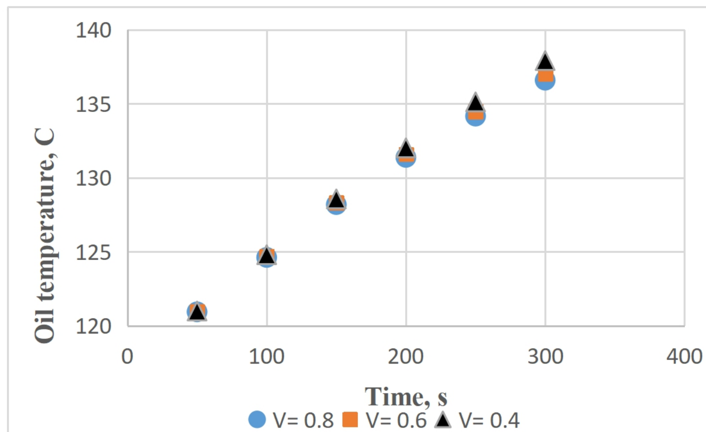


FIGURE 11: VARIATIONS OF OIL TEMPERATURE AFTER ENGINE IN TIME

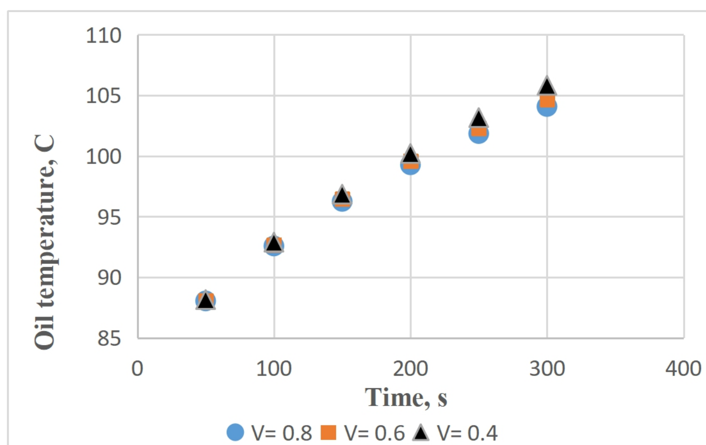


FIGURE 12: VARIATIONS OF OIL TEMPERATURE AFTER FUEL COOLED OIL COOLER IN TIME

During low gas mode the oil temperature in the tank increases in time from 85.0 ºC to 103 ºC. (see Fig. 13), however, the fuel amount does not influence the process.

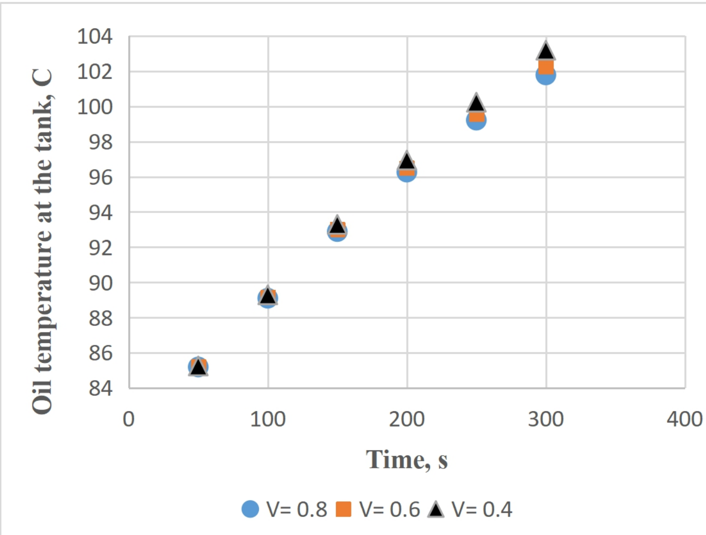


FIGURE 13: VARIATIONS OF OIL TEMPERATURE AT THE TANK IN TIME

Changing of fuel volume in the supply tank shows a non-linear increase of the fuel temperature in the fuel supply tank. The results are presented in Fig. 14.

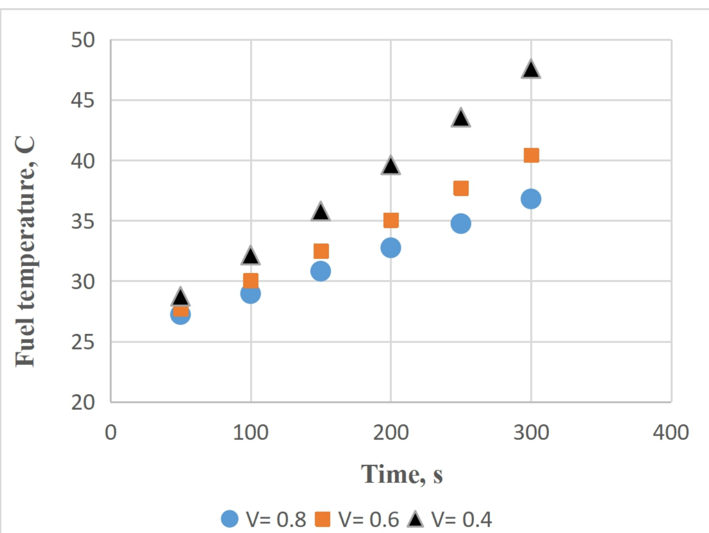


FIGURE 14: VARIATIONS OF FUEL TEMPERATURE AT THE TANK IN TIME

The obtained fuel temperature values before burner do not reach 109 ºC.

FRV failure simulation discussion. The simulation of FRV failure provides an understanding of system behavior. For some reason FRV can start working incorrectly and this could influence the entire fuel-cooled oil system. The FRV contains solenoids, hi/low flow valve, shut-off valve, and orifice. Because of FRV complexity, any component can break down - for example, spring stiffness can be changed after a long time of service or solenoids could stop work. Wang Liwen et all considered the mathematical model of FRV and its failure [8].

The influence of FRV failure from a thermal point of view has been established using transient analysis. The system was analyzed assuming that the fuel amount in the supply tank keeps falling, but when the fuel temperature in the fuel supply tank reaches 52.5 ºC the FRV is considered not to be closed. Thus, the maximum allowable fuel temperature before the burner is about 149-162 ºC. Above this range it leads to a number of negative effects, such as fuel gumming, varnishing or coking, causing fuel nozzles and heat exchanger degradation and fault [9]. The estimation of FRV failure was performed for next fuel volume in tank: 0.2, 0.08, 0.04 m^3.

The analysis showed that the fuel volume 0.2 and 0.08 m^3 leads to oil overheating up to 151 ºC (see Fig. 15). The most stressful picture was obtained for supply fuel volume 0.04 m^3. The oil temperature after the engine reaches critical 160 ºC. The temperature after the fuel cooled heat exchanger reaches up to 144 ºC, which leads oil temperature in the tank to be up to 127 ºC (see Fig. 16).

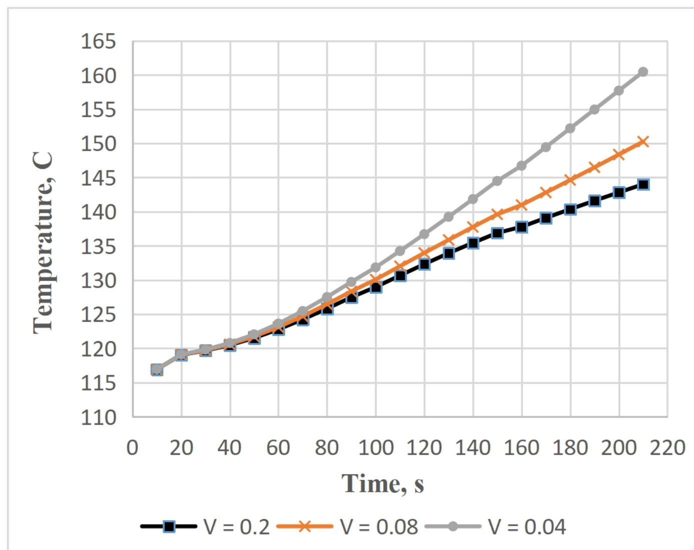


FIGURE 15: FRV FAILURE - VARIATIONS OF OIL TEMPERATURE AFTER ENGINE IN TIME

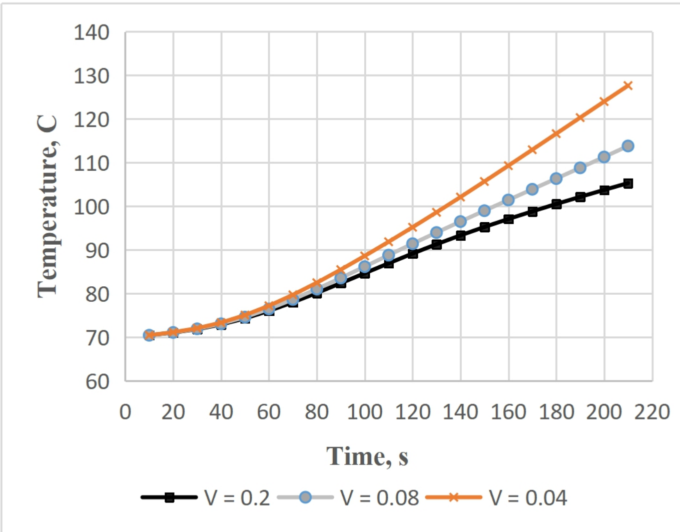


FIGURE 16: FRV FAILURE - VARIATIONS OF OIL TEMPERATURE AT THE TANK IN TIME

7. CONCLUSION

Transition to gliding from a high altitude, when the engine is running at low gas, is the most critical regime for a fuel-cooled oil system, which can lead to overheating not only the oil part - cooling and lubricating units, but also the fuel. The transient analysis of such a regime has been performed using the thermal-fluid network approach. The approach provides the possibility to evaluate the acceptable time of fuel bypass to the supply tank, taking into account emergency situations such as failure of equipment. The transient analysis of the regime can accurately predict the necessary design considerations of the system components depending on the temperature distribution in the system at the early project stages.

Both tasks have been completed - the transient calculation of the thermal management system at low gas power mode and FRV failure were estimated.

The low gas power mode analysis of the system was performed depending on the fuel amount in the supply tank starting from nominal mode results. At the end of the considered time, the oil temperature is high, and weakly depends on fuel amount in the supply tank. However, the fuel temperature values at the tank is changed non-linearly.

The results of FRV failure analysis show that the fuel temperature increasing to critical values in the supply tank that leads to bearings overheating. Failure of FRV can lead to the fuel temperature rise up to 145 ^{0}C at supply tank, while the averaged oil temperature after engine reaches up to 160.4 ^{0}C. All these temperature variations could lead to unacceptable work of the entire engine.

The early stage prediction could estimate possible issues and provides the possibility to take necessary actions and give recommendations to cool the system and organize the heat removing.

In future studies, usage of thermal-fluid network approach can be applied for the analysis of subsonic and supersonic engine systems and it could be applicable to assess the entire flight mission.

REFERENCES

[1] Bich, M. M., Weynberg, E. V., Surnov, D. N. *Lubrication of aircraft gas turbine engines*. Moscow, 1979.

[2] Leschiner, L.B., Uliyanov I. E. *Aircraft fuel system design*. Moscow, 1975.

[3] Andreas Linke-Diesinger. *Systems of Commercial Turbofan Engines*. Springer-Verlag Berlin Heidelberg, 2008.

[4] https://www.softinway.com/software-applications/1d-system-simulation/

[5] Skubachevskiy, G. S. *Aviation gas turbine engines. Design and calculation of details*. Moscow, 1974.

[6] Zysina-Molozhen. *Heat transfer in turbomachinery*. 1974.

[7] Yashovardhan, S. Chati, Hamsa, Balakrishnan "Analysis of Aircraft Fuel Burn And Emissions in the Landing and Take Off cycle using Operational Data", *6th International Conference of Research in Air Transportation* (ICRAT 2014)

[8] Wang, Liwen, Zhang, Lu, and Shi, Xudong "Fault Modeling and Simulation of Aviation Engine Fuel Returen Valve." *The Open Automation and Control System Journal*, 2014.

[9] Gray, C. N., Shayeson, M.W. "Aircraft Fuel Heat Sink Utilization" *Technical Report AFAPL-TR-73-51.* GE, 1973.

[10] Khandrymailov, Andrii, Moroz, Leonid, Yevlakhov, Viktor, Staple, Shanel and Vogel, Gregory. "An Approach to Model a Lossless Junction for Fluid Network Calculations in Turbomachinery." *Proceedings of ASME Turbo Expo 2020*. GT2020-16079. September 23, 2020.

[11] Idelchik, I. E. *Handbook of Hydraulic Resistance*. Begell House, New York (2008).

[12] Yashovardhan, S. Chati, Hamsa, Balakrishnan "Static Modeling of Aircraft Engine Fuel Flow Rate" *30th Congress of the International Council of the Aeronautical Sciences,* September 25-30, 2016.

[13] https://blog.softinway.com/aircraft-fuel-systems/

[14] https://blog.softinway.com/modeling-and-simulating-bearings-bearing-leakages/

[15] Soheil Jafary, Theoklis Nikolaidis. *Thermal Management System For Civil Aircraft Engines: Review, Challenges and Exploring the Future.* 2018.