

Boiling Coolant Vapor Fraction Analysis for Cooling the Hydraulic Retarder

Wei Liu, Gangfeng Tan, Xuexun Guo, Jiafan Li, Yuanqi Gao, and Wei Li
Wuhan University of Technology

ABSTRACT

The hydraulic retarder is the most stabilized auxiliary braking system [1-2] of heavy-duty vehicles. When the hydraulic retarder is working during auxiliary braking, all of the braking energy is transferred into the thermal energy of the transmission medium of the working wheel. Theoretically, the residual heat-sinking capability of the engine could be used to cool down the transmission medium of the hydraulic retarder, in order to ensure the proper functioning of the hydraulic retarder. Never the less, the hydraulic retarder is always placed at the tailing head of the gearbox, far from the engine, long cooling circuits, which increases the risky leakage risk of the transmission medium. What's more, the development trend of heavy load and high speed vehicle directs the significant increase in the thermal load of the hydraulic retarder, which even higher than the engine power. Conventional engine cooling system could not meet the demand of the hydraulic retarder heat rejection within the same installation space.

In this research, independent two-phase evaporator was adopted to strengthen the coolant heat absorption capacity from the transmission medium at the oil outlet of the retarder and increase the thermo stability of heavy-duty vehicles during high working load operating conditions by means of the vacuum flow boiling heat transfer [3,4,5]. As a result, the reliability during driving could be guaranteed.

The basic idea of this study is determining the thermal load characteristics and the transmission medium circular flow by a heavy-duty vehicle operating conditions. Then, the geometric dimensions of the evaporator are confirmed according to the optimal temperature range of the transmission medium. Finally, analysis is conducted for the influence of the coolant flow variation on evaporation system heat transfer power under different operating conditions. The mathematical model of the coolant vapor fraction, the flow and the oil temperature is obtained.

The result shows that under different operating conditions, the designed two-phase evaporation system could stabilize the transmission fluid in the best operating temperature ranges. On the premise of the system's stability, the cooling medium's vapor fraction variation could dynamically respond to the change of vehicle driving conditions. Moreover, this mediates the variation of the recovery flow in the cooling medium and reduces the volume flow significantly. Above all, this improves the controllability of the two-phase evaporation system and decreases energy consumption of this system in the meantime.

CITATION: Liu, W., Tan, G., Guo, X., Li, J. et al., "Boiling Coolant Vapor Fraction Analysis for Cooling the Hydraulic Retarder," *SAE Int. J. Engines* 8(4):2015, doi:10.4271/2015-01-1611.

INTRODUCTION

When the hydraulic retarder is working in the heavy-duty vehicle, all of the braking power is transformed into the thermal energy of the transmission medium. If the temperature of the transmission exceeds 130°C, this will influence the life span of the transmission medium significantly and do harm to the stability of the auxiliary braking system seriously. For the purpose of stabilizing the thermal equilibrium of the hydraulic retarder and reducing the damage leading by thermal fatigue, the transmission medium should be cooled down timely, as the hydraulic retarder will stay stabilized during working. In this research, the schematic of the two-phase evaporation system for the hydraulic retarder is shown in Figure 1. The essence of the hydraulic retarder is a energy transformer device. The resistance moment formed by the transmission medium on the working wheel of the hydraulic retarder could transform the mechanical energy, inputted by the retarder input

shaft, into the kinetic energy of the transmission medium, make frictions between the transmission medium molecules and generates large amount of thermal energy. The hydraulic working characteristic of the retarder could be seen as the combination characteristics of a pump and a heat source [6] which makes the transmission medium circulates in the two-phase evaporator. The evaporator is the heat-exchange device where the heated transmission medium meets the cooling medium. The outlet of heated transmission medium on the hydraulic retarder is conducted to point a in the inlet of the evaporator. In the evaporator, the transmission medium and the cooling medium exchange their heat by countercurrent circulation. The cooling medium is under sub-cooled state 2 in the inlet of the evaporator. The cooled loop water is heated to the saturated liquid state 3 through the preheated zone. Then the cooling medium continues absorbing thermal energy of the transmission medium, and reaches the two-phase state 4. Correspondingly, the heated transmission medium enters the evaporator, then after cooling down the

transmission medium b enter the retarder again and the cooling medium which taking part in the heat exchange enters the next turn circulation again through the return line. In one working condition, the hydraulic retarder will enter a steady working status as the circulation goes on. The characteristics of the hydraulic retarder which matches the evaporator is shown in Table 1. The geometrics dimensions of the evaporator are shown in Table 2, and the total heat-exchange area on the transmission medium side is 16.5m². The geometrics dimensions of the cooling medium are shown in Table 3.

Table 1. Geometrics dimensions of the hydraulic retarder and the transmission medium

Parameter Item	Value	Unit
Frictional resistance loss coefficient	0.22	/
Impact pressure loss coefficient	1	/
Radius of the Central axial flow line in the entrance of the impeller	0.08	m
Radius of the Central axial flow line in the outlet of the impeller	0.16	m
Vane incline angle inner of the wheel	140	°
Vane incline angle outlet of the wheel	40	°
Density of the transmission medium	890	Kg/m ³
Specific heat capacity of the transmission medium	1800	J/(kg*°C)
The most optimal temperature of the transmission medium	80 to 120	°C

Table 2. Geometric dimensions of the evaporator

Parameter Item	Value	Unit
Heat transfer area inner the tube	3.5	m ²
No fin area outer the tube	3.5	m ²
Fin area outer the tube	13	m ²
Total area outer the tube	16.5	m ²
Thickness of the partitions of the transmission medium side	4	mm
Fin height of the transmission medium side	56	mm
Fin spacing of the transmission medium side	56	mm
Fin thickness	4	mm
Fin height	27	mm
layout	Staggered arrangement	/
Pipe material	Stainless steel	/
Fin material	Stainless steel	/

Table 3. Geometric dimensions of the coolant

Parameter Item	Value	Unit
Temperature at point 2	80	°C
Temperature at point 3	81	°C
Temperature at point 4	81	°C
Boiling temperature	81	°C
Pressure of the coolant	49.3673	kPa

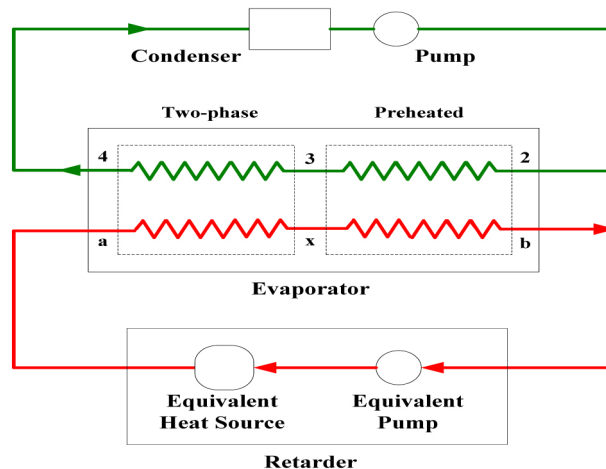


Figure 1. Schematic of the two-phase evaporator system

MATHEMATICAL MODEL

By studying the characteristics of thermal energy produced by the transmission medium in the hydraulic retarder [7] along with the change of the speed of the heavy-duty vehicle, the transmission medium heat generating model and the evaporator heat-transfer model could be set up. In the evaporator, the cooling medium is divided into two sections, the preheated zone and two-phase zone. Therefore, the evaporator is also divided into two sections as the heat transfer model setting up. For each section of tube side and shell side fluid thermal state, the corresponding thermal physical property values could be calculated, based on which the relationship between two-phase cooling evaporation system's heat transfer performance and the cooling medium's vapor fraction at point 4 could be analyzed. Then according to the geometric dimensions of evaporator, respectively using the heat-transfer correlations, the corresponding convective heat-transfer coefficient could be worked out. Furthermore, each section's heat transfer performance could be also determined. Finally, the analysis of the characteristics of heat transfer energy with the change of the cooling medium's vapor fraction under different driving conditions.

Transmission medium heat generating and flow model

Center line of the transmission medium runner needs to be determined firstly for analyzing the hydraulic retarder braking performance [8,9,10,11]. Inlet and outlet of the hydraulic retarder is defined as the intersection of center line of the runner and inlet and outlet of the hydraulic retarder blades. Figure 2. shows the schematic diagram of the hydraulic retarder rotor's center line of the runner.

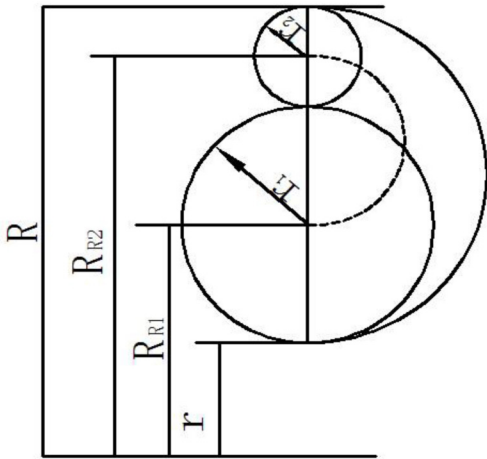


Figure 2. The hydraulic retarder rotor's center line of flow path

$$\left\{ \begin{array}{l} R_{R1} = \frac{r}{2} + \frac{\sqrt{2R^2 + 2r^2}}{4} \\ R_{R2} = \frac{R}{2} + \frac{\sqrt{2R^2 + 2r^2}}{4} \\ r_1 = \frac{-r}{2} + \frac{\sqrt{2R^2 + 2r^2}}{4} \\ r_2 = \frac{R}{2} - \frac{\sqrt{2R^2 + 2r^2}}{4} \end{array} \right.$$

In the study of the performance of the hydraulic retarder, for convenience, the relevant parameters of the rotor are indicated by the subscript R while the relevant parameters of the stator are indicated by the subscript r. For the rotor, working liquid in the inlet and outlet velocity triangle could be shown in Figure 3. According to the theorem of moment of momentum, the torque rotor role in working liquid could be worked out.

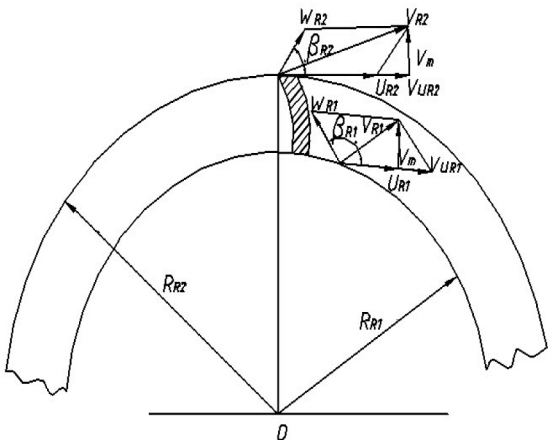


Figure 3. The transmission medium speed relations

According to the conservation of energy, the energy rotor on the working fluid is equal to the energy produced by the working fluid on the stator with all kinds of energy loss during the working fluid movement.

$$\begin{aligned} & \frac{\omega_R R_{R2}}{g} (\omega_R R_{R2} + v_m \cot \beta_{R2}) - \frac{\xi_{cj}}{2g} (\omega_R R_{R1} + v_m \cot \beta_{R1})^2 \\ & - \frac{\xi_{cj}}{2g} (\omega_R R_{R2} + v_m \cot \beta_{R2})^2 - \xi_{mc} \frac{v_m^2}{2g} = 0 \end{aligned}$$

According to this equation, After knowing R_{R1} , R_{R2} , $\cot \beta_{R1}$, $\cot \beta_{R2}$, ξ_{mc} , and ξ_{cj} from the design of the hydraulic retarder, the speed of certain angular velocity could be obtained.

For the flow section area:

$$A_m = 4\pi R_{R1} r_1$$

For the flow section volume flow:

$$Q = v_m A_m$$

For the hydraulic retarder braking torque under the condition of full of the working fluid:

$$T_R = \rho Q (v_{uR2} R_{R2} - v_{uR1} R_{R1})$$

If η represents filling ratio, under different η , the hydraulic retarder braking torque is:

$$T_R = \eta \rho Q (v_{uR2} R_{R2} - v_{uR1} R_{R1})$$

The evaporator heat transfer model

When the hydraulic retarder working under a stable operating condition, the temperature at state a, the inlet of the evaporator, and the temperature at state b, the outlet of the evaporator is known, the specific enthalpy of transmission medium at state a and b could be calculated. Then, the overall heat transfer rate in the heat exchanger is determined by:

$$Q = m_{oil} (h_a - h_b) = m_{water} (h_4 - h_2)$$

The thermodynamic properties of the transmission oil at state x could be calculated according to the energy equation for each zone. Then the temperature of the oil at state x could also be obtained. Hence, the heat transfer rates of all zones are obtained:

$$Q_{ep} = m_{oil} (h_a - h_x) = m_{water} (h_4 - h_3)$$

$$Q_{ph} = m_{oil} (h_x - h_b) = m_{water} (h_3 - h_2)$$

Considering the transmission medium at the crust side of the evaporator, the convective heat-transfer coefficient could be calculated with the Gnielinski correlation[12]: