MPR

Clinton Power Station - Division 3 Shutdown Service Water Pump (1SX01PC) Failure to Start - Calculation Summary

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1.0 Purpose

This report provides an overview of the scoping calculations performed by MPR as part of an independent evaluation of the Clinton Power Station Division 3 Shutdown Service Water Pump (1SX01PC) failure to start on September 16, 2014.

2.0 Introduction

2.1. Background and Scope

On September 16, 2014 the Clinton Power Station Division 3 Shutdown Service Water Pump (1SX01PC) tripped on thermal overload and failed to start. During the subsequent disassembly of the pump, it was determined that the suction bell bushing had failed and seized (Reference 1). Evaluations performed by the pump vendor (Sulzer) and its subcontractor (ATS), determined that the specific initiating factor for the failure could not be determined due to the extent of damage (References 2 and 4). However, the vendor report identified a potential contributor as the "water-starving" of the bearing as a result of sediment buildup in the bearing flush groove gradually over many years resulting in heating of the bearing and hard-faced sleeve components (Reference 2).

As part of an independent evaluation of the cause of the pump bearing failure, MPR performed several scoping analyses to assess the potential for sediment buildup to be a principal contributor to the pump bearing failure. Those analyses are summarized in this report. Note that the calculations discussed in this report represent only a portion of the work performed as part of the independent failure assessment which is summarized in Reference 3.

2.2. Pump Overview

The Division 3 Shutdown Service Water pump is a Sulzer model 8X14A VCM pump with a Siemens-Allis type RGV motor. The pump is a vertical two-stage deep well pump that consists of a discharge head, 6 columns and a bowl assembly to extend approximately 30 feet below the floor elevation. The pump is driven by a 75 horsepower, 1760 RPM 460 volt motor (Reference 16). The pump supplies cooling water to safety related Division 3 loads during emergency conditions. (Reference 1)

A cross-section of the pump is provided in Figure 1. As shown in the figure, the pump is equipped with a total of 9 radial bearings. The suction bell is located at the bottom of the pump shaft with a hub that houses a bushing to stabilize the shaft. The bottom bell bushing is located about 21 ft below the normal water level, and about 11 ft above the bottom of the suction bay. (Reference 3). The bushing is equipped with a spiral cooling channel (Figure 1), and is interference fit to the suction bell housing, and positively locked in position with a set screw. The sleeve has a hard facing with a loose fit tolerance between the sleeve and shaft of +0.001" to +0.003", and the joint is keyed (Reference 4). The clearance between the shaft sleeve and bushing is about 0.007 inches diametric (Reference 10).

3.0 Summary

As part of the independent assessment of potential contributors to the Clinton Power Station Division 3 Shutdown Service Water Pump (1SX01PC) failure to start in September 2014, calculations were developed to provide insight into the following areas:

- Bearing Design Performance
- Bearing Cooling Flows
- Bearing Temperatures and Thermal Stresses

An overview of the calculations developed in each of these areas is provided in Section 4.0. The overview provides a summary of the calculation (and verification) approach, and the conclusions developed from the calculations.

The following observations are developed from the calculations:

- The flow and thermal characteristics seen by the Suction Bell Bearing are not significantly different from those seen by the other pump radial bearings. In particular,
  - The flow through the bearing is primarily a function of the shaft turning in the bushing and not the differential pressure across the bearing. Since all of the bearings see the same shaft speed, the flow through the suction bell bearing is similar to that seen by the other pump bearings.
  - The suction bell bearing is lightly loaded under normal operating conditions, and the other radial bearings are expected to be similarly lightly loaded.
- The temperatures seen by the suction bell bearing are well below the capabilities of the bushing and hard facing materials:
  - Even with a conservative estimate of the bearing heat load, contact area, and heat dissipation capability, the peak temperature in the bearing is estimated to be 158°F (Reference 6). This is well below the melting temperatures of the bushing and hard facing materials of 620°F and 1900°F, respectively (Reference 3).
- The thermal stresses seen in the hard facing are small and are not expected result in failure of a properly applied hard facing:
  - Startup and Steady State: Hard facing stresses are small (less than 2 ksi) and compressive during startup and steady state operation (Reference 6).
  - Normal Shutdown: The normal shutdown transient was not explicitly evaluated in the calculations summarized here. However, temperature gradients during the slow cooldown associated with a normal shutdown (no forced cooling with shaft at rest) are expected to be low and would be bounded by the postulated rapid (forced cooling) scenario discussed below.
  - Postulated Intermittent Rubbing Transient: Even for a postulated intermittent rubbing transient in which the bearing is:

- First assumed to rub with no cooling long enough to reach a steady state temperature distribution where the heat dissipation rate is equal to the conservatively estimated rubbing heat load of 100 watts, and
- Then the bearing cools rapidly as flow and bearing film is assumed to be reestablished so that the heat load is reduced to zero and the water cools the already heated components,

The peak tensile stresses in the hard facing are relatively low (i.e., about 6 to 10 ksi) (Reference 6).

As discussed in Reference 3, the fatigue properties of the hard facing material are not well known. Published information for a nickel based hard facing with chemical composition which is similar to the Colmonoy 6 material used in the 1SX01PC pump indicates that the conservatively estimated stresses discussed here would be less than the endurance strength of the hard facing (Reference 12). However, the vendor published tensile strength (Reference 13) for the Colmonoy 6 material used in the pump is lower than the reported endurance strength from the test data discussed above. An endurance strength limit derived from the vendor published tensile strength would be expected to be significantly less than the tensile strength. In this case, the potential for the hard facing to experience high cycle fatigue damage under the postulated intermittent rubbing scenario cannot be completely ruled out (i.e., the calculated cyclic stress starts to approach the range of expected endurance strength). However, considering 1) the conservative assumptions used to develop this postulated alternating stress (e.g. conservative radial loads and heat generation rate, loads carried (by rubbing) over just the top 1 inch of the bearing, intermittent rubbing and heating to steady state temperatures followed by reestablishment of cooling water and rapid heating), and 2) that the bearing would have needed to experience one of these postulated cycles once every 2 seconds in order to see the endurance limit of 10^6 cycles (the pump has ~590 hours of operation), it is judged that if the hard facing is applied in accordance with general industry best practices, then the hard facing would not be expected to crack due to thermal stresses.

In summary, the evaluations show that, even if the cooling channels were completely blocked with silt, a properly applied hard facing would not be expected to crack or delaminate due to high temperatures or thermal stresses, and the temperatures would not be expected to damage the bushing.

4.0 Calculation Overview

Figure 2 provides a high level road map to the calculations. As shown in the figure, in addition to calculations aimed at developing insights into bearing cooling flows, temperatures, and thermal stresses, calculations were also performed to develop insights into the expected performance of the bearing under a range of conditions. An overview of the calculations in each of these areas is provided below. The overview addresses the analysis approach, the relationship between calculations, and the steps taken to verify the analysis results. In this regard, it is noted that while the calculations are not safety related and were not performed under a 10CFR50, Appendix B Nuclear Quality Assurance Program, all of the calculations were prepared in accordance with the MPR Standard Quality Assurance Program, which includes a formal check

and review as indicated by the signatures on the cover page of each calculation. This is discussed further in Section 4.4 below.

4.1. Bearing Design Performance

- Calculation 0065-0055-010, "1SX01PC Suction Bell Bushing Performance" (Reference 8)

  In order to develop insights into the design performance of the bearing, the bearing was analyzed for a range of potential conditions using the software program JURNBR. This program is part of the Advanced Rotating Machinery Dynamics (ARMD) suite of tools and is widely used in the analysis of bearings with over 350 users worldwide including major bearing, turbine, and pump vendors (Reference 15). MPR has previously verified the JURBNR (for journal bearings) and THRSBR (for thrust bearings) portions of the ARMD code by performing formal verification calculations, in which the results from the software package were compared against published solutions for selected analysis cases. The Commercial Software Verification performed by MPR is formally documented in accordance with MPR's QA Program requirements.

  The calculation provides general insights into the design bearing performance (e.g. eccentricity, peak pressure, viscous heat load, and film thickness) as a function of water temperature and load. The analysis shows that the spiral cooling water groove in the bushing essentially breaks the bushing into several segments of shorter length with the peak hydrodynamic film pressure at the center between the grooves.

  While the calculation shows a viscous fluid film in the bearing, the output of this calculation is not directly used in the final estimate of worst case bearing temperatures and thermal stresses. Rather, as discussed in Section 4.3 below, the thermal analyses of the bearing under postulated silted conditions conservatively consider the impact of a complete loss of the hydrodynamic film, and therefore rubbing between the shaft sleeve and bushing, on the bearing temperatures.

4.2. Bearing Cooling Flows

- Calculation 0065-0055-014, "Computational Fluid Dynamics Analysis of Flow in the Suction Bell Bearing of the Pump 1SX01PC" (Reference 10)

  A CFD analysis was performed to provide insights into the following areas:

  - Flow through the bearing cooling channel and annulus:
    - During Normal Operation
    - With Partial Blockage of the cooling channel
    - With Complete Blockage of the cooling channel
  - Sensitivity of Bearing Flows to the differential pressure across the bearing

  A detailed discussion of the modeling approach, assumptions, and boundary conditions for each analysis case is provided in the body of the calculation.

The analyses were performed using the ANSYS CFX computer code. CFX is a commercial program that is validated against experimental data for various test cases (Reference 11). Given the complexity of the CFD analyses, scoping calculations were also performed to further validate the model. These analyses included 1) a comparison of flow velocities at the inlet to the bearing calculated in the CFD analyses, to the nominal approach velocity at the bell inlet diameter calculated by hand, 2) a comparison of the flow rate seen in the bushing cooling channel under normal operation, to that estimated using a hand calculation method in published literature. The hand calculation methodology is provided in Appendix F of the calculation. Both evaluations showed reasonable agreement between the hand and CFD analyses. For example, the flow rate in the bushing channel calculated by the hand calculation of 0.099 gpm matches reasonably well with the CFD analysis value of 0.084 to 0.088 gpm.

The flow analysis provides the following insights into the performance of the bearing under design and off-design conditions:

- The flow through the bearing is largely driven by the rotating shaft; the differential pressure across the bearing does not have a significant influence on the bearing flows. As a result, the flows seen by the impeller and column bearings under normal operating conditions are expected to be similar to that seen by the suction bell bearing.
- Under normal operating conditions, the flow through the bearing is dominated by the flow through the channel (i.e. flow through the channel is ~0.086 gpm, and ~0.024 gpm through the annulus).
- The impact of cooling channel blockage on the total bearing flow rate is highly dependent upon the extent of blockage. For example:
    - Partial blockage of the cooling channel [i.e. complete (but local) blockages near the channel entrance and exit] had no impact on the flow through the annulus, and resulted in only a small reduction (i.e. ~10%) in the flow through the cooling channel. This is because flow was able to effectively bypass the blockage through the annulus.
    - If the flow channel is completely blocked, then water is circulated into and out of both ends of the bearing, with a stagnant portion at the center, and no net flow through the bearing.

  The thermal analyses discussed below in Section 4.3 conservatively consider the impact of a complete loss of all cooling flow on the bearing temperatures. Therefore, the output of the CFD analysis is not directly used in the final estimate of worst case bearing temperatures and thermal stresses.

4.3. Bearing Temperatures and Thermal Stresses

The temperatures and thermal stresses experienced by the bearing are dependent upon 1) the heat generated in the bearing, and 2) the heat removal capability of the bearing. The approach used to develop a conservative estimate of these parameters under off-design conditions is described below. Note that, due to the conservative assumptions on heat generation and cooling flow

discussed below, the output of the Bearing Performance and Bearing Cooling Flow calculations discussed above were not used or needed in this calculation. Therefore, the calculation of peak hard facing temperatures and thermal stresses does not directly utilize the CFD software or bearing software (JURNBR) discussed above.

- <u>Bearing Heat Generation</u>

  The heat generated in the bearing is a function of a number of factors including 1) bearing load, 2) the bearing film thickness and the potential for rubbing between the bushing and shaft sleeve, and 3) viscous heat generation. These parameters are addressed in the calculations as follows:

  – Bearing Loads: By design, the loads on the radial bearings of a vertical pump are typically very low. This was confirmed for the shutdown service water pump. Loads on the Suction Bell Bushing were estimated to range from less than 1 lbf under normal conditions to 32 lbf under very conservative conditions in which the pump wear rings are assumed not to provide any lateral support, and moderate to severe impeller radial loads and misalignment are assumed to exist.

    The impeller loads are estimated using hand correlations which consider constant and dynamic hydraulic loads, hydraulic unbalance, hydraulic broad-band noise, and mechanical unbalance loads. The distribution of the impeller loads across the various shaft support points (bearings and wear rings) is estimated using the ANSYS structural analysis code and considers the impact of potential misalignment on the bearing loads. See Calculations 0065-0055-006, "Impeller Radial Thrust Loads of Pump 1SX01PC" (Reference 5) and 0065-0055-012, "Impeller Radial Load Distribution" (Reference 9).

  – Bearing Film Thickness and Heat Generation: The general bearing evaluations developed in Calculation 0065-0055-010 (Reference 8) discussed above show that the bearing would be expected to carry the small loads seen by the bearing without rubbing. However, it is conservatively assumed in this evaluation of the bearing temperatures that the shaft sleeve rubs against the bushing with the conservative load of 30 lbf. For these conservative conditions, the heat generated due to rubbing contact in the bearing is estimated to be ~100 watts. See Calculation 0065-0055-008, "Contact Heat Generation" (Reference 7).

  – Viscous Heat Generation: The contribution to the heat generated in the bearing due to viscous friction is shown in Calculation 0065-0055-010 (Reference 8) to be small (e.g. less than 8 watts).

  The analysis of the bearing temperatures is performed considering a bearing heat generation of 100 watts. As discussed above, this assumes full rubbing contact in the bearing and is considered to be a very conservative estimate of the heat which would be expected to be generated in the bearing. Even though some additional viscous heat generation will occur during full rubbing contact conditions, it will be small (a few watts) compared to the rubbing contact heat generation and therefore it is considered acceptable to not include this in subsequent calculations of peak temperatures and thermal stresses.

- <u>Bearing Heat Removal and Calculation of Peak Hard Facing Temperatures and Stresses</u>

  The bearing thermal and structural analyses conservatively assume that all of the heat generated in the bearing is removed either 1) through the bushing to the bushing housing, and then to the water on the OD of the bearing housing, or 2) through the shaft sleeve to the shaft, and then by convection to the water passing around the shaft. The calculation conservatively assumes that no heat is removed by water flowing through either the bearing channels or annulus. This represents, with some conservatism, the condition of fully blocked bearing cooling channels.

The temperature distribution and corresponding thermal stresses in the bearing components are calculated in Calculation 0065-0055-007, "Finite Element Analyses of the Div. 3 Shutdown Service Water Pump (1SX01PC) in the Vicinity of the Suction Bell Bushing" (Reference 6). The analyses are performed with the ANSYS finite element analysis code. ANSYS is a commercial program that is validated against test data for various test cases (Reference 14). Simple engineering evaluations of the FEA analysis results such as scoping analyses of the relative thermal growth of the hard facing and shaft were performed as part of the evaluation process.

The analyses include a sensitivity evaluation in which the shaft is assumed to be rubbing only over one inch at the top of the bushing rather than over the entire length of the bearing. The results for this bounding condition are summarized here.

- Steady State: This analysis case considers the steady state temperatures seen for the case in which the bearing is water starved and rubbing. The analysis shows that approximately 72% of the heat generated in the bearing is dissipated through the shaft sleeve to shaft, while the remaining 28% is dissipated through the bushing and housing to the surrounding water. The maximum temperature experienced by the bearing is ~158°F and the stresses in the hard facing are compressive and less than 2 ksi.
- Startup Transient: The stresses in the hard facing are compressive during the startup transient and less than 2 ksi.
- Intermittent Rubbing/Cooling Transient: In order to fully assess the potential for the hard facing to experience the type of high tensile stresses which could result in fatigue damage, a postulated transient in which the bearing was allowed to fully heat up and then rapidly cool was defined as follows:
  - The shaft is first assumed to rub with no cooling water flow long enough to reach a steady state temperature distribution.
  - It is then assumed that the bearing water flow is reestablished (e.g., due to shaft loads decreasing leading to development of a hydrodynamic fluid film at the bearing surface) so that the heat load is reduced to zero (i.e., the shaft is no longer rubbing), and the water flow through the bearing is allowed to rapidly cool the already heated components,

  The peak tensile stresses calculated in the hard facing for this severe transient are relatively low and range from ~6 to ~10 ksi, depending on whether the shaft is assumed to rub over the entire length of the bearing, or only over the top one inch (Reference 6).

A detailed discussion of the modeling approach, assumptions, and boundary conditions for each analysis case is provided in the body of the calculation.

The bearing temperature and thermal stress analyses results were evaluated against the available hard facing material information as previously discussed in Section 3 above.

4.4. Quality Assurance

MPR performed the evaluations for this independent assessment of the potential causes of the pump bearing failure under our Standard Quality Assurance program. An overview of the analysis verification process required under this program is provided below:

- Commercial Software is controlled and verified to be suitable for use for its intended purpose. This qualification is formally documented under our Standard QA Program. The qualification of Commercial Software is based upon such factors as industry experience with the software, and MPR experience with the software, including results of verification analyses performed by MPR or the vendor. The method of qualification for the Commercial Software used in the pump bearing analyses (i.e. JURBNR, ANSYS CFX and ANSYS FEA) was previously discussed in Sections 4.1 through 4.3 above.
- A check and review is performed of all software inputs and outputs against appropriate references and assumptions are documented. The reviewer also has the responsibility to ensure the software program and analysis models are appropriate for the analysis being performed, in addition to ensuring the outputs of the analysis are reasonable and consistent with expectations based on the reviewer's engineering experience and judgment.

5.0 References

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8. MPR Calculation 0065-0055-010, "1SX01PC Suction Bell Bushing Performance," Revision 0.
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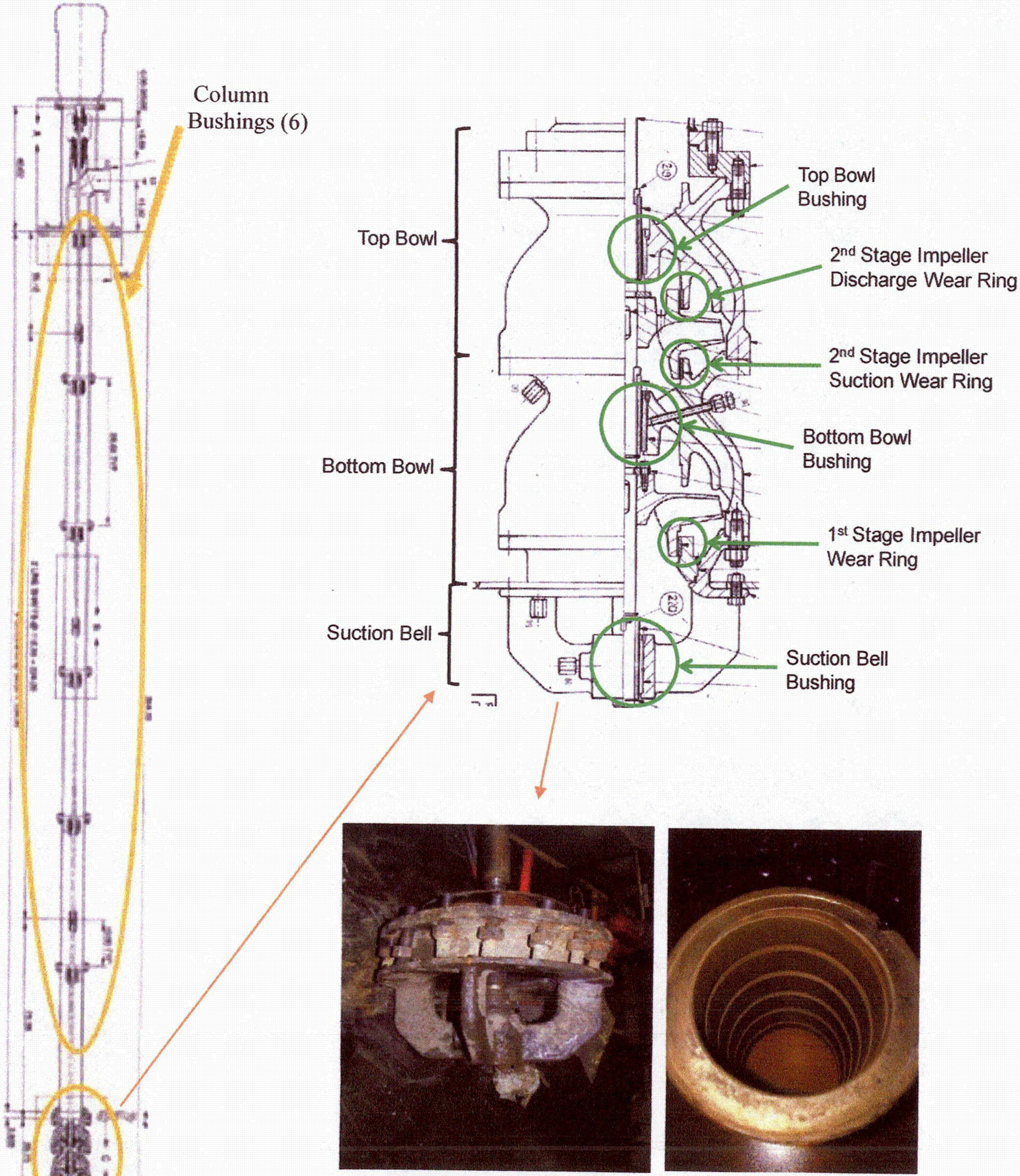

Typical Bushing Showing Spiral Channel

Figure 1. Pump Cross Section

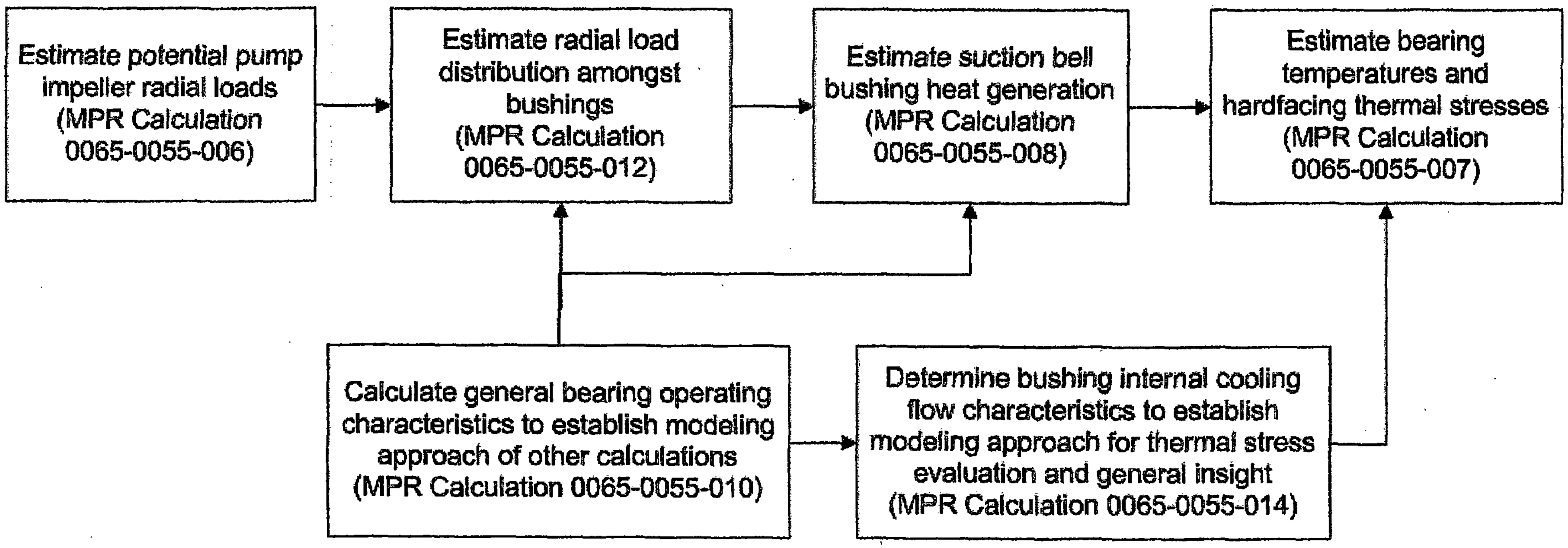

Figure 2. Calculation Roadmap