# Case Histories at a Nuclear Power Plant

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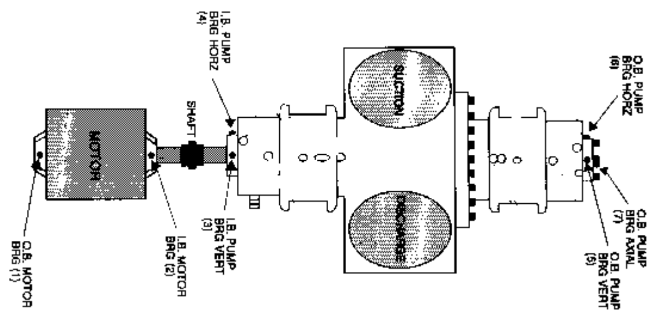
## System Cooling Pump - Bearing cooling water too cold?

It is February, and Operations starts a system cooling pump. The pump is used to circulate 270 deg. F. water through a heat exchanger, and the bearing cooling water temperature (Lake Temperature) was 42 deg. F. After the pump was started, operators received a high temperature alarm (160 deg. F) on the pump's inboard bearing and stopped the pump. The redundant pump was started and pump inboard bearing temperature also increased and exceeded its alarm setpoint (160 deg. F). Operators secured the second pump. These pumps have not had any problems in the past, so what is happening?




The pumps are single stage, high flow, low pressure design with double angular contact thrust bearings located at the outboard end, and a floating deep groove ball bearing inboard. The bearing housings were changed from cast iron to stainless steel in 1992 due to cooling water corrosion problems.

After the bearing temperature transients, the history of the components was researched, and it was determined that Predictive Maintenance had changed the oil viscosity from a VG-22 oil to a VG-46 oil a few months before this incident. The oil type change was initiated due to an analysis performed on the outboard thrust bearings, which indicated the need for a higher viscosity oil. The outboard bearings were experiencing "frosting" of the balls, which then caused excessive cage wear. This "frosting" is due to the lack of preload on the face to face installed thrust bearings. This normally promotes ball skidding, but analysis of the bearing races indicated no skidding. Analysis suggested that impacting with the cage was causing the ball roughness. No preload could be applied due to pump design, but higher viscosity oil was expected to mitigate the problem. Metal particle generation dropped significantly with the higher viscosity oil installed.

After both pumps failed to run under the cold cooling water and hot process water conditions, the oil was changed back to a VG-22 oil, and the pumps were run again. This time the first pump performed adequately by not exceeding the bearing temperature alarm limit, but the second pump did not.

The cause of the event was determined to be an equipment design problem. The design of the pump did not adequately consider the relationship of:

1. the range of bearing cooling water temperature
2. the clearances between the inboard pump bearing and bearing housing.

Failure to adequately consider these parameters allowed an interference fit between the inboard bearing and its housing when the pump was asked to pump 270 deg. F. water with bearing cooling water below 60 deg. F. These factors contributed to an interference fit, which caused the bearing to bind in the housing instead of moving to compensate for axial thermal growth of the housing and the shaft. The resulting compressive and axial loading on the inboard bearing caused excessive heat generation and the high temperatures experienced. It is suspected that if the pump had not been shutdown when the high temperature alarm was received, the inboard bearing would have failed due to fracture of the cage from excessive axial load.

A contributor to the event was the oil viscosity. In combination with the cold Service Water temperatures, the magnitude of the bearing housing shrinkage and the bearing fit (at the upper limit of the tolerance), the heavier oil viscosity resulted in greater heat generation. The higher viscosity causes internal heating which causes greater thermal expansion of the bearing race, making the interference fit worse.

**Modification to Correct the Problem**

A calculation was performed to predict theoretical regions of acceptable operation in terms of bearing housing temperature, bearing temperature and a range of possible installation fits between the bearings and housings. This calculation proved by analysis that the pumps were susceptible to high interference fits at low cooling water temperature, and this problem was made worse by installing stainless steel bearing housings. Stainless steel has a larger coefficient of thermal expansion and a lower heat transfer coefficient. Consequently, the material change to stainless steel exacerbated the original problem by further decreasing the housing bore for low cooling water temperature.

The pumps were modified to correct the problems. This modification included:

- Identified and verified proper critical dimensions in both pumps.
- Returned inboard bearing housings to original cast iron material.
- Replaced inboard bearings with 6217 C4 bearings (from C3), which have larger internal clearances.
- Increased ID of inboard bearing housing to a modified H7 fit in the upper half of the tolerance band.

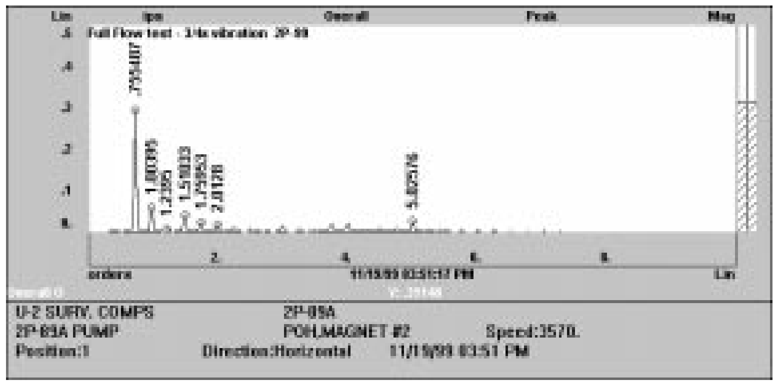
Testing was done after the modifications were completed. The pumps were tested by operating with 270 deg. F. process water and a bearing cooling water supply at ~32 deg. F. and then with a bearing cooling water supply at 121 deg. F.

A review of other pumps in the facility found another pump from a different manufacturer with the same type of problem, and in that instance the pump supplier had been given a specification that included all possible ranges of process and service conditions.
A generic issue may exist with pump vendors and designers following traditional methods of design even when informed of significant variations in the pumps service conditions.

Now that the thermal binding problems are solved, we are faced with the problem of the accelerated wear on the outboard thrust bearings. All testing was done with ISO-22 oil and the calculations for bearing heating, thermal growth, etc… all used a ISO-22 oil as the lubricant. Using a higher viscosity oil in the inboard bearing could cause more heating and growth than is desirable even with the modified dimensions. This essentially prohibits us from changing to any other oil type or viscosity grade in the future. The wear rate on the outboard thrust bearing cages (from the rough ball surfaces) is quite rapid. Analysis of bearings removed in the last several years indicates the outboard bearings exhibit extreme cage wear at the ball pockets after 2000 hours of operation (~90 days continuous running). Since these pumps are normally in standby, this translates to changing the outboard bearings on both pumps approximately every three years.

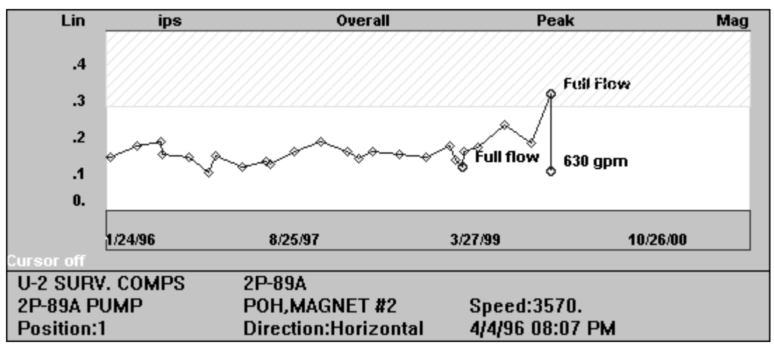
## High Pressure Injection Pump - 3/4x vibration

2P-89A is a Ingersoll-Rand 9 stage high pressure injection pump driven by a Allis-Chalmers 600 hp 3600 rpm motor. Normal quarterly testing is done through a minimum recirculation line at a very low flow rate. The pump is tested at Full Flow (750 gpm) every 18 months. While testing the pump at Full Flow conditions during a recent unit shutdown, vibration amplitudes were above the acceptable normal range on the pump outboard horizontal point. Analysis of the vibration data indicated the vibration increased from .117 ips to .320 ips. The increase in vibration occurred at 3/4x running speed (2685 cpm). This 0.75x frequency was predominate on all points during the Full Flow test at 775 gpm, but was not present during a subsequent test run at 630 gpm. The 3/4x frequency was not present during the previous Full Flow test done 6 months earlier.

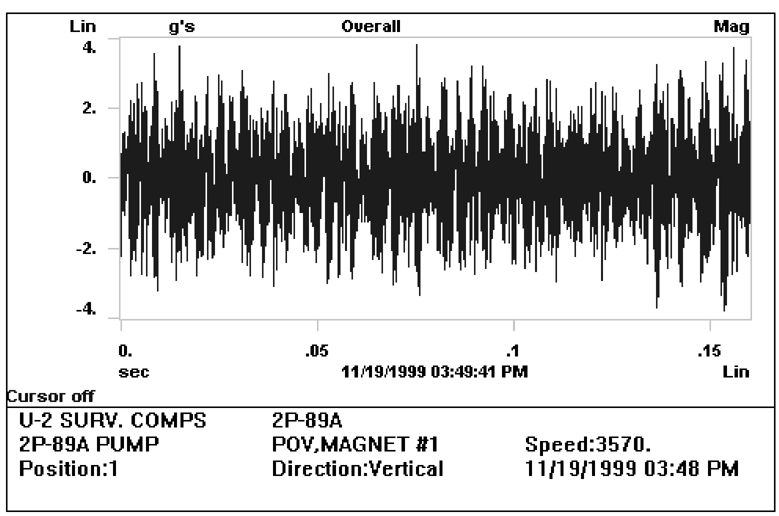



Detailed data was collected on the pump during the Full Flow test, including high-resolution spectrums and waveforms. We theorized that the 3/4x vibration was a rub, but the waveforms did not have any rub characteristics.

A history search indicated the outboard bearing was changed out prior to the Full Flow test due to early indications of bearing wear detected by Oil Analysis. A shim pack used in the bearing housing affects the location of the pump rotor in the casing. This maintenance could be the cause of the significant difference in Full Flow vibration readings from the previous full flow test to the current one.




The history search also revealed that this frequency was observed once before on another High Pressure Injection pump, 2P-89C, during a Full Flow test in April of 1994. The .75x frequency (during that test) also disappeared as flow was decreased on the pump. The amplitudes of the .75x frequency approached .7 ips on some points. Significant Analysis was done at that time to explain the frequency. 2P-89C also failed its DP criteria during that full flow test and was declared inoperable. 2 months later, 2P-89C failed a seal and seized up during a system integrity test.



The Predictive Maintenance analysis on 2P-89C in April of 1994 came to the following Conclusions:

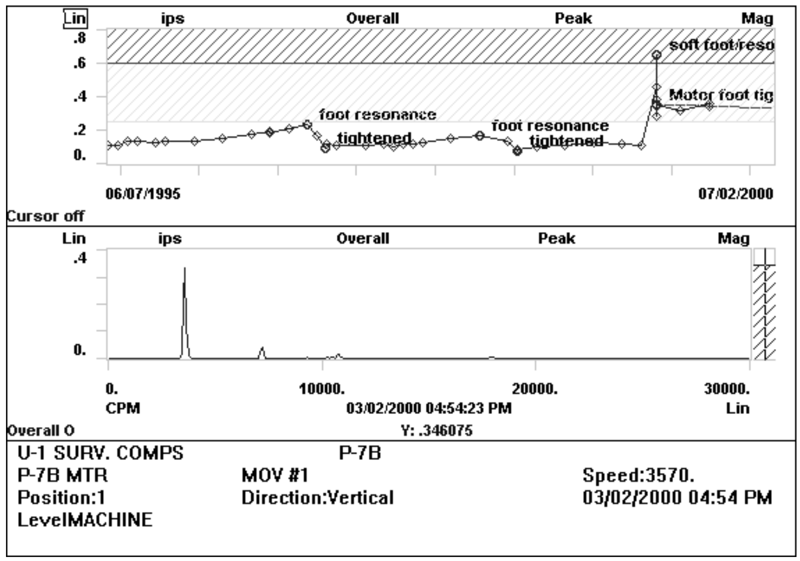
- *The subsynchronous vibration could be the result of hydraulic "swirl", excited by interstage leakage. It is noted that as the pump flow is reduced, and interstage pressure is increased, the .75x energy abates. The increased interstage pressure provides the condition for the casing rings to act like small hydrostatic bearings; dynamic dampening, known as the Lomakin Affect also occurs as the ring differential pressure increases.*
- *This condition could also be internal contact between the impellers and casing rings, or balance drum and sleeve. It is noted that the vibration is highest on the outboard/thrust bearing, possibly pointing to the drum and sleeve.*
- *Pumps of similar construction, at another installation, have exhibited this .75x vibration. Disassembly revealed opened clearances due to rotor contact at very high BEP flows. The "wide open" clearances made it difficult to define if the .75x was a rubbing or high interstage leakage hydraulic subsynchronous swirl. The Pump is essentially performing hydraulically to the original design performance curve. The .75x may be rubbing contact at high flows, lower delta-P, between the drum and sleeve face. It is recommended that the axial balance drum setting be checked at the plants earliest convenience.*

During the pump rebuild, excessive internal clearances were found at the wear rings.

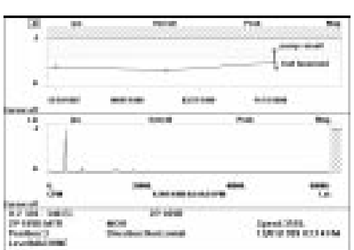
With this information from the earlier analysis on an identical pump, and the absence of rub indications on the waveforms, it was concluded that he indications were the result of recirculation or interstage leakage. These phenomena are most likely caused by axial mis-positioning of the pump rotating element during the outboard bearing change.

## Foot Related Resonance, it can happen to you!

Foot related resonance is a much more common problem than you might think. Foot Related Resonance occurs when a motor or pump frames stiffness can be significantly affected by the tightness of the mounting bolts, and with all of the bolts tight the frames resonance is very close to a forcing frequency. Typically, if the resonance is at running speed, a multiple of running speed, or 2x line frequency, then a high vibration will result at that frequency. As with most resonance, the resulting high vibration is very directional, causing a high vibration in the Vertical, Horizontal, or Axial direction, but only one of the three. The problem has been confused with Soft Foot, but you can tell the difference by the fact that the problem will manifest even when a soft foot check is good. Also, it usually does not matter which of the 4 hold down bolts are tightened or loosened to bring down the vibration. As our maintenance and alignment practices have improved over the years, I have run into several cases of foot related resonance.



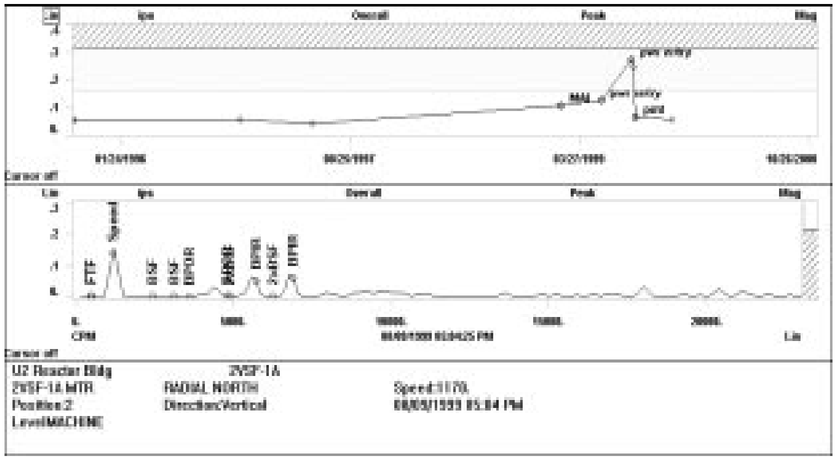

**P-7B** is an Emergency Feedwater Pump used to send cooling water to the plants steam generators for cooling of the Reactor Coolant System in the event of a loss of conventional cooling. Vibration data is collected on the motor and pump during routine surveillance. In 1996 it was noted that vibration amplitudes were trending up on the outboard motor vertical point. No change in amplitudes was noted in the axial or horizontal direction. All of the increase was at the motor running speed. The bearing is a plain journal bearing, and oil analysis did not indicate any problems. The motor feet were tested for a developing soft foot, and none was found, but it was noted that after the foot was re-tightened the vibration returned to normal. We began to suspect a foot-related resonance. In 1998 vibration amplitudes started trending back up. The right outboard motor foot was tightened again, and again vibration returned to normal. A scheduled pump rebuild was carried out in 1999, and the motor was completely removed for access. Vibration amplitudes on the outboard motor vertical point were very high during post maintenance testing. All other points were normal. A soft foot check was performed, and no problems were found. Testing was performed which consisted of loosening and tightening the motor foot bolts with the motor running. It was found that vibration amplitudes on the outboard vertical point could be significantly affected by bolt tightness, but no amount of tightening or loosening could return the vibration levels back to the previously low level. We have theorized that the motor foundation has powdered grout or voids under the plate that is causing the changing stiffness and we can now, no longer move the resonance far enough away from the running speed to achieve the previously low levels. Maintenance is scheduled to check for this problem and do more troubleshooting in the near future.



**2P-109A and B Reactor Makeup Water Motors:**

Both the 2P-109A and B motors exhibit foot related resonance at the running speed in the horizontal direction. These are small frame Toshiba motors that have had this problem since installation. Part of the post maintenance testing of these motors after alignment is to "tweak" the tightness of the motor foot bolts until satisfactory vibration is achieved in the horizontal direction.

**Joy Axial Vane Fan:**
During a refueling outage in May 1997 the 2VSF-1A Containment Cooling Fan Motor was rebuilt according to procedure. New NTN shielded deep groove ball bearings were installed in the motor and the fan was placed back in service. Post Maintenance Vibration testing was satisfactory. This fan design is essentially a motor inside a duct. Vibration data is collected on the outside of the duct (housing) several inches from the actual motor casing.




Vibration analysis during a refueling outage in Feb. 1999 indicated slight bearing degradation in 2VSF-1A. Due to time restrictions, a decision was made to change out the motor bearings during an upcoming mid-cycle outage, with periodic data collection via power entries in the interim.
Vibration data collected during a subsequent power entry on 5/21/1999 indicated linearly increasing degradation and vibration levels were well below Predictive Maintenance alarm limits.
Data collected on 8/4/1999 indicated a significant change in level of degradation.
Additional data collected on 8/9/1999 confirmed the high level of degradation. The fan was declared inoperable based on Predictive Maintenance judgement that the fan could not fulfill it's required 30 day accident mission time.
2VSF-1A's bearings were changed out while the unit was at power between the dates of 8/10/1999 and 8/15/1999 via a combined site effort. Upon disassembly of the motor, electricians noted that a winding heater lead was broken inside the "opposite drive end" of the motor. The following is documentation of the broken heater lead and possible cause.

> *The heater lead failure was caused by the wires that feed the internal motor heater on 2VSF-1A making contact with the internal rotor fan blades in the outboard end of the motor. Although the heaters (3) are secured in place with clips, the heater wires going to the heater box termination on the outside of the motor are free and un-tethered. They run between the outer side of the stator windings and the motor casing. The heater wires are intentionally left long to allow for the installation of the endbell. The problem arises when the heater leads are not gathered and pulled tight inside of the termination box when the endbell is bolted in place. This allows the excess wire to gather at the end of the motor in the endbell area, adjacent to the rotating fan blades. This is what caused the heater lead to contact the rotating element in 2VSF-1A. Because the heaters are attached to the endbell it is hard to judge the slack left in the heater wire when the endbell is bolted in place on the end of the stator. There is no way to visually check to see that the wires won't make contact with the rotor fan blades. One way to make sure that the slack is removed from the wire is to gradually pull it into the termination box as the endbell is being installed, and pull it tight and secure it in the box with a tie-wrap or tape after the heater termination is made. Although the wires' proximity to the fan blades can't be visually checked, it can be reasonably assured that pulling them tight in the box will keep them out of the rotational path of the fan blades on the inside of the stator. The important thing is to prevent any excess wire from spooling down into the endbell.*

Inspections revealed the heater lead had contacted the motor rotor fan fins. This contact rubbed a significant amount of metal (aluminum) from the fins and wore through the heater lead.

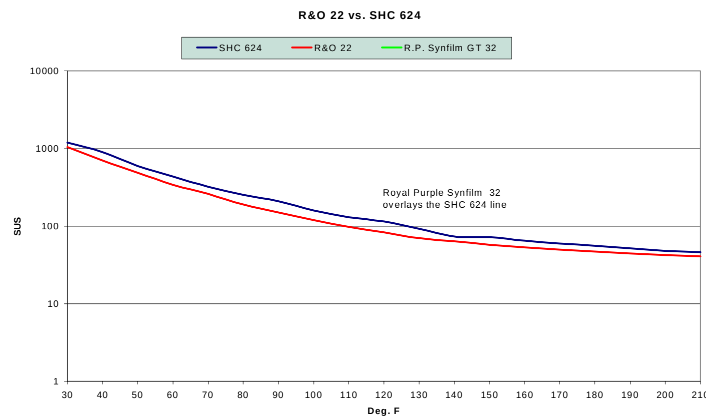
When the drive end bearing was removed from the motor shaft, it felt rough when turned by hand and showed a tendency to "lock up". The opposite drive end bearing felt smooth when turned by hand. New bearings were installed in the motor, the heater lead was repaired, and the motor was reassembled. Special care was taken to pull the excess heater lead wiring out of the endbell into the termination box to prevent reoccurrence of contact with the motor rotor. Post Maintenance vibration testing indicated acceptable amplitudes and the fan was declared operable.
The "bad" bearing was sent by Predictive Maintenance to the manufacturer (NTN) for failure analysis and the Failure Analysis Report was received on 10/29/1999.

NTN Analysis:
The NTN report concludes that the bearing failure mode is lubricant deterioration and contamination, which lead to heavy wear, raceway and rolling element flaking, smearing and debris damage. The size, shape, depth, and regularity of the damage is fairly common in bearings that have been subjected to electrical arching damage.

ANO PdM Analysis:
The expected life of this bearing is on the order of 12 years or greater. During disassembly it was found that a motor heater lead had contacted the motor rotor cooling fan fins, causing wear of the fins and generation of a large amount of aluminum dust. It is concluded from the NTN bearing analysis that arcing inside the bearing from the energized heater lead rubbing against the rotor fins and generation of aluminum dust contaminating the bearing grease were the primary causes of the bearing failure. The heater's are interlocked to deengerize when the motor is started. This fan was started and stopped several times after the bearing change in 2R12 and the heater lead was automatically energized each time the motor was stopped. The partially worn lead would then have caused arcing against the rotor fins and through the bearings. Based on the amount of wear on the rotor fins, the heater lead rubbed for quite some time before complete separation occurred. Even without the arcing occurring, the aluminum dust contamination of the grease would be sufficient to cause the degradation seen in the bearing.

**Synthetic Oils, Solutions for difficult problems:**

Synthetic Oils enjoy somewhat of a mythical status among many people. In most cases a high quality mineral oil will perform just as well and cost much less. Synthetic oils do enjoy some properties that make them excellent choices in certain situations. Because synthetics have none of the waxes that are in most mineral oils, they are the best choice for very cold conditions. They enjoy a significantly lower pour point, than mineral oils, and typically have a flatter viscosity curve (high viscosity index). An example of this is a comparison of a PAO (poly-alpha-olephin) type synthetic such as Mobil SHC 624. This oil is billed, as an ISO 32 grade oil because it's viscosity is close to the VG 32 standard at 100 deg. F. but due to it's flatter viscosity curve it has a viscosity at 32 deg. F that is very similar to a mineral based ISO 22 oil, and at 212 deg. C it has a viscosity very similar to a ISO 46 oil. For a component that experiences extremely cold and hot ambient temperatures, this type of curve can be a real help. Synthetic oils also have a much greater resistance to heat, allowing operating temperatures of 200 deg. F or hotter without breakdown or sludge buildup. The VI improving molecules in synthetics actually increase in size as temperatures increase. Also, mineral oils contain aromatic chains that make the viscosity unpredictable at extremes of temperature. Synthetic oil molecules are extremely consistent making their performance much more consistent across all temperature ranges.




In certain situations, the better friction characteristics can allow a component to run cooler as well. Recently, we needed to specify an oil for a component that must operate in several extremes of temperature. The bearing cooling water temperature ranges from 33 deg. F to 121 deg F. depending on the time of year. The fluid being pumped can range in temperature from 70 deg. F. to 270 deg. F. A mineral oil that met the bearing manufacturers recommended minimum viscosity of 70 SUS for all operating conditions did not exist. Our evaluation turned to synthetics and a Mobil SHC series oil was found that met the requirements. As an added benefit, after the oil was installed it was found that the bearing temperatures ran from 10 to 15 deg. F cooler with the synthetic oil installed.