침식마모 손상된 차량용 워터펌프의 성능저하 연구

전승원 · 박찬성[†*] · 김윤호 · 이규정
고려대학교 기계공학과 · *국립과학수사연구소중부분소
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Study on the Performance Deterioration of Erosion-corrosion Damaged Automotive Water Pump

Seung-Won Jeon · Chan-Seong Park[†*] · Yoon-Ho Kim · Kyu-Jung Lee
Department of Mechanical Engineering, Korea University
*Department of Physical Engineering, Central District Office of National Institute of Scientific Investigation
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Abstract : A flow analysis for the erosion-corrosion damaged automotive water pump which causes vehicle fire is numerically performed using the CFX program, computational fluid dynamics (CFD) code. The blade bending deformation and the blade clearance enlargement are considered in the analysis of performance reduction. For the cavitation analysis, the homogeneous multiphase model is adopted using the Ralyleigh-Plesset model for the rate equation controlling vapor generation and condensation.

초 록 : 침식마모로 인하여 손상되어 차량화재를 유발한 차량용 워터펌프의 성능저하를 CFX프로그램을 이용하여 해석하였다. 손상에 따른 성능저하를 해석하기 위하여, 블레이드의 굽힘변형과 간극의 증가를 고려하였다. 캐비테이션 해석 시에는 포화증기 발생과 응축 과정을 모델링하기 위하여 Rayleigh-Plesset 모델을 적용하였다.
Key Words : flow analysis, performance reduction, water pump, cavitation, CFX

1. Introduction

It is reported that the number of vehicle fires in 2006 was 5,929 and the rate of occurrence of vehicle fires is continuously increasing[1]. For vehicle fires caused by malfunction, the case of engine overheating is noticeable because sometimes it leaves characteristic traces. The main causes of engine overheating which results in a vehicle fire are classified in two categories. The first is overheating of the exhaust system which happens when the driver in a parked vehicle continuously accelerates the engine. The second is the malfunction of the engine cooling water system[2,3]. It is reported that when the unburned fuel flows into the operating catalytic converter, afterburning can happen and this causes the melting of the ceramic monolith. This phenomenon happens when the misfire rate increases and the ignition timing is retarded[4]. Recently a closed-coupled catalyst(CCC) has been commonly used in the exhaust system with an underfloor catalyst(UCC) to reduce the engine cold-start emissions by utilizing the energy of the exhaust gas. Unfortunately this makes a vehicle fire possible by exhaust overheating because of the enclosed location of the CCC in the exhaust manifold with poor heat exchange conditions with ambient air. Lee[5] studied the exhaust heat exchanger which is installed between the exhaust manifold and the CCC to decrease the exhaust gas temperature under high load conditions. It is reported that the CCC temperature is successfully controlled under the high load condition but deterioration of exhaust emission happens at cold start due to delay of catalytic light-off time. Fig. 1 and 2 show typical shape of burned vehicle caused by continuous acceleration and the thermally damaged CCC of a burned vehicle[2].

† To whom correspondence should be addressed.
color@nisi.go.kr


Fig. 1. The typical shape of vehicle fire caused by exhaust overheating(3000cc class, V6 SI eng.).

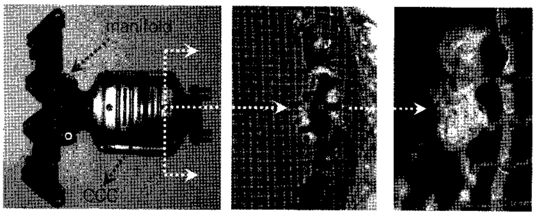
Fig. 2. The shape of thermally damaged CCC which caused vehicle fire.

The second main cause of engine overheating which results in vehicle fire is the malfunction of engine cooling water system itself. Water pump plays the key role in the cooling system, which rejects approximately 30% of the heat of combustion to ambient air, so the malfunction of the water pump directly affects engine overheating which can lead to a vehicle fire[6]. The main cause of water pump malfunction is the erosion-corrosion damage which is accompanied by cavitation. Cavitation normally occurs at low pressure regions of rapidly vibrating liquid/metal interfaces. Where the local pressure in a liquid is reduced below the vapor pressure of the liquid without temperature change, a condition may be reached where bubbles/cavities start and grow within the liquid[7]. Damage is caused when the bubbles make contact with the surface and collapse due to a localized increase in the pressure. It is reported that repeated implosions of bubbles are sufficient to cause cavitation at the surface, especially if corrosion products are present in the working fluid[8]. Cavitation erosion with corrosion accelerates the rate of attack because of the relative movement between the fluid and the metal surface. Generally, this movement is quite rapid so mechanical wear from abrasion can be involved. Material is removed from the component's surface as dissolved ions, or in the form of solid corrosion products and accelerate the erosion-corrosion damage[8]. This present study focuses on the analysis of performance deterioration of the automotive water pump which can cause a vehicle fire. The characteristics of suction performance of a prototype pump are developed by considering a multiphase homogeneous model. The damaged impeller is modeled by considering the bending of the blade and the clearance enlargement between the blade tip and the casing, independently. From the results of numerous CFD analyses, performance reduction is predicted by considering an appropriate exponent for similitude.

2. Numerical analysis

In general, a low specific speed centrifugal pump without shroud is used for automotive water pump based on economics. Fig. 3 shows typical erosion-corrosion damaged impeller of automotive water pump which is the focus of this study. The blades become thinner as a result of severe erosion. As the blade becomes thinner, the bending of the blade and the clearance between the blade and the casing is increased. For the analysis of performance reduction, the bending deformation of blade and the clearance enlargement between the blade tip and the casing are considered. For blade bending deformation, bending angles of 5° and 15° are considered. A blade clearance increase of 3mm and 4mm is considered. Because of the asymmetry of the volute geometry, the full geometry of the impeller with volute is modeled and evaluated. The discharge of the water pump is directly connected to the engine block so the pump discharge is assumed as an extension of the volute exit. Water at 80℃ is considered a working fluid. Unstructured meshes of 470 thousand nodes are used based on preliminary mesh tests. Fig. 4 shows solid body configurations of prototype with computational grid. Steady three dimensional turbulent flow with the standard k-ε model is considered for the analysis of performance reduction. For the cavitation analysis, the homogeneous multiphase model is adopted using the Ralyleigh-Plesset model for the rate equation controlling vapor generation and condensation. Pressure boundary condition is used for pump outlet and flow rate for pump inlet. The MFR(Multiple Frame of Reference) and frozen rotor methods are used to consider the rotational effect of impeller[9].

Fig. 3. Comparison of erosion–corrosion damaged automotive water pump with prototype(2000cc class, SI eng.).

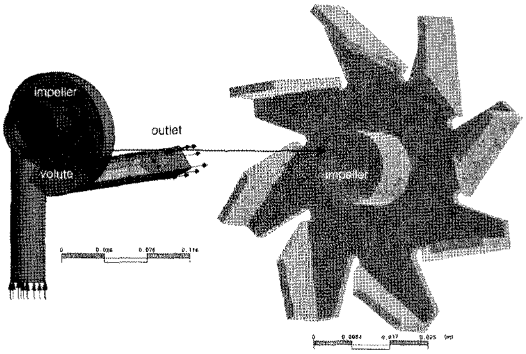

Fig. 4. Solid body configuration and mesh of prototype impeller.

3. Results and discussion

The calculated results are compared with experimental data offered by a pump manufacturer(Fig. 5). The calculated results are in good agreement with the experiment data at 3000rpm, but have deviations of 10% or less at 6000rpm. It can be concluded that the prototype of the pump is appropriately modeled regardless of some deviations because the gradient of calculated performance curve at 6000rpm is similar to that of experimental data. Fig. 6 shows the calculated suction performance of prototype pump. In general, NPSHr(Net Positive Suction Head required) is obtained experimentally by considering the operation point where TDH(Total Difference Head) drops to 3%, as the suction pressure of pump gradually reduces with constant flow rate[10]. The NPSH is defined as follows.

$$NPSHa = \frac{p_T - p_v}{\gamma} \tag{1}$$

where p_T is the total pressure at pump inlet, p_v is the vapor pressure of water and γ is the specific weight of water.

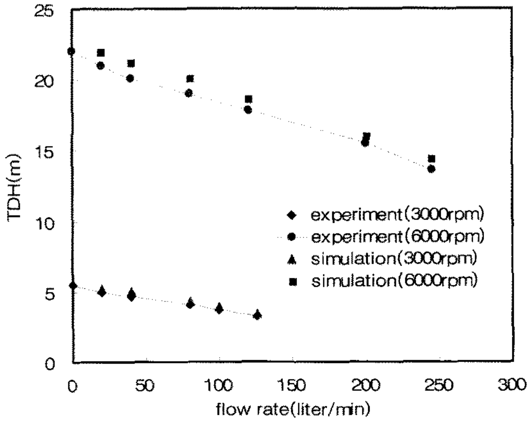

Fig. 5. Pump performance comparison with experimental data.

In Fig. 6, the predicted NPSH available(NPSHa) is calculated based on the vapor pressure of water at 80℃ with the flow rates of 80liter/min and 126 liter/min. It shows that the calculated NPSHr is approximately 1.0m or less without a distinct difference between the given flow rates. The Hydraulic Institute(HI) has suggested the empirical correlation of NPSHr in the form of equation (2) from compiled experimental data.

$$NPSHr = 0.415 \times 10^{-6} \times N_s^{5/3} \times TDH \tag{2}$$

where N_s = specific speed(gpm, rpm, ft).

Using the experimental data given for 3000rpm at 80liter/min with a TDH of 4m, the NPSHr is cal-

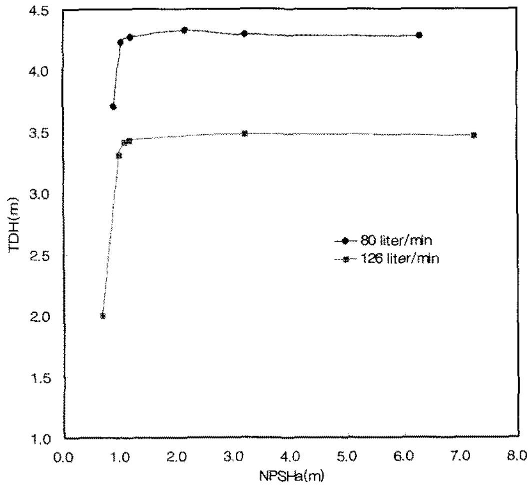

Fig. 6. Suction performance prediction(3000rpm).

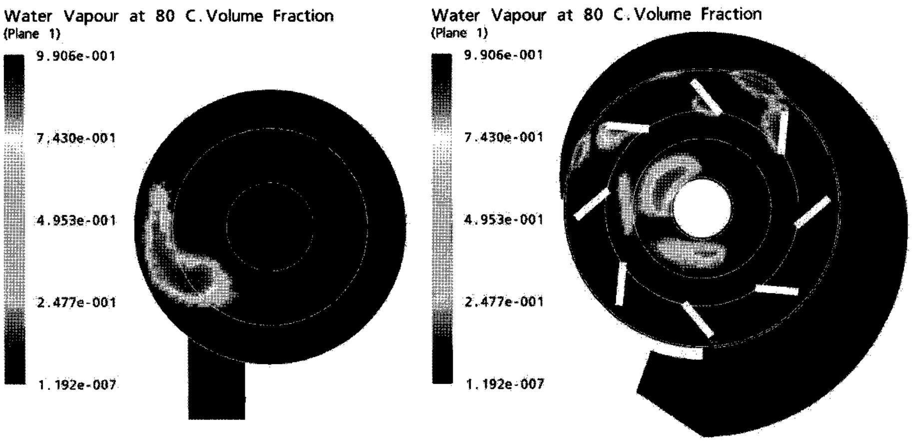

Fig. 7. Volume fraction calculation(80liter/min, exit pressure of 10^5pas, 3000rpm).

culated as 0.5m from equation (2). It is approximately 50% less than the predicted result of 1.0m from Fig. 6. This can be explained by considering the volume fraction at the pump inlet. Fig. 7 shows the predicted volume fraction of water vapor at the pump inlet(left hand side) and at the blade(right hand side). The water vapor is formed at the suction side of the blade but there is a peculiar location at the pump inlet where water vapor forms. The extended impeller shaft with bearing housing is installed at the entrance of pump which considerably reduces the area of the flow passage. This results in higher velocity head which causes the lower static pressure at a given flow rate. From these results, it can be concluded that the NPSHr of the prototype pump is higher than that determined by equation (2), regardless of the shape of the impeller, because of the poorly shaped flow passage at the entrance. This can aggravate the erosion-corrosion damage of the impeller.

Fig. 8 and 9 show the predicted performance reduction of a damaged pump assuming that the clearance is enlarged by erosion-corrosion. The prototype pump has a clearance of 1mm between the blade tip and the casing. As the clearance is increased to 3mm and 4 mm, the performance is reduced considerably. In general, the unknown pump performance can be predicted by using the performance data of a known pump with an appropriately selected exponent n when conditions of similitude are met. Equation (3) is the form for similitude between the prototype and the model with exponent n[10].

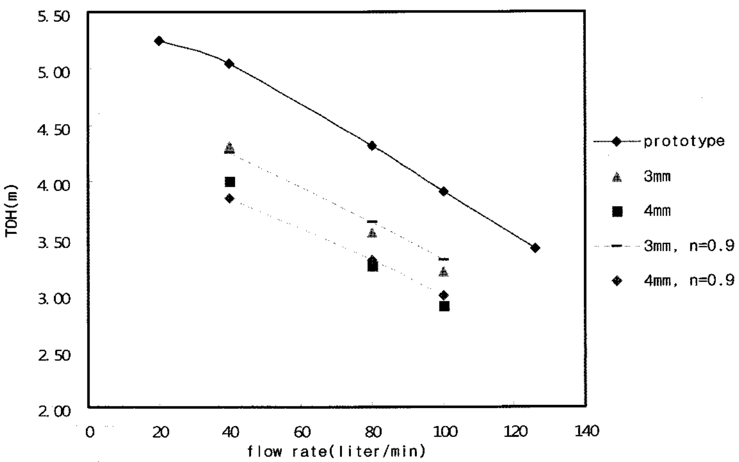

Fig. 8. Performance reduction by clearance enlargement (3000rpm).

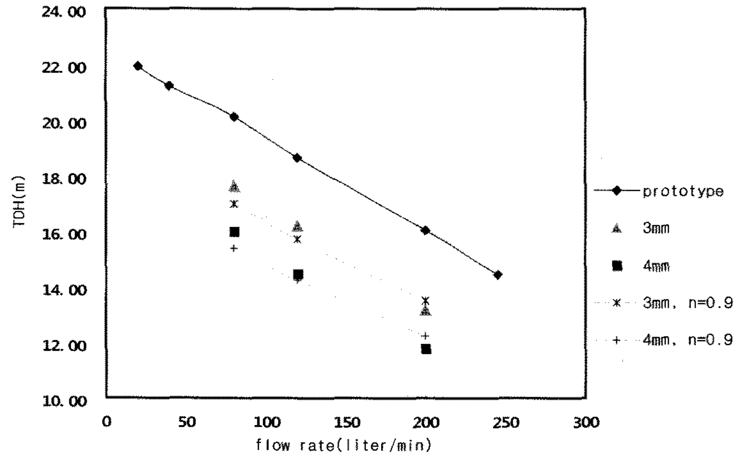

Fig. 9. Performance reduction by clearance enlargement (6000rpm).

$$H_m = H_p\left(\frac{h_m}{h_p}\right)^n = H_p(r)^n \tag{3}$$

where $h = \dfrac{b_1 + b_2}{2}$ for the clearance increase and h = deformed angle for the blade bending deformation, H_m = TDH of model pump, H_p = TDH of prototype pump, b_1 = blade width at impeller inlet, b_2 = blade width at impeller exit, r = geometric ratio, and n = similitude exponent.

When the conditions of similitude are met exactly, the exponent n corresponds to -2.0. By assuming an exponent n of 0.9, the clearance increase is approximately predicted for both 3000rpm and 6000rpm. The predicted results have a steeper gradient than that of the similitude rule with n = 0.9 with the deviation less than 10%. Fig. 10 and 11 show the predicted performance reduction of damaged pump assuming that blades are equally bent by reduction of strength caused by erosion-corrosion. When the deformed angle is less than 5°, the performance reduction can be predicted by assuming a similitude rule exponent n of 0.3. In this case, geometric ratio r is defined by the deformed blade angle divided by blade angle of prototype impeller. As the deformed angle increases, the performance deviates considerably from the similitude pattern and the performance reduction can not be predicted by the similitude rule.

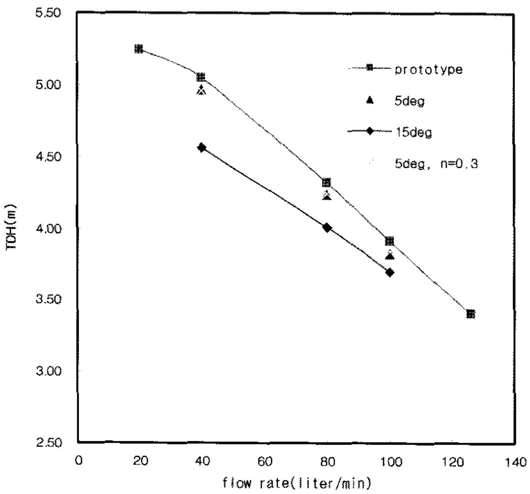

Fig. 10. Performance reduction by blade bending deformation (3000rpm).

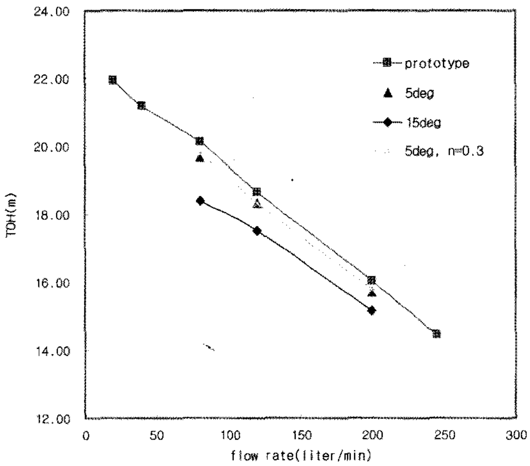

Fig. 11. Performance reduction by blade bending deformation (6000rpm).

4. Conclusions

In this study, a flow analysis of an erosion-corrosion damaged automotive water pump which can cause a vehicle fire is numerically performed using the CFX program, a computational fluid dynamics(CFD) code. The prototype pump shows poor suction performance which is not related to impeller geometry itself but directly related to poor suction geometry. The performance reduction of a damaged pump, assuming that the clearance is enlarged by erosion-corrosion, can be predicted using the similitude rule with an exponent of 0.9. The performance reduction of a damaged pump, assuming that blades are equally bent by reduction of strength caused by erosion-corrosion, can be approximately predicted using similitude rule with exponent of 0.3 when the deformed angle is less than 5°. From these results, it can be concluded that the performance reduction of a pump which is not severely damaged could be predicted simply by appropriately selecting an exponent n for the similitude rule. It can be deduced that when numerous experimental data is given, the damaged state of a pump could be predicted by examining the pump operating point.

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