ENCLOSURE 2

Oyster Creek Nuclear Station
Technical Background, Basis and Report
Diesel Generator Cooling Fan Drive Test

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(Document containing Engineering Observations and
Parts Comparisons made during Diesel Generator Testing
@ The Joliet Generating Station and
a detailed discussion of the results of the test)

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OYSTER CREEK EDG COOLING FAN DRIVE TEST
Technical Background, Basis and Report

DISCUSSION:

Per the test plan, an agreement was developed with Midwest Power to use one of five EMD engines located at the Joliet Power Station in Illinois to perform the Fan Drive Test. The five diesel generator units at Joliet are combined into a 10 Megawatt EMD design MU Peaking Load Station with five MP36 engine generators and an MC master control unit housing the circuit breakers, major controls, regulators and the starting battery system. Unit #1 diesel generator was selected to perform the required simulated testing. Engine Systems, Inc. (ESI) prepared the test document #6012458-TP-1 to perform the test sequence after restoration of operational readiness to the unit #1 diesel generator.

The Oyster Creek diesel generator units are EMD MP45A units which were equipped with all the MU power plant controls, however, two complete master units (with no slave units) were shipped to Oyster Creek in 1967 for the emergency bus power requirements. The MP45A units contain all controls within the individual housed generator units and each has its own switchgear cubicle unlike the common MC control house at Joliet.

This review will discuss the degree of similarity between the Oyster Creek diesels and Unit 1 at the Joliet plant that allow this test to simulate conditions found on OC EDG1. It will also discuss the variations, which may have affected the resulting test conducted.

EXECUTIVE SUMMARY:

The test was performed on the Joliet MP36 diesel generator with the lower shaft bearing bolts in the as found condition for the May 17, 2004 event at Oyster Creek on EDG 1. With these conditions and equal or equivalent components used, the Joliet engine was operated for a period of 6 hours prior to shutdown caused by engine over temperature. Since EDG 1 operated for 3 hours before manifesting the as found conditions, there is an apparent demonstrated EDG 1 run time of 9 hours before considering test differentials.

Evidence of belt condition, remaining tension and recorded engine temperature shows that EDG 1 was not slipping or losing tension to the same degree as the Joliet test. MPR Associates, Inc. was able to develop a model from the test data and information including calculations of EDG 1 fan speed, cooling system performance and evaluation of radiator performance for EDG 1 during emergency operation under these conditions (provided as attachment 3). Results of the calculations show that airflow would have been maintained at 90% of design with engine temperatures near or at normal levels. Under these conditions, EDG 1 could operate indefinitely. Further evaluation shows that lube oil will remain within necessary limits for airflow as low as 55% of rated design, which is, calculated airflow during the Joliet test. Oyster Creek controls bypass all engine temperature trips in emergency mode per regulations and would not trip for elevated engine temperature.

Oyster Creek concludes that the Joliet run time of 6 hours is confirmed as valid and that Engineering analysis shows EDG 1 would not have lost capability to perform its intended safety system function.

COMPARISON:

The MP36 diesel units use 16 cylinder 645E4 engines while the MP45A diesel units use 20 cylinder 645E4 engines with proportionally more output KW and horsepower. The actual individual cylinders have the same component parts since they are both 645E4's. While the output power varies and some of the accessories are different (lube oil filters, water expansion tank, standby pumps, fuel xfer pumps in unit #3, etc.), these variables are not considered to have any impact on the forces at the lower fan shaft equipment. Review of prints for controls and control functions determined that control differences will have no impact during the simulated test. One prominent variation exists in that the EDG 1 event occurred with the unit at full speed and load while the Joliet test will require a startup sequence with the bolts positioned as found after EDG 1 had been shutdown.

The differences with the lower fan shaft pillow block bearing caused the team to have a spare shaft and bearing assembly, identical to those on EDG 1, shipped to Joliet Station and installed for the test. The MP45A lower fan drive shaft has 10 holes in the coupling flange end to accept the rubber inserts on the engine flexible coupling. The MP36 engine coupling has only 5 rubber inserts that were used to make up the coupling. ESI has indicated that the five element coupling is rated for a maximum of 167 hp, 975 lb-ft torque @ 900 rpm and the 10 element coupling is rated for a maximum of 335 hp, 1950 lb-ft torque @ 900 rpm. The parasitic fan load on the engine is rated at 100 hp (full fan CFM @ 637 rpm) in the MP45A specifications and the MP36 unit is proportionally lower for full CFM @ 571 rpm. It is also noted that the shaft length is 8 ft 8 5/8 inches and the flexibility of the drive coupling with 5 or 10 rubber inserts with the significant weight of the shaft and sheave will not cause any significant stiffness variations or affect observed bearing motion in test results. Since the flexible coupling is designed to run continuously within the offset range of the adjusting screw and the fan load is well within the load capacity, the five-element coupling is adequate to drive the fan and fan shafts and does not introduce a variation for the demonstration.

The upper and lower sheave diameters for the diesels are different to provide respective airflow requirements for the engines (shown in comparative parts table that follows). The belt length is the same for the Joliet and Oyster Creek diesel units, however, the MP36 uses an 8 belt matched set and the Oyster Creek MP45A uses a single 8 ribbed belt that has a "web" connecting the ribs. Since the 8 ribbed belt used at Oyster Creek has superior ruggedness and reliability, ESI supplied and installed an identical belt assembly at Joliet to more closely simulate the belt drive of the Oyster Creek MP45A diesel units.

The upper foot of the pillow block bearing was drilled lightly and ground to simulate the depression found on the Oyster Creek bearing assembly under the adjusting screw contact area. The actual depression created on the test-bearing surface became slightly deeper than that on the Oyster Creek bearing (1/16" OC and 1/8-3/16" Joliet). Review of the motion and forces at the bearing depression on the monitors during the test allowed a determination that there was a potential influential factor as discussed in Attachment 1.

The actual bolts that were installed in EDG 1 during the event were used at Joliet to mount the pillow block bearing to the pedestal. The bolts are 4 ½ inches long versus the 3 ¾ inch bolts used in the MP36 unit due to the variations in the bolt step dimensions of the bearings of the MP36 and MP45A lower fan shafts. These bolts were used since they were cadmium plated with an extra flat washer on the nut end and it had been viewed as a potential factor that could cause more rapid loosening of the hardware during the test.

A steel guard was welded in place for equipment and personnel safety only to restrict the shaft movement to less than 1 inch if it should pull away from the pedestal violently during the test such as for fan belt or other equipment failure. During the first test demonstration, the shaft was contacting the guard in this position. The guard was moved to the right an additional 1-inch to prevent restriction to actual shaft movement.

The radiator-cooling fan is the same for each unit although the sheave sizes provide 571 rpm for the MP36 and 637 rpm for the MP45A. However, engine temperature controls are different between the units. The MP36 unit uses an "AMOT" thermal proportioning valve (four internal thermostatic valves) to proportion coolant flow and maintains engine temperature around the nominal element setpoint of 160 deg Fahrenheit. With the MP36 arrangement, the inlet cooling air louvers are fixed and the fan shaft load is constant. The MP45A uses an inlet cooling air louver that is position modulated to proportion the air flow across the radiator assemblies. A thermistor in the water manifold controls engine temperature around a nominal setpoint of 175 deg Fahrenheit by actuation of the louvers in the open or closed directions by means of an electric motor actuator. The fan shaft load for the MP45A constantly cycles from louvers closed (nearly zero air flow) to full open (approximately 125,000 CFM) with the unit in full load operation. The significance of this variation was recognized on posttest review and is discussed further in Attachment 1 and is further analyzed in the MPR reports in Attachments 3,4 and 5.

The following focuses on the Fan Shaft assembly and related components to assess the relative comparison of the two MP units as a basis for the documented simulation. EMD parts catalogs of varying vintage show that many of the parts are the same. Some parts are different in order that they meet the specific needs of the unit's cooling capacity requirements. There have been some upgraded parts made available for the MP45A units (that were installed in Oyster Creek diesels) while no improved or changed parts are listed for MP36 units and there is no evidence of upgrades performed on Joliet units.

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The following table shows the comparable Fan Drive component/assembly parts:
(* = Presently used parts for Oyster Creek MP45A diesel units)

Component	MP36 Part		MP45A Part
1. Engine Aux Drive Coupling	8345709	orig.	8345709
		newer	8323180*
Note: The only difference is newer has 10 rubber flex joints vs 5 for orig. part.			
2. Lower Fan Drive Shaft	8262069 (8'8 5/8")	orig.	8262069
		newer	8408113*
Note: Newer has 10 holes for rubber joints and uses ½" key for lower sheave.			
3. Lower Pillow Bearing	9416979 (SKF 479215-215)		9428697 (SKF 476215-215)*
Note: Casting differences and bolt step is 1 ¼" for MP36 and 2 ¼" for MP45A.			
4. Pedestal Assembly	8262158		8262158*
Note: Minor variations in construction but dimensionally they are the same.			
5. Adjusting Bolt & Nut	8262449 & 219753		8262449* & 219753*
Note: Identically the same parts.			
6. Lower Sheave Assembly	8301755 (12.5" OD)	orig.	8301755 (12.5" OD)
		newer	8444353 (15.0" OD)*
7. Upper Sheave Assembly	8301756 (19.7" OD)	orig.	8367634 (16.0" OD)
		newer	8394107 (21.2" OD)*
8. Fan Belt Assembly	8301757 (8 V belts)	orig.	8301757 (8 V belts)
		newer	MKN51840 (8 ribs)*
9. Upper Fan Drive Shaft	8262142 (4'9 ½")	orig.	8367803 (5' 9 ½")
		newer	8441753 (5' 9 ½")*
Note: Original shaft was a hollow tube. Newer shaft is a solid shaft assembly.			
10. Upper Pillow Bearings	9416979		9424990 & 9424991*
Note: Casting differences and bolt step is 1 ¼" for MP36 and 2 ¼" for MP45A.			
11. Upper Shaft Pedestal	8269647		8367758*
Note: MP45A Pedestal is 1' longer due to longer Radiator Assemblies used.			
12. Radiator Cooling Fan	8366614 (10 blade-7' Dia)		8366614*
Note: Minor differences in fan mounting hardware used in each fan unit.			

The setup was as close a simulated demonstration as could be achieved. We found on review of the test results that there was a very good representation of what occurred at Oyster Creek with a few apparent variations that are shown in attached reports to make the Joliet diesel testing significantly conservative when compared to the actual event at Oyster Creek. Attachment 1 discusses a visual assessment in words of test observations to provide understanding of the motions and forces that were observed during the Joliet test. Attachment 2 contains the Engine Systems, Inc. (ESI) test report that includes specific test data and results along with post-test review of fan belt and bearing condition.

TESTS CONDUCTED:

Following return of the MP36A unit to operational readiness, a test run with the bolts properly torqued down was conducted at idle speed and full load to characterize the level of vibration for normal operation of the engine. Information gathered showed the Joliet unit was reasonably close (fan shaft was higher) to previous data from Oyster Creek units and within manufactuer's general guidelines (both units less than half of field values). With good data for vibration, we were able to commence the demonstration testing.

The first test was conducted with both bolts backed out ¼ inch on the lower fan shaft pillow block bearing. During the start and during acceleration from idle speed to full speed, the belts were heard squealing with some slippage, however the upper fan shaft (which was being monitored for speed) attained full normal speed through the test. Very little motion was observed at the bearing and there was no belt slippage after the unit stabilized at full load. The vibration profile was taken to determine range of shaft motion.

The second test was conducted with the lower bearing bolt removed and the upper bolt backed out ½ inch to simulate the as found conditions after the EDG 1 event. This would commence the actual demonstration test as planned. As the unit rose to idle speed, there was immediate movement of the bearing lower foot and significant belt squealing with an odor of burnt rubber briefly. The diesel logic for a normal start holds idle speed for 90 seconds and rises to full speed, synchronizes and runs up to full load. After achieving full load the motion of the lower bearing increased to about 1-¾ inches at the lower foot of the bearing and approximately 1-inch deflection at the shaft. The shaft and bearing were alternately striking the guard (1 inch clearance allowed) and banging up against the pedestal at an estimated frequency of 100 strikes per minute. There were one or two periods where the deflection appeared to be reduced significantly but then returned after a short period of time. Upper fan shaft speed was observed initially at about 371 RPM and gradually fell to approximately 320 RPM versus the normal speed of 571 RPM. Lube oil temperature appeared to stabilize but at a higher level of 223 instead of the apparent normal of 190 to 200 degrees F. After a 20-minute run it was determined that the test would be terminated since the guard was restricting the shaft/bearing lateral motion. This did not meet test criteria and the test could not be considered valid under such conditions.

The guard was moved over in the direction of shaft travel about 1 inch further to allow 2 inches of movement for a subsequent test. It was observed for the event on EDG 1 that the bolts loosened while at full load of the unit. Therefore, a jumper was installed in the controls to bypass the 90-second idle period and allow a faster run up to rated speed and load. The fan belt was inspected and found to be significantly glazed for 2/3 the distance from the tip of the ribs back toward the web (area in contact with the sheave grooves). A loss of approximately 1/32 to 1/16" rubber from the surface was due to the belt slippage and squealing that occurred previously. There was some loss of tension observed so the belt was retensioned per methods used in Oyster Creek maintenance prior to restarting.

At 6:22 PM in the evening the test was recommenced. Similar conditions existed during this test with the shaft movement slightly greater than ¾ inches and the lower bearing foot about 1-½ inches. The upper fan shaft speed was about 370 RPM initially and falling to 307 RPM over time. The bearing was consistently striking the pedestal with the same frequency of 100 beats per minute and an occasional whiff of burnt belt rubber in the air was apparent. There were additional points where the movement of the bearing was reduced and then picked up once again. Due to heavy vibration in the fan structure, the optical speed pickup was intermittently losing signal but was reading the same lower speed of the fan shaft when readings could be taken. One hour and 25 minutes into the test, a generator trip occurred due to a reverse power condition found due to the main Fuel Tank valve being closed. The engine remained running at idle speed and the unit logic was reset, the generator reloaded and the test continued. It was judged that the change in speed of the unit with momentary drops in engine temperature and additional belt squeal on re-acceleration would have insignificant impact on the test. After six hours operation under these conditions, an engine trip occurred due to high engine water temperature (trips at about 205 deg. F. after a time delay). It was observed that the engine oil temperature rose at a rate of 1.0 deg Fahrenheit over a 10-minute period to 245 deg at the time of the trip. The test was terminated and results were reviewed.

CONCLUSIONS:

The series of tests demonstrate that, even with bolts in the condition discovered on EDG 1 at Oyster Creek on May 17, 2004, there was no impediment to the starting and loading functions of the diesel. Thereafter, at least six hours of operation has been demonstrated and possibly some portion of an hour longer since no automatic oil trip signal would be permitted for the Oyster Creek diesel. Oyster Creek controls have an automatic bypass of all engine temperature trips for emergency mode engine starting. Oil temperature would be allowed to rise as high as 250 to 260 degrees Fahrenheit where it is assumed oil will begin to breakdown with probable bearing and/or engine failure, with continued engine operation.

Further review of test results and a reassessment of the test demonstration factors that show similarity or dissimilarity to conditions during the Oyster Creek EDG 1 event indicate the probability of an even longer operating period. The discussion provided in Attachment 1 describes the motion of the shaft and bearing as observed on the remote video monitors that were recorded during the demonstration. The action described as belt "slipping" appears to be influenced by the magnitude of shaft bearing block movement, causing loss of belt tension and by belt heating, glazing or loss of rubber observed in the test. The belt "skipping" action appears to be created by the waves of excess belt slack traveling between sheaves and momentarily lifting the belt off the respective sheaves causing the speed and vibration trace variations observed in the test data. The vertical alignment of the sheaves and lateral alignment of the lower fan shaft may have an effect on the magnitude of the bearing movement. These variable factors and observations from the Joliet test showed the need to provide further analysis of apparent differences for the response of EDG 1 at Oyster Creek. MPR was contracted to provide this analysis.

ATTACHMENT 1
FAN SHAFT, BEARING AND BELT MOTION

OBSERVATIONS:

During the tests there were two video cameras positioned to view the lower shaft bearing with one camera at each side of the assembly. From these positions the shaft, bearing, lower sheave and fan belt were observed, however, the upper sheave was not in view since we did not anticipate the extent of belt motion at the upper fan sheave. Analog parameters of upper fan shaft speed, water temperature, lube oil temperature, lube oil pressure and three axis vibration at the top of the bearing pedestal were monitored. Some difficulty was encountered with the optical shaft speed sensor, which was being affected by the high levels of vibration present yielding intermittently accurate signals.

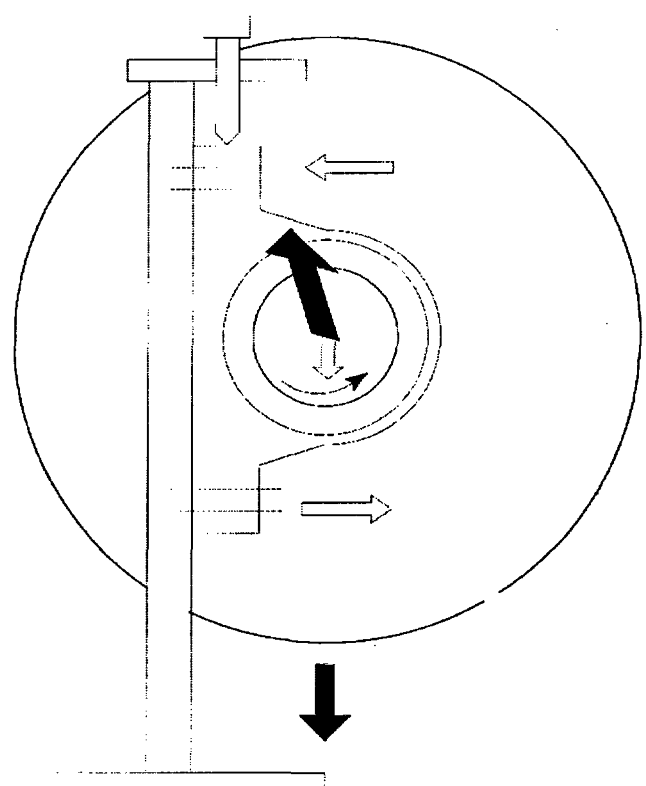
With the bottom bolt removed and the top bolt backed off ½ inch from the lower shaft bearing, there was an immediate defining motion of the lower shaft and bearing. There was movement of the lower foot of the bearing away from the pedestal (to the right side) of approximately 1-½ inches while pivoting on the adjusting screw at the upper foot of the bearing. The shaft was moving about ¾ inches which will be about half of the lower bearing foot deflection since the bearing is centered in the 10.5 inch length of the bearing housing. These approximate distances were estimated from the visual scene on the video monitors with knowledge of the dimensional properties of the bearing assembly. The motion was regular with a consistent 100 beats per second (beats/bangs against pedestal).

Motion of the bearing is due to shaft motion since the bearing is press fit on the shaft and locked in place with the "grub" screws on the inner race collar. Movement of the shaft will be due to forces imparted by the fan belt or by gravity with the weight of the shaft, bearing and sheave during periods when fan belt tension is lost. With the bearing fixed in place and bolts torqued properly, there is no possible shaft movement laterally and belt tension remains constant. In this condition, with the shaft rotating in the counterclockwise direction, there is a resulting force upward on the shaft in an approximate 11:00 direction due to the left side belt resistance force generated from fan drag transmitted to the upper sheave. This force is slanted to the left side in the direction of the left side fan belt because the driving lower sheave is turning in that direction placing the full torque on the left side belt surface. An additional pair of forces act at the bearing due to shaft rotation (CCW) that place a force to the left at the upper bearing foot and to the right at the lower bearing foot. This pair of forces is somewhat smaller in magnitude since the bearing has very little friction to resist the shaft rotation force. The shaft vibration will also cause a force to act from the center of the shaft due to the imbalance, displacement or velocity present as a result. This force is described as a vector rotating 360 degrees at a frequency of 900 rpm (with multiples) as the lower shaft turns at full speed. Baseline vibration levels for both units were similar with Oyster Creek approx. 0.9 in/sec and Joliet approx. 1.6 in/sec (note: some energy comes from the engine through the floor structure). Review of the relative magnitudes of this force shows it would be small compared to the principle forces observed in the test and the team concluded it was not a significant factor in the demonstrated shaft motion.

Except for rotation, the pedestal, the bearing hold down bolts and the adjusting screw restrain all the forces that would cause shaft movement in the normal bolted condition. With bearing hold down bolts in the test directed positions, the lower shaft and bearing assembly are no longer restrained in the right side lateral direction. The pedestal will restrain physical motion to the left side and will be the source of the "banging" heard during the test runs. Review of forces present in the assembly with bolts in test positions shows that the primary forces acting to describe shaft motion are the forces imparted by reactions of the fan belt to the lower (and upper) sheave as well as gravity on lower shaft.

Forces were seen to change as the motion of the bearing moved to the right away from the pedestal and back to the left against the pedestal. As the bearing moves to the right, the shaft moves to the right with a slight lift in the vertical direction that removes some of the belt tension due to the adjusting screw at the upper foot of the bearing. The upper foot of the bearing was observed on the monitor to move as much as ¼ inch away from the adjusting screw until forces changed to pull the bearing back against the pedestal. As the

FIGURE 1
NORMAL CONDITION FORCES



bearing moved back against the pedestal, there was a distinctive movement upward to place the upper bearing foot hard against the adjusting screw and the bearing housing flat against the pedestal. This portion of the movement was clearly restoring the position of the bearing to the "normal condition" as shown in Figure 1 above. The restoring force is the 11:00 vector force that signifies left side belt tension returned and the force pulling the bearing away from the pedestal was absent at that moment.

During the demonstration, the belt was observed "flapping" vigorously at right side of the sheaves and nearly the same motion at the left side. Excess belt slack was the cause for this action. The belt slack resulted from the vertical movement of the shaft when moving to the right away from the pedestal. This "flapping" is the visual impression from excess belt length traveling like a wave from the bottom sheave right side up to the upper sheave right side and the portion then traveling down the left side back to the lower driving sheave. The deflection of the belt at the right side was as much as 5 inches (visual at the monitors). The deflection of the belt at the left side appeared to be less; however, the camera angle was not optimum for this view. The rate of the belt slack waves traveling up the right side belt appeared to be related to bearing motion that occurred at a repetition rate of 100 beats/second as seen on the monitor.

Review of the belt wave motion determined that as the wave arrived at the upper sheave it would lift the belt off the sheave until the remaining belt/sheave contact area was not sufficient to hold the belt. At this moment the belt would "skip" across the upper sheave by an amount related to the excess belt length. The lower driven sheave would pull the belt down the left side until the belt at the upper sheave regained contact to drive the fan shaft once again. It is probable that a momentary "slip" of the belt occurs as the friction forces allow the belt to re-establish contact on the upper sheave which accounts for the low grade "burnt rubber" odor continually present immediately near the diesel. The downward trend of fan speed was due to continued belt stretching during the test. The skip/slip action of the belt caused less belt length to be in contact with the upper sheave and thereby causes the fan to be driven at a lower speed than the sheave sizes provide in normal service. The reduction of belt length contact is equivalent to reducing the size of the lower sheave with lower fan shaft speed as a result and a corresponding reduction in fan air flow or engine cooling capacity. The effect on cooling capacity is the most prominent influential factor on engine performance under demonstrated conditions.

There was motion of the fan belt in a similar manner at the left side of the sheaves and it could be observed that some degree of skip/slip action occurred at the lower sheave as well. During a couple of brief periods during the test, the motion of the lower bearing decreased from approximately 1 ¾ inches to about ½ inch with a corresponding change or settling of belt motion at the lower sheave. At this point in time, it appeared that the skip at the lower sheave was absent and then returned after a few minutes. The fan shaft speed sensor was not available at that time to observe any change to resulting fan speed. During these periods there was an increased belt burn odor in the air around the site that seemed to dissipate after a few minutes indicating additional slippage of the fan belt.

As the belt regains contact with the upper sheave following a skip over the sheave, the right side belt slack is removed and there will be a force imparted to the right side of the lower sheave that is upward in line with the right side of the belt (approx. 1:00 force). This is the motive force that pulls the bearing and shaft assembly away from the bearing pedestal creating the described signature motion of the bearing. As the belt "slack wave" moves down the left side, there is a skip at the lower sheave followed by slack removal on the left side and restoration of the normal force condition. The normal 11:00 force vector returns the bearing housing up against the adjusting screw and the cycle repeats.

During the test run with bolts torqued normally, there was no belt slippage observed. This test was performed with the new 8-ribbed single web belt that is identical to the belt used at Oyster Creek. There was no change to the belt surface following this test run. The next test was conducted with both bearing hold down bolts backed out ¼ inch. When the diesel was started and during acceleration from 400 to 900 rpm after the 90-second idle period, the belt was heard squealing from serious slippage and then settled at full speed. Observation of the belt surface following this run showed approx. 1/32 inch of the rubber covering the cords of the ribs was lost (glazed) for approx. 2/3 the width of the belt (portion resting in V-groove of the sheaves). The web of the belt did not show any similar glazing at this time. The demonstration test was started twice with the bolts backed out to test directed positions. The first terminated at approximately 20 minutes into the run where the guard was contacted by the shaft and required the guard to be moved over for more space. The belt was retensioned to remove belt stretch that had occurred and the second run under these conditions produced the approximate 6 hour demonstration interrupted by the engine overheat trip. Following this test the belt was observed with additional slack and there was a more substantial rubber loss than before as expected. Belt cords were clearly visible at the side of the ribs and there was significant loss of rubber along the full width of the belt seen at the side. This indicated that some of the rubber had burned away from the inside of the web as well which shows the belt moved further into the Vgrooves. There was evidence of belt slack again that would result from belt glazing in operation and that was caused by the rubber loss effectively reducing the inside belt diameter. The final condition of the belt, however, showed that there was substantial construction that remained and it was likely; the belt could still be operated for an extended time period (see Attachment 2 and Attachment 3 belt reviews). It is significant that there were no signs of belt burn or slack found on Oyster Creek EDG 1 belt following the event and the three-week-old belt was placed back in service.

There are complex details of this motion that are not possible to quantify for more rigorous analytical presentation. However, it is concluded that this description of the notable events, data acquired and observations fully describes the motions and influential factors involved with the test demonstration as well as impact to engine operation. From this perspective we are able to provide the differential factors between the Joliet test engine and expected performance of the Oyster Creek EDG 1 with reasonable accuracy. The ESI report and EDG 1 event information allowed MPR to provide a calculational model that provides additional analysis of EDG 1 operation for the May 17, 2004 event.

COMPARATIVE REVIEW:

Operations reported in the log that an abnormal noise and vibration drew their attention to the problem on EDG 1 during the event. There were also comments of "banging" and bearing/belt looseness observed during interviews. The operators were able to continue engine operation for enough time to obtain the final surveillance readings while still at full speed and load. When the unit was given a normal stop signal, the generator auto unloaded, tripped the breaker, returned to 400 rpm idle speed and remained at idle speed for another 9 minutes until operators actuated the overspeed trip lever manually to stop the engine rotation due to observed abnormal conditions. No burnt rubber smell existed. The engine parameters recorded with the engine still at full load showed they remained in the normal band given in the surveillance procedure indicating no significant impairment to engine temperature had occurred at that point. While these conditions appear to show that the signature bearing motion was present as observed at Joliet, the engine parameters and physical as found evidence on EDG 1 suggest a wide variance in actual conditions.

During inspection following the event, the lower bolt was found completely out and on the floor behind the pedestal with the upper bolt backed out ½ inch (the nut remained with just full thread engagement). The "free float" position of the bearing had the lower foot approximately 1/8 inch away from the pedestal and the upper foot still in contact. There was substantial (almost normal) belt tension applied with the adjusting screw hard against the bearing housing. Deflection of the belt with the fingers showed essentially normal tension with about 3/8 inch deflection and the lower shaft could not be moved without extreme force in the lateral direction (actions by engineer during inspection). There were no signs of belt distortion, damage or any indication of belt burn or slippage. The bearing showed only a normal degree of grease drip at the side of the bearing that was the same as could be observed on the EDG 2 fan bearing (normal operation). After repairs were made on EDG 1, the PMT run showed no vibration frequencies present in the ranges that the bearing manufacturer suggests would be there for a worn bearing.

1. Sheave Alignment:

The vertical alignment of the upper and lower sheaves on the Joliet engine was measured (using a stone and string plum bob with a ruler) and found with the lower sheave 1/8 inch to the right of the upper sheave centerline when the bolts were torqued normally. When bolts are removed, the "free fall" position of the bearing lower foot was ¾ inches away from the pedestal (shaft is approx. 3/8 inches further away from pedestal) as seen in an ESI photograph with a ruler present in the picture. Some of this nominal movement may be due to additional misalignment of the lower shaft from the engine relative to the upper fan sheave, which could not be measured at the time.

The vertical alignment of the sheaves for Oyster Creek showed the lower sheave to be left of the upper sheave centerline by approximately ¼ inch as observed visually through the fan guard mesh screen that was in place. With the bolts in the as found condition, the "free fall" position of the lower bearing foot was 1/8 inch away from the pedestal (shaft is therefore 1/16 inch further away from pedestal) as observed visually during inspection.

These dimensional differences and "free fall" tendencies will produce a variation in the magnitude of force needed to deflect the bearing to the right when in operation with the test directed bolt positions between the two units. This variation can be seen as a change in the degree of movement or deflection of the bearing. The resulting impact will be that less loss of tension will exist and less belt slack will be created for smaller bearing/shaft deflections that in turn will cause higher fan speed and better cooling function than the test case at the Joliet station. Specific quantification of this factor cannot be made, although, we consider this variation as a significant factor among others in the determination of reasons for differences of the physical evidence found between the diesel engines.

2. Upper Bearing Foot Cavity:
The cavity or depression that was simulated on the new bearing upper foot surface became enlarged to a greater dimension than that found on EDG 1 as indicated earlier. The EDG 1 depression was approximately 1/16 inch deep at the center and extended around the adjusting screw point as much as 1/8 inch larger diameter. There was a casting mold remnant between the point of the adjusting screw tip and the pedestal that provided a slight rise in bearing surface above the level of the screw tip. The simulated depression on the bearing at Joliet Station was created initially using a ½ or 5/8 inch drill bit that was inadvertently drilled to a depth of about 3/16 inches deep. The edges of the hole were then ground out to the approximate diameter of the depression observed at Oyster Creek.

The impact of this unintended variation was observed on the videotape monitor as permitting the adjusting screw to move down into the hole when forces moved the bearing housing up against the adjusting screw and the screw was deflected to the hole. The result of this variation would be to decrease the belt tension more from the start of the test when the signature bearing motion began. This factor is moderately significant.

3. Idle versus Full Speed Operation:
The Oyster Creek event occurred with the diesel at full speed and load during operation for surveillance with fan speed or momentum fully developed and no apparent loss of speed that would start an elevated engine oil or water temperature condition. Necessity dictated that the Joliet unit was started from standstill, idled, synchronized and loaded. During the demonstration runs the engine never achieved full fan speed or momentum since the signature motion began immediately (initial 380 rpm vs 571 rpm normal). As the engine heated up (approx 30 min), the point where engine parameters appeared to stabilize was significantly higher level than normal due to slower fan speed and airflow.

This difference between the two units in operation is considered very significant. The difference factor will place the onset of the beginning of engine temperature degradation for EDG 1 later in time such that engine-cooling function would remain longer than observed in the Joliet tests. The engine may have operated for a period of additional integer hours before the difference factors arrived at the point of elevated temperatures observed occurring upon initial startup of the Joliet engine. EDG 1 as found evidence shows specifically that even with the bolts backed out the cooling system function was not yet impacted and was not imminently threatened by fan belt slack or skip present. We reasonably envision this factor to extend operation for a significant period of time.

4. Belt Heating, Slack and Material Loss:
Previous discussion shows that the EDG 1 belt suffered no significant slack or loss of rubber material at the time it was taken out of service during the event. It is apparent that there was some skipping of the belt to produce the audible noise/vibration the operators observed and that there would therefore be some fan shaft speed drop. This evidence can be seen as supporting a conclusion that the bearing motion was smaller in the EDG 1 case and that there was less belt slack and lower rate of heat input to the belt. Therefore there would be less magnitude in the excess belt length wave that causes the belt skip and less speed drop at the fan resulting. A smaller skip magnitude and greater fan speed causes the force pulling the bearing away from the pedestal (the 1:00 force vector) to create less of the noted deflection at the bearing. This set of cyclic reduced parameters appears to be adequately supported by the evidence and would produce a factor that would lengthen the time for belt heat glazing or slippage to begin to threaten engine operating parameters. This factor is considered very significant and would add many hours to EDG 1 run time.

5. Fan Shaft and Sheave Torque:
The MP36 cooling water system has been described as water flow proportioning using thermostatic valve elements in the AMOT valve that controls water flow to the radiator. The fan louver blades are fixed and the airflow remains constant with fan speed. During the Joliet test it was observed that the lower fan speed developed an elevated temperature of water (and therefore engine oil) from the very start and throughout the demonstration. The fan shaft torque also remains constant with the given fan speed and this will in turn determine the frictional forces, belt forces and shaft loads that control bearing motion.

The MP45A cooling water system does not have water flow proportioning and instead uses variable airflow across the radiators to cool jacket water. The fan louver blades are position modulated by means of an electric motor actuator that receives the open/close signals from a Barber-Coleman temperature controller (with thermistor input). While the fan speed remains constant, the fan shaft torque varies with the louver position and the resulting variable airflow fan load. There is an inherent mismatch in the response of the temperature controller and the lag time for water to be cooled by increased air flow that causes the louvers (and air flow) to continually cycle full open and closed in operation. When the louvers are full open, there is full fan airflow with associated shaft torque and when the louvers are full closed there is no airflow with minimal inertial shaft torque. The forces that control the bearing motion will therefore vary as the shaft torque varies.

Performance of the MP45A louver controlled cooling system monitored over the years at Oyster Creek shows that the open/close louver cycle is from 2 to 3 minutes long with seasonal variation. The louver actuator drives full stroke in approximately 45 seconds in either direction and during the remaining cycle time the louvers are in the open or closed position. The dwell periods in the open and closed position are generally equal and can be viewed as the relative margin of cooling capacity. The response of this system to elevated water temperature levels will be to leave the louvers open for a longer dwell time. The stated existing dwell times offer as much as 100% more time to restore an elevated water temperature excursion (as from slower fan speed) to the normal 175 deg F. setpoint.

This compensating capability means that engine temperature parameters and cooling function are NOT immediately degraded by fan shaft speed drop as for the MP36 unit. In addition, the continual cycling of fan load from full to minimal with substantial time in each state can be viewed as alternately applying and removing the forces causing the signature bearing motion. The most probable result during the period when minimal shaft torque exists will be return to normal fan belt contact and fan shaft speed due to minimal bearing motion. Fan speed drop is then only observed during approx. 50% of the cycle.

The difference in cooling system controls and impact to extended engine performance for the stated condition shows this factor to be extremely significant for the MP45A unit. The two factors discussed are viewed as potentially extending engine operation by a few to several hours or longer since the factors also reduce belt heating and slack that retard further bearing motion. The magnitude of the belt skip/slip action is reduced as well.

CONCLUSIONS:

There are complex proportions of these factors and resulting force magnitudes that describe the ultimate performance of the MP45A diesel versus the Joliet MP36 diesel. Engineering review of the given variations with knowledge of the physical operation of the mechanical systems involved shows the cumulative result of these factors will be considerably positive to extension of the MP45A response beyond the Joliet test. Our conclusion is that these factors conservatively account for a very significant extension of EDG 1 operation beyond the demonstrated 6 hour run conducted at the Joliet plant.

The further analysis provided by MPR in their reports (Attachment 3,4 and 5) confirms impact that these variations would have for EDG 1 operation with loose bearing bolts. Conclusions of the report include the following:

1. Fan speed would only drop to 90% of rated with air flow of 112,400 SCFM. This air flow produces normal engine oil temperature and no degradation of engine function.

2. For the conditions found in item 1, the belt life would have been one month although near the end of belt life fan speed would likely degrade further with more belt slippage.

(The ESI report, Attachment 2, shows the bearing would not limit extended operation).

3. If fan speed drops to 55% of rated (which is the approximate fan speed observed during the Joliet test), the resulting engine lube oil inlet temperature will rise to 239 deg. F. remaining below maximum limits with no degraded engine function. There will be no engine trips since the temperature trips would be bypassed on EDG 1.

4. MPR Associates concluded that the EDG #1 operation would not have been limited by increasing engine jacket water and lube oil temperatures. EDG #1 would not have prematurely been shut down by a high jacket water temperature trip.

Oyster Creek concludes that the Joliet run time of 6 hours is confirmed as valid and that Engineering analysis shows EDG 1 would not have lost capability to perform its intended safety system function.