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발전기 여자제어 시스템 교체를 위한 축계진동 해석 및 설계

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Rotorting System Design and Analysis for Exciter Retrofit

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ABSTRACT

발전기에 직류 전압을 인가하는 여자제어 시스템(exciter)은 여자 방식에 따라 간접 여자 방식과 직접 여자 방식으로 분류할 수 있다. 그러나 여자제어 시스템의 세계적인 추세는 직접 여자 방식을 채택하고 있으며, 기존의 설치된 간접 여자 방식도 직접 여자 방식으로의 교체를 추진하고 있는 실정이다. 본 논문에는 500 MW 발전기의 간접 여자제어 시스템을 직접 여자제어 시스템으로의 교체를 위해 수행된 축계 진동해석과 설계 검토 사례를 소개코자 한다. Steady 베어링의 설치 여부를 위한 설계 기준은 축계 진동해석 결과에 바탕을 두고 있으며 steady 베어링의 선정에는 안정성을 이유로 tilting pad 베어링을 선택하였다. 해석 결과를 바탕으로 실제 축계를 설계, 제작하여 운전한 결과 양호한 결과를 얻을 수 있었다.

1. INTRODUCTION

Generator excitation systems have been used for supplying the field currents to the generators. Many years ago most of large capability of generators required alternators for excitation system. Such excitation systems were so big, very complex and expensive. Nowadays, new excitation system, which does not require an alternator any more, is developed since the improved design technology in the generator excitation system. Advantages of new one are compactness, efficiency, convenience, and availability. Therefore, new excitation systems have been replacing existing old ones.

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The rotor dynamic analysis is the most important item to increase of the reliability of rotating machinery [1-5]. The dynamic characteristics of large rotating machinery are heavily dependent not only on the rotor itself, but also on the supporting conditions, such as bearing stiffness and damping, bearing bracket and foundation. In particular, the bearing stiffness and damping coefficients have a large influence on the rotor dynamic characteristics [2,3]. Therefore, in order to replace old excitation systems with new ones successfully, designers should consider carefully dynamic characteristics of rotor-bearing system form view points of how to eliminate the existing alternator rotor, whether or not a steady bearing to be necessary, and what type of a steady bearing to be optimum, etc. A steady bearing is an additional bearing required for supporting the end of collector shaft. The installation of the steady bearing is determined according to the results of rotor dynamic analysis.

This study describes an experience gained from the retrofit of generator excitation system, from with an alternator to without an alternator, for a 500 MW two-pole generator that runs at 3,600 rpm. Rotor dynamic characteristics, such as mode shape, critical speed, and unbalance response, were simulated with or without a steady bearing, to design an appropriate modified rotating system. Both the bearing stiffness and damping coefficients were calculated. In addition, verification testing of field test was carried out to confirm analytical results

2. RETROFITS AND DESIGN CRITERIA

Figure 1 shows the existing generator excitation system with an additional generator, an alternator, which is coupled to the end of the generator rotor. In order to change the excitation system from the old one to the new one, first of all, the alternator rotor should be eliminated. And the next must be considered how to modify the collector rotor system. Figure 2 shows the schematic view of modified rotor system with a steady bearing and without a steady bearing, respectively. As mentioned above, the installation of the steady bearing is determined according to the results of rotor dynamic analysis since the dynamic characteristics of the rotating system are highly dependent on the bearing stiffness and damping coefficients.

In the present study, three kinds of design criteria were considered to decide the acceptability of the modified collector rotor system both statically and dynamically. The first criterion is based on the static stiffness of collector rotor overhang. If the stiffness is less than the design limit, a steady bearing should be installed at the end of collector rotor. This design limit was established several years ago and so reluctant since that excessive vibration problems were experienced on the collector rotor designed by this rule. The second criterion is to evaluate the sensitivity of generator rotor near the resonance of the system. Shirake and Kanki [4] proposed this criterion based on the operating experience and theoretical analysis for large turbine generator sets. The variable used to describe resonance sensitivity to unbalance is the Quality factor of resonance or Q-factor. This factor is commonly used to quantify the damping in order to predict the forced response and to evaluate system susceptibility due to forced excitation at a resonance. Thus Q-factor is determined based on the shape of response curve. A high damped response has a broad peak with low response value and yields a low Q-factor. While resonance with low damping has a sharp peak and yields a high Q-factor. Q-factor is determined at a particular resonance defined as $Q= f_p/(f_2-f_1)$; where f_p is frequency at the resonance, f_1 is frequency less than f_p where the amplitude is 0.707 times the amplitude at f_p, and f_2 is frequency greater than f_p where the amplitude is 0.707 times the amplitude at f_p. Figure 3 shows an example of acceptable design criteria of Q-factor as a function of the ratio of critical speed to operating speed. This criterion indicates that design is acceptable when the Q-factor is below the limit line. The third criterion is that the vibration amplitude evaluated at the operating speed should be less than the limit value. The accurate vibration amplitude could be predicted because of improved rotor dynamics computer programs.

3. ANALYSIS AND MEASUREMENTS

There exist two requirements, i.e. theoretical analysis and measurement, for the successful retrofit. Theoretical analysis was performed to make a verification of design criteria mentioned above. First of all, the shaft stiffness was calculated for the collector rotor overhang, as shown in Fig.2(a). It was found that the shaft stiffness was lower than the limit value so that a steady bearing was required. Rotor dynamic analysis was investigated by FEM based on Timoshenko beam theory [1]. Figure 4 shows a vibration mode of the generator rotor system without a steady bearing at 3,758 rpm, which is the closest natural frequency to the operating speed (3,600 rpm). It can be seen that there exists a natural frequency near the operating speed and overhang mode occurs. Figure 5 shows the unbalance response mode at 3,600 rpm. In order to the unbalance response, it is assumed that 1,153 g-cm (1 pound-in) unbalance weight places at the end of the collector rotor. It can be seen that the excessive vibration amplitude as large as 1, 271 μm peak to peak, which is much larger than the design limit, occurs at the collector end. The Q-factor at 3,758 rpm is about 8.8, which is higher than the design limit. Therefore, it can be concluded that a steady bearing should be installed as shown in Fig.2(b).

In order to select an adequate steady bearing, three kinds of bearing types, cylindrical, elliptical, and tilting pad, were considered. It was found that any kind of bearing hardly changed natural frequency near the operating speed but both the unbalance response and Q-factor were reduced drastically because of the steady bearing. However, both the cylindrical and elliptical bearing have a possibility of oil whirl and/or oil whip occurrence since the actual load acting on the bearing is so small [2,5]. In consequence, a six-pad tilting pad journal bearing was selected as a suitable steady bearing. Figure 6 shows the schematic view of the collector rotor system with a steady bearing, which is finally modified for the retrofit. Figures 7 and 8 show the vibration mode and the unbalance response, respectively. The modified system might be worried about that the collector rotor would run near the resonance whose speed is about 3,715 rpm. However, because of the damping effect of the steady bearing, both the expected maximum unbalance response and Q-factor are very small; which are about 11 μm peak to peak at 3,600 rpm and 1.78, respectively. Since these values are much smaller than the design limits, it is assured that modified rotating system would be run successfully during the operation.

Finally, verification testing of field test was carried out to confirm analytical results. Figure 9 shows the vibration amplitude (peak to peak) measured at the collector end as function of the rotational speed with or without a steady bearing. As sown in this figure, the vibration amplitude of one without a steady bearing increases sharply as the speed and results in impossible in operation over 2,500 rpm. On the other hand, the vibration amplitude of one with a steady bearing is very good in the whole range of operation. The measured vibration amplitude at the operating speed is as low as 30 μm peak to peak but is larger than the expected value by analysis. This reason is caused by the uncertainties such as unbalance weight and misalignment, etc. It can be concluded that test results show a good agreement with theoretical results. Consequently, the retrofit of generator excitation system from with an alternator to without an alternator, for a 500 MW two-pole generator, was achieved successfully. Figure 10 shows a photograph of newly modified collector rotor system.

4. CONCLUSIONS

This case study was gained from the successful retrofit experience of generator excitation system from with an alternator to without an alternator, for a 500 MW two-pole generator. In order to modify and to design the collector rotor system, static and dynamic characteristics, such as shaft stiffness, natural frequency, mode shape, Q-factor, and unbalance response, were simulated by the analysis. Field test results show a good agreement with theoretical results.

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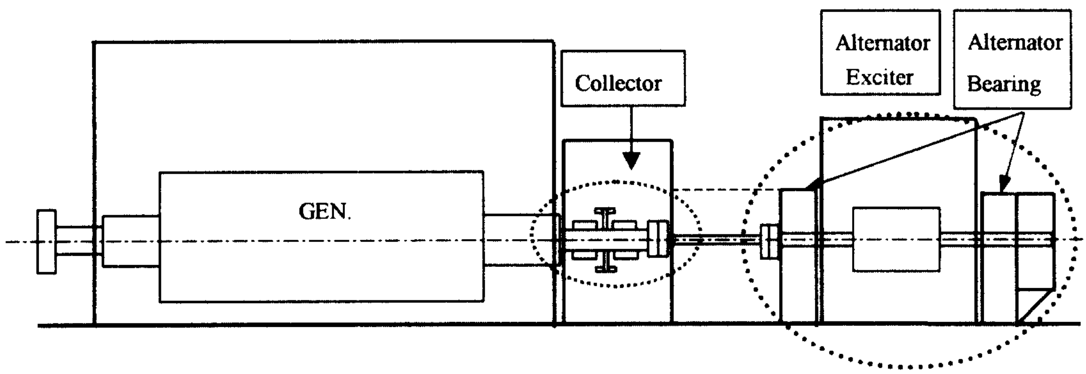
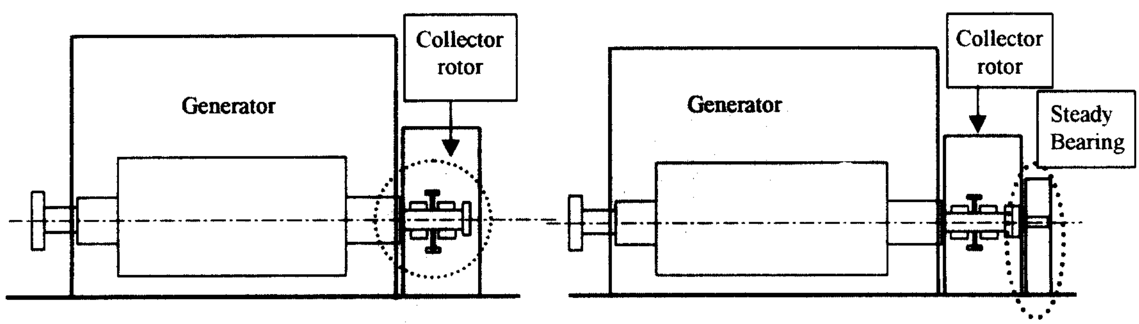


Fig. 1 Schematic view of old generator excitation system with an alternator



(a) Without steady bearing (b) With steady bearing

Fig. 2 Schematic view of new generator excitation system without an alternator

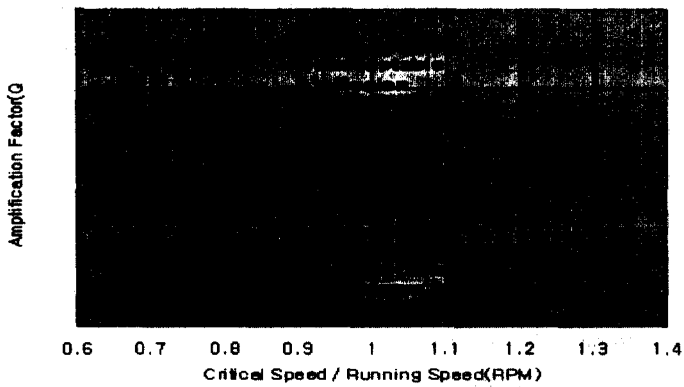


Fig. 3 An example of Q-factor criteria

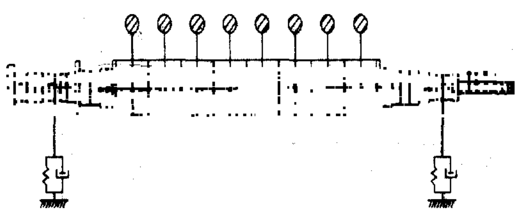


Fig. 4 Mode shape at 3,758 rpm (without steady bearing)

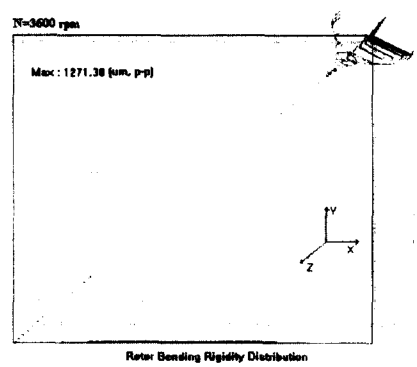


Fig. 5 Unbalance response at 3,600 rpm (without steady bearing)

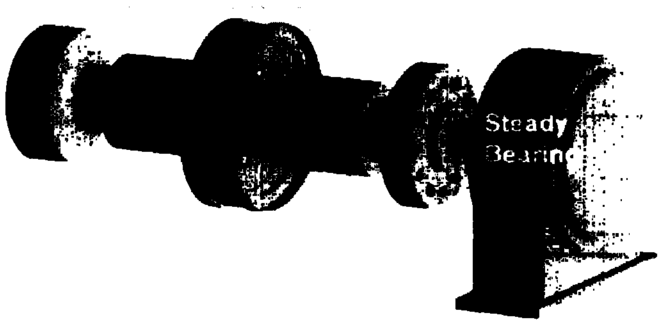


Fig. 6 Schematic view of modified collector rotor system with a steady bearing

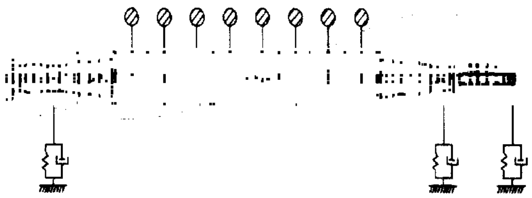


Fig. 7 Mode shape at 3,715 rpm
(with steady bearing)

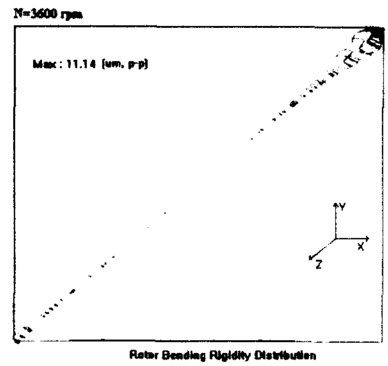


Fig. 8 Unbalance response at 3,600 rpm
(with steady bearing)

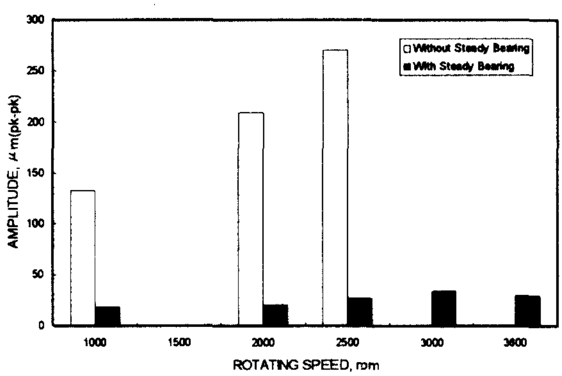


Fig. 9 Comparison of measured vibration amplitude between with and without steady bearing

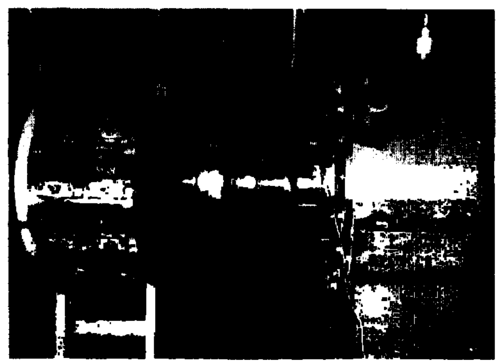


Fig. 10 A photograph of collector rotor system modified successfully