Diesel Generator Set Vibrations
Analysis and Mitigation

ARNE LINDHOLM
Foreword

Emergency diesel generator sets are vital parts of the safety system in a nuclear power plant. Problems have been experienced with vibrations, causing unnecessary wear and tear of the machines and costs due to safety related reduction of the power stations load. There are currently no well proven standards for allowed vibrations levels in emergency diesel generator sets covering all aspects of the complex system. The report explains what is special with combustion engine vibrations, gives a review of four nuclear power stations collected experiences and offers purchasing rules based on actual historical experiences. If used, it may improve the reliability of the existing and new sets considerably.

In this project, proactive maintenance experienced Arne Lindholm from PME Predictive Maintenance Engineering, has performed a mapping of historical vibration problems in the emergency diesel generator (EDG) sets in the nuclear power plants in Forsmark, Olkiluoto, Oskarshamn and Ringhals. He has worked with diagnostic analysis and practical mitigation design in EDGs in both nuclear and other applications for more than 50 years.

This project has been carried out within the framework of the Energiforsk Vibrations research program. The contributors to the Vibrations Program are Vattenfall, Uniper, Fortum, TVO, Skellefteå Kraft and Karlstads Energi.
Sammanfattning


Baskraven är desamma som för alla normala maskiner, men här formuleras de tillägg för det som uppstår i en dieselmotor; vridmomentpulsar i vevaxeln från varje cylinder och flödespulsation i rörledningar.

Rapporten sammanfattar baserat på historiken hur man bör hantera:
- resonanser i aggregatet med sina delar på plats i anläggningen
- vridmomentpulsationen i motorns arbete som beror på fyrtaktsmotorns funktion, naturliga krafter från ofta även turbulent gas- och vätskeflöde
- obalanser från tillverkning, revisionsarbeten och lossnande delar.
- upprustning mellan motor och generator, såväl statiskt som dynamiskt
- generatorns naturliga inre elektromagnetiska kraftväxling vid 100Hz


Resonanser som exciteras vid naturliga impulspjässor tas inte alls upp. Betydelsen av att använda dämpning för att begränsa vibrationer saknas helt. I många olika industrier utanför NPPs används resonansbegreppet och många olika former av dämpning i stor utsträckning. Detta tas därför även upp i rapportens sista del.

Mot bakgrund av de mycket höga tillförlitlighetskrav som ställs på alla nödaggregat bör man begränsa urvalet av nya aggregat till att helt undvika aggregat som är prototyper.

Rapporten är avsiktligt hållen på en nivå avsedd för den som söker enkla tekniska förklaringar snarare än information baserad på aktuell skolmatematik. Den vetgirige kan mycket enkelt söka vidare på webben och finner där bra material på alla nivåer.
Summary

Emergency power for all electric facilities such as hospitals and power plants is a special field of interest, partly because of the extremely high demands on the starting safety and operating reliability. The Swedish (1971-) and Finnish (1975-) nuclear power plants that have been in operation for over 40 years, starting from more than forty such units developed by the then ASEA Atom with French and Swedish motors and generators. History of the selection, purchase, installation, commissioning and maintenance from the vibration point of view has been collected in a neutral unit independent description of well over 100 aggregate year experiences. It can be used both for continued operation and to formulate requirements for the acquisition of new units. It gives a broad basis for the assessment of the special vibration aspects related to a diesel engine as the prime mover. Based on the historical experiences it finally gives a detailed appropriate guidelines and suggested rules for emergency power units.

The additional requirements for a diesel engine are focusing the torque pulses from each cylinder.

The report concludes, based on the history, how to deal with:

- Resonances in the unit with its parts in place in the plant
- The torque ripple in motor operation due to the four-stroke engine properties, the natural forces of the often turbulent gas and liquid flow.
- Imbalances from manufacturing, outages and loose parts.
- Alignment between the motor and generator, both statically and dynamically
- The generator’s natural internal electromagnetic force at 100Hz

Current providers are now ISO certified. For the normal characteristics of a machine, which is important for its vibrations and therefore often the reliability and longevity, are ISO standards as suppliers undertook to follow. Especially for the diesel engine related aspects ISO standards lacking which agrees well with the actual outcome. Considering only the vibrations measured on the main bearings but refrains from setting clear boundaries and raises the importance of vibrations in the secondary help system. It leaves to buyers and sellers to agree on the vibration characteristics and level limits.

Resonances excited at the natural impulse frequencies and the importance of damping to mitigate vibration is widely absent. Many industries outside NPPS use the resonance concept and many different forms of damping widely.

In view of the extreme demands on the reliability of all emergency power units, the importance to refrain from considering prototypes is widely underestimated. This means that the selection of new units should completely avoid considering prototypes.

The report is deliberately kept at a level intended for those looking for simple technical explanations rather than information based on the current school mathematics. The inquisitive can very easily search on the web and find where the good materials at all levels.
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1 Background

The nature of a vibration related problem in an EDG is often complex and may sometimes cause long periods of shut down or at least reduced power due to very strict functional requirement rules for the plant, leading to large losses of income. Using the now well established modern knowledge in this field from many different areas of engineering, good methods are available to practically avoid future loss of reliability. Especially during power up rate projects, introduction of new equipment, for example complete diesel generator sets.

Vibration problems in nuclear power plants can originate from different sources. For instance, in the diesel generator set, resonance excitations in installed piping, support equipment and simple structures such as turbocharger brackets. It is therefore important to take this into consideration when buying new diesel generator sets. Good reliable requirements and testing methods need to be agreed upon.

The objective of this project is to assemble the historical knowledge and experience in the area of diesel generator set vibration related issues. Study in some depth how they were investigated and mitigated successfully. Information has been retrieved through interviews with the staffs at Forsmark, Olkiluoto, Oskarshamn and Ringhals. Technical reports from these nuclear power plants from more than 40 years have been studied.

Since most engineers today have a good knowledge in normal rotating machinery vibrations, this has been almost excluded in the report. Therefore, to avoid making a new textbook, most theory has been avoided.

This report is intended to create a basis for elementary understanding of phenomena related to diesel generator set unique vibration properties. A stepwise action procedure for understanding, correction or mitigation of alarming diagnostic results is included.

In addition there is also a detailed recommended set of rules for a successful procurement of new emergency diesel generators, to be installed and used in any plant with strict reliability requirements.

A typical EDG in one of the Swedish NPPs.
2 Introduction to EDG Sets

2.1 The Operating History From Past 40 to Less Than Some Years

The four NPPs have been equipped with seven different makes of EDG in the range 1-3 MWe. The oldest sets have been used in several hundred short start and load operations as functional verifications and almost no intentional real emergency use at all. The most recent sets of experience are a broad variety from smooth perfect operation to long periods of ongoing installation work to reach safe and acceptable EDG active operation.

The application of all EDGs is intentionally many thermal cycles, short full load time and relatively very small total number of operating hours. Using an IC engine this way is presenting the designer with demands outside what is normally found for IC engines in power stations and ships. This report is a summary of the practical results of using these during the recent four decades of operation as found during visits in the NPPs during 2015 and parallel investigation in some thousand pages written reports from actual diagnostic work with the oldest installations.

2.1.1 Sorting Historical Data without EDG Identification

The sections below are a listings and descriptions of the actual cases but without any reference to a specific case or make. The aim is to create a knowledge base of the history with a supplier and user neutral description and look forward to an optimal maintenance and purchase of new EDG regardless of usage. Apart from nuclear power plants, hospitals and IT Service are examples of applications where the demand of quick start and long reliable operation is just as high. In the section 2 this information is used to formulate a set of paths for keeping the EDG operational and also a set of rules of requirements acquire new sets with terms to be used in the contracts.

2.1.2 Experience from More Than 100 Operation Years

The collected practical experience has shown that the expected long term quality of the EDG designs have not met demands in general. Instead, the results have been fluctuating. It is correct to claim that the expectations and demands for very high reliability are mandatory. Apart from long term operation of diesel engines in ships and similar applications, the emergency standby generator sets have a range of special consideration that must be met for a long term life.

The investigation behind this report was therefore aimed to find as much practical information as possible from the history of the four running NPPs in Sweden and Finland with a similar source of equipment originating from historically ASEA Atom with 45 EDG sets from ASEA, Nohab, SACM, Hedemora, MTU, Caterpillar, Wärtsilä and Mitsubishi. The gathered information has been sorted into common aspects common to all engines and generators regardless of makes and ages. Each aspect of concern is described with as little theory as possible since today’s open engineering world is already more than full with such information.
2.2 THE EDG SET HAS A LOT IN COMMON WITH MOST ROTATING PROCESS MACHINES

Most general rotating machinery in industry have a prime mover and a driven load as for instance the most common pumps, ventilators and gearboxes. All those machines normally have in common a work with a constant torque. The general procurement process, the first installation and the long term diagnosis for a proactively supported maintenance to reach a long life cycle for all those machines is well spread and known.

The NPPs have been using a common internal specification which is very similar to international ISO standards (basic examples are for balancing ISO 1940 and vibrations ISO 10816) as well as parts of the earlier procurement rules collected in SSG3030 of the Swedish SSG standardization system for process plants, also based on the same ISO Standards.

The visited NPPs all have a system for collecting and trending vibrations of a varying completeness and depth of all the rotating plant equipments.

Since the EDGs only do a test run periodically, to include a regular routine vibration measurement during the test run has challenges. Even if only the total levels are measured, the very short thermal transient such as the relatively very short time on full load is not an optimal basis for a reliable trending. Large faults will be readily detected. With growing experience, the technician performing the test run has a good fault detection capacity almost as the traditional motorman onboard a ship enroute. A pending malfunction is often heard and felt instantly opening the door. Additional systematic vibration analysis has the great advantage to often pinpoint with surprisingly good precision exactly what is wrong already as the experienced visitor may just feel something is a bit different.

The number of EDG in a group of the same make and size is an important information source for a systematic routine monitoring. Using a common database system and using computer power to compare and trend and identify growing errors would easily increase the detection and identifying capacity many times if just really used.

Since almost any fault in an EDG set has a very large repair cost included, the price for a systematic approach has a remarkably strong payback. It can be estimated in just hours of a year total accumulated repair costs saved. The focus on the importance of complete documentation of each fault and all actions of repair and maintenance work undertaken has really been strong in this collecting work. Sharing all collected know how between NPPs or EDG users in general is so far almost nonexistent. This is unfortunate, since too many times it is possible to detect in the investigations that the same expensive damages occur in parallel over years in different NPPs. A common knowhow has a high cost saving potential.

2.3 THE IMPORTANCE OF A SIMPLE LEVEL VIBRATION STATUS SCREENING

Each engine and generator with parts is in an individual functional condition. This is the result from many sources such as systematic design faults, individual wear, and actual assembly tolerances and so on. The make and the design variations reflect more of a general starting point which indeed can be a good or a bad one. A given fixed level in vibration at any point and direction cannot be used as a threshold to accurately call a fault condition. When a measured level is above what has been traditionally seen as reasonable, the risk for unexpected fault developments and even direct breakdowns of parts or the full EDG is just gradually higher. The investigation for this report has given
good signs that all too often basic common sense judgment is impaired by other much stronger demands. Examples are saving money and time far outside the actual EDG function in the NPP.

The fault condition should never be left without careful investigations. There is always a need for at least a frequency analysis to begin any fault identification from traditional points and directions with points covering parts with high subjective levels. In most cases this has proven very effective to diagnose and suggest actions of mitigations or even fault condition elimination.

Active search for other end users and makers own experiences have not been found. Already small efforts during the autumn 2015 gave very good additional system knowledge. One example is the fact that EDG engines from SACM are also used in a few hundreds of French NPPs. Maintenance and fault information have historically been reported by French authorities to Sweden. Although for unknown reasons only to the NPP site Studsvik. No external facts have been used or included in this report.

2.4 MORE ON HOW THE REPORT IS ARRANGED FOR THE READER

All machines have many traditional vibration sources in common. The reader is supposed to have a good thorough knowledge of all kinds of power plant machinery excluding the current theme emergency diesel engine sets. Although sometimes also within EDG sets, plain normal rotating machine behaviour is found, we will mention and discuss them, but mainly focus on such vibrations which only come from EDG sets.

2.4.1 Section 3.2 is EDG characteristic Vibrations

The Section 3 is a first introduction to what is unique in an EDG where the Section 3.2 is a simplified listing of what is found in the vibration frequency range. The listing has a follow-up with more explanations and a bit more details.

2.4.2 Section 4 is about torque in an combustion Engine

Section 4 is a logical stepwise introduction to the background of the dominating property of the combustion. The purpose is to explain the phenomena for the person who does not have pre-knowledge.

2.4.3 Section 5 is about the used Vibration Measurement Methods
2.4.4 Section 6 is about actual Mitigation Work Performed
2.4.5 Section 7 is a Summary of applicable ISO Standards
2.4.6 Section 8 is the Aspects of Procurement Guidelines

The contents are based on the investigation experiences which can be used for primarily new EDG sets, but also supporting activities to improve the current reliability of existing EDG sets.

2.4.7 Section 9 sums up an Overview of Conclusions
2.4.8 Section 10 has a few Suggestions for further Work
3 Unique to EDG concerning Vibrations

A diesel engine simply adds a range of additional aspects related to its own internal properties such as pulsating torque. It generates a pulsating stress to the crankshaft, the coupling and driven generator as well as the stationary engine cylinder block and mounting system all the way to the generator stator.

A listing and brief description of all historical reliability but carefully kept as anonymous problems can be found in the section 3.1 Vibration related causes below.

Very basic EDG examples are failure to start when initiated, failure to synchronize the generator to an already live power line and increase load to full power or unstable or even broken operation. Since these faults are often a secondary fault, caused by elementary vibration related problem before, it is not detailed separately in depth. In the section 3.2 below, at first starts looking at what a simple vibration measurement can show during a normal test run.

The next section 4 is describing how a fault has shown up when identified and mitigated. It is based on the collected history of performed or planned actions on the EDG during the years. Explaining briefly why a measurement point shows of vibration level of concern and what vibration content should be an issue to correct.

The special aspect of direct diesel engine vibrations is the fluctuating torque. The Section 4.5 below is trying to explain this property in a simple manner. It covers the sources of this torque pulsation in engines with 1 cylinder and up to modern multi cylinder EDG engines.

3.1 VIBRATION RELATED CAUSES

A linear or torsional vibration motion problem in a diesel engine can be attributed to basically eight sources in order from the simplest to a bit secondary generator issue:

- Misalignment of engine and driven equipment (rotors in line not well concentric inside tolerances in parallel and angle).
- Unbalance of rotating parts, engine, coupling, torque damper, generator rotor parts. Unbalance due to traditional unbalance not related to combustion forces.
- Resonance from structural mass (weight) and stiffness (rigidity) combinations, all parts. Offering unwanted multiples of amplification of dynamic forces.
- Torque reaction of the rotor line as well as combined aggregate structure. Due to the attempt from each firing cylinder to twist the rotors with a strong turning push and the counter reaction from the stationary parts along them.
- Cylinder misfiring. For a multi cylinder engine, all cylinders are expected to offer as precisely similar torque forces as possible. A cylinder with a reduced or no existing torque force will cause not only reduced total power but also through gradually failing torque balancing functions often initiate damaging wear. Pending the level of fault, this can affect a broad train of frequencies which can start at half the speed for a typical four stroke engine.
- Combustion forces. Mostly pulsation forces in the piping system. Both liquids and gases. Often a low frequency and many harmonics due to distortion of pulses which prevents a sinusoidal force at a single frequency
- Unbalance of reciprocating parts as well as driven generator rotor parts. The vibrations are normally 1xRPM of the rotor, and can be the reason for different
levels in horizontal/vertical direction or even high in axial direction due to
dynamic moments. Compare to car wheels with two planes for balancing.

- Electromagnetic forces at 100 Hz. From using 50Hz power lines. When the exciter
  has other number of poles or even thyristor driven, the forces are normally the
double fundamental frequency. A good looking distortion free sine wave is often
present.

3.2 SIMPLIFIED SOURCES OF THE VIBRATIONS

This is a listing and short form explanations of the seven common findings.

<table>
<thead>
<tr>
<th>Vibration Characteristic</th>
<th>Correctable Causes</th>
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<tr>
<td>1 One component vibrates</td>
<td>Wrong mounting or use (example: flow) of component</td>
</tr>
<tr>
<td>2 1/2 x engine rpm (one-half order)</td>
<td>Misfiring of one or more cylinders. Regularly or occasionally.</td>
</tr>
<tr>
<td>3 1 x engine rpm (first order)</td>
<td>Unbalance (missing balancing weight or assembly run out between rotor parts)</td>
</tr>
<tr>
<td>4 2 x engine rpm (second order)</td>
<td>Misalignment, out-of time balance weights, also crankcase oil overfill</td>
</tr>
<tr>
<td>5 1 1/2, 2 1/2, third and higher orders</td>
<td>Normal cylinder or higher orders combustion (normally not correctable)</td>
</tr>
<tr>
<td>6 Large vibration motion (often directional, radial or axial)</td>
<td>Unwanted resonance excited by EDGs normal forces. Dominating direction</td>
</tr>
<tr>
<td>7 Motion increase with torque when load is applied</td>
<td>Insecure mounting or inadequate base structure</td>
</tr>
</tbody>
</table>

The above most dominating findings are explained a bit further below.

3.2.1 Only one Component vibrates

If one component on the aggregate in general (a pipe, a bracket, the generator regulator box, etc) is the only item with excessive motion, the component mounting (design, stiffness and damping) will have to be altered until the motion is reduced to an acceptable level at operating speed and loads. Refer also to resonance below.

3.2.2 1/2 Order Vibration

When the vibration motion measurements show that 1/2 order (the frequency is half of the main EDG rotation speed) is causing the problem, the engine fuel and governor system should be serviced to eliminate engine misfiring. No other work should be attempted until engine misfiring is eliminated as a reason.

3.2.3 Irregular, Unstable Vibration or Unstable Load

Irregular and unstable function of the injection pump due to internal wear is present.

3.2.4 1st Order Vibration

Refer to Balancing and Resonance further down in this guide.
3.2.5 2nd Order Vibration

When excessive second order vibration occurs on, also the timing of the second order force balancers should be checked.

3.2.6 Higher Order Vibration

Other high order vibration levels cannot be corrected with flywheel balance weights or balancer timing. Usually these orders involve the structural characteristics of the generator and base which will need to be altered. Local resonances of coolers or turbo charger brackets or exhaust piping and cooling and fuel and oil piping are also such examples often present.

3.2.7 Non-Diesel Engine Vibrations

If the vibration motion involves non-engine mounted structures and the engine vibration motion is acceptable, either the off engine components must have their mounting altered or proper vibration isolators must be installed between the engine or generator set and the structure or similar structure to isolate the offended component.

3.2.8 Excessive Engine Motion

If the diesel engine unit has excessive but stable motion, it will generally be due to misalignment/unbalance, resonance or torque reaction. This does not exclude the more obvious cases of just loosening or broken fixations of components. However, if the vibration motion for the first and second order still remains excessive after examination and correction attempts, a full aggregate modification should be investigated.

Another condition that causes excessive engine motion is the rigid body mode. Due to sizing and selection of engine or eventual resilient spring (rubber blocks or spiral steel springs) mounts, the engine and/or skid can move in a rigid or semi-rigid body mode. This can only be corrected by changing the dynamic properties of the EDG set.

If vibration is present, but the engine vibrations are within agreed limits, a vibration specialist should be consulted. If the uncoupled engine already in idle exceeds the limits, engine components rotating at engine speed (first order) or twice engine speed (second order) should be inspected.

The next section in 4 below takes a broader view and relates the actual behaviour to support differentiating between similar phenomena. Since the torque fluctuation is the dominating issue of diesel engine vibrations, the nature of the pulsation is explained in steps starting with an engine with only one single cylinder. This engine has very strong pulsations and as more cylinders are added, the actual torque pulsation is smaller and smaller.
Diagnostic and Mitigation based on History

4.1 MISALIGNMENT/UNBALANCE

Most linear vibrations of generator sets or other packaged units are caused by misalignment or unbalance of the rotating members. This typically results in first order vibration which can be corrected in the field. It is identified through its changes with speed as follows.

Misalignment/unbalanced vibration motion is relatively linearly increasing up to the service speed, as shown in the figure above, but exceeds accepted limits. For an EDG, this may be determined by manually operating up to the normal speed.

Misalignment/unbalanced vibration motion is not changed much by load changes. Although an increased torque may in combination with misalignment, force the engine or the generator to move sideways (transverse to the rotors centre line, normally horizontally, but also vertically) if the fixation below is not correct. Also refer below.

If misalignment or unbalance is identified it is good to first check the alignment of the unit. Check for soft foot, radial and angle (axial run out at coupling rim). If vibration is still excessive, consider trim balancing. If vibration is still present after the balance procedure for a rigid foundation, mount the unit on isolators and repeat the balance procedure until a satisfactory level of vibration is obtained. Consider also next section about resonance.

4.2 AGGREGATE RESONANCE IN A LINEAR FASHION OR FROM A ROTOR TORQUE

Resonance occurs when a large vibration motion is found within a narrow speed range, as shown below.

The vibration may occur on diesel engine itself, the generator and/or the attached equipment, such as piping, turbo chargers, exhaust piping and air coolers. When vibrations peak out in a narrow speed range, the vibrating component is in resonance.
There are three primary methods of reducing resonance vibration levels worth mentioning.

- Changing the natural frequency of the part that is resonant.
- Reduction of the exciting force.
- Adding damping to the resonant system which can mitigate well.

If the following checks at first focusing only linear vibrations show that the cause of the problem is the structure that the diesel engine itself, the generator, or other packaged unit such as turbo is mounted on, the system designer should be consulted. If the situation allows, these activities have been effective:

- Check the general alignment of the unit. Primarily the skids flatness to floor. Secondly each main components freedom from forces and skewed shape (often called soft foot) and finally the concentricity of the main driveline. This can be handled with the same care and accuracy as any other rotating machine in the same order of size. Resting play in sleeve bearings should be considered.
- If vibration is still excessive, consider if this is a new issue after revision work. Check if the radial and angle fits internally between rotor parts are restored within tolerance.
- Investigate if an onsite trim balancing can reduce the vibration. See below.
- If vibration is still present after above suggestions, consider replacing existing fixation to floor with soft elements and retest. If the EDG is clearly changed, the problem may be an improper or defective mounting system which requires changing.
- To trim the residual unbalance distribution along the driveline may, after proper resonance free mounting, prove to be possible to improve with an onsite balancing. The sensitivity to a test mass should be compared to the ISO 1940 residual eccentricity calculations to eventually identify a system with too high sensitivity caused by a rotor or rotor part resonance. See below.
4.3 VIBRATION RELATED TO TORSION

In a more general sense, torsional vibration primarily simply refers to irregularities in the speed of rotation in a shafting system. In the context of a diesel engine driven systems, the shafting system is referred to as the “driveline”. It formally includes all of the subsystems (e.g. rotor parts as well as other stationary external parts) directly connected to the crankshaft with bearings, both internal to the engine and external. Torsional vibrations appear in the driveline as a result of engine combustion impulses. The reciprocating motion of the pistons and connecting rods is the main reason. Also to a much smaller degree caused by the operating characteristics of the driven equipment. This may be the generator, propeller, pump, turbo and starting air compressor.

Also gear-driven equipment such as a fire fighting pump driven from an angle gear box or other auxiliary components do contribute to the overall dynamic behaviour of the driveline. The driveline has all the parts from the torsional damper all the way through the engine, coupling, the primary gear, the driven gear, a soft of rigid coupling, and often several shaft parts down to the tip of the impeller.

4.4 ROTOR PARTS

We start this part discussing only the rotating parts. For the EDG set it may normally appear as just two rotors with a soft rubber element coupling in between. In a few cases, especially older units, the generator has a system to magnetize the rotor with an added small exciter rotor added as a flying outboard rotor with a stiff flange to the generator rotor. There are in addition further exciter components which rest with bearings on the rotating generator shaft as well. In a modern generator this is fully electronic which of course has other installation components.

Torsional vibrations may be understood as the rotational equivalent of linear vibrations. In a similar manner many of the dynamic concerns that must be addressed for linear vibration also apply for torsional vibrations. A driveline is designed for transmitting the torque that the engine delivers to the driven equipment, but it must also be capable of withstanding the oscillating energies of torsional vibrations without damage or wear. The dimensional design of each of the components in the driveline contributes torsional characteristics to the driveline. It functions as a dynamic system with its own fundamental characteristics and natural frequencies. If the frequency of the excitations contributed by the torsional vibration sources match the natural frequency any of the components in the system, a resonance will be excited.

Typical range for the lowest of such an EDG global resonance is below 10 Hz.

Any shaft rotating with a mass attached at each end may experience torsional vibration. The simplified drive train in the figure below illustrates a torsional system formed by the piston, connecting rod, and crankshaft within a reciprocating engine. Even without combustion impulses, the forces generated by inertia as the piston and connecting rod change direction at each end of the stroke, are enough to cause a fluctuation of the torque that can be measured in the crankshaft. The bearings and engine block transmits the same force out to the machine with opposite direction.
4.5 TORQUE REACTION

When the vibration motion increases as load is increased, as shown below, torque reaction is often the problem. With a two bearing generator, it may be caused by insecure mounting of the engine or generator to the steel frame and/or by a frame not sufficiently rigid to withstand the associated forces.

Torque reaction problems are not found with close-coupled generators. The rigid joint between the engine body through the stationary flywheel housing and the generator body is generally well designed and adequate to withstand torque. This is how most car engines are joined to the gearbox.
4.6 WHERE THE TORQUE ACTUALLY COMES FROM

The power of a complete EDG is often noted as the electric power from the generator in kWe. If a load of 2500 kWe can be accepted, disregarding for a moment the losses, it is actually a mean value combined from all combustion torque impulses added from each of the cylinders. They all occur for each piston for every fixed sector of rotation and all cylinders in a train of torque pulses through the cylinders until all have added their share over a full complete combustion cycle and the cycle starts again.

4.7 A SINGLE CYLINDER DIESEL

In order to build an understanding for the specific properties of a diesel, a single cylinder engine is at first considered. It may be as lawn mower or chain saw has a very sharp and strong positive crank pulse as much as 15 times the average torque used. As the piston returns from its suction and starts compression before next ignition it is even breaking its motion with a negative torque in the order of 5 times the average torque. This can only be handled if the engine has been designed with a flywheel with enough polar inertia to limit the destructive forces. The gas pressure can pass 100 bars at the moment of firing.

The aspect of perfect balancing quality of a diesel engine is not so strong when the force from the torque is compared to unbalance mass centrifugal forces. We will see how increased number of cylinders will reduce the torque pulsation, but never eliminate it. Several active design features are installed on all IC engines to mitigate the pulsation. It is necessary to fully understand what they do.
A single cylinder engine such as a lawn mower running at 2000 RPM is therefore shaking strongly at 2000 / 60 / 2 Hz = 17 Hz. Even if the flywheel is heavy, since the whole machine structure (wheels, casing and handle) has a rather small inertia opposing an imposed horizontal twisting rotation.

This picture is a single cylinder diesel running in idle in a test run. Note the pair two large flywheels. The make is Mars 5 Hp. When loaded it keeps the set speed well but starts jumping away if not secured since the output load has the large torsional rocking force opposed to the rotating pulsation at the frequency of half the speed.
4.8 INCREASING TO FOUR CYLINDERS

To study the importance of the basic diesel engine property used in an EDG, we first add more cylinders to look at a normal car engine with four cylinders. The torque variation is now at a frequency equal to half the rotation speed.

Figure 7

To make the example general, we set the average output level in the vertical scale to 0 in the figure above. The torque pulsation is almost 5 times the average load. But we know how relatively smoothly a car engine is. We need to understand how the pulsation is handled.

4.9 INLINE AND V ENGINES VARY IN DESIGN

For instance a 16 cylinder engine, the angle between two consecutive firing moments over one full turn, is normally evenly spread over a full turn. The V angle of the cylinder rows and the crankshaft crank angles act to distribute the load well. Due to the mechanical design, (again crankshaft crank angles and cylinder row V angle) this can also be a series of 22° - 50° - 22° - 50° and so on angle distances. Please note that this is a first simplified explanation. We must also study what happens when different number of cylinders is added.

A diesel with many cylinders may also have a design in two banks with the cylinders arranged in a V-shape with 50°, 60° or even 90° of angular distance. All pistons are normally identical, but some designs have a pair of piston rods working on the same crank but with different rod lengths. There is also a design with a pair of rods where
one shorter in the pair actually has its own bearing situated on the other longer rod very near to the crank bearing. This is all the designers trying to make the best design.

In the simplest engine with only four cylinders they are all in line and work individually in their four individual quarters of the two turns per cylinder. The order of action depends on the crankshaft. The camshaft is designed to do the valve work in the right order and precise fashion. For this example a flat crank shaft is used, where all four cranks are in 0° or 180° directions. The firing order here is 1-3-2-4.

When No 1 fires and does the torque push, the No 3 is doing the compression to make its firing as the next and so on. The four firing cylinders will do the torque work at a frequency of half the rotation speed. Check the diagram above where two full turns are shown.

Figure 8

4.10 THE DMF, DOUBLE MASS FLYWHEEL

The goal of the firing sequence is to distribute the load as evenly as possible concerning primarily stress in the material and hence vibration, the thermal load and also handle pressure pulsation from the exhaust pipes as smoothly as possible.

A diesel car engine has some 4 to 5 times higher cylinder compression than a petrol engine as shown in 4.8 above. Most diesel driven cars are however very smooth with regard to vibration and noise. There is today a special design of the torque pulsation damping in a car diesel used. It contributes well to the comfort in the car and it has a similarity of vibration mitigation to a larger engine for an EDG. It is therefore a good reason to look at, and understand the design before we go to the bigger EDG engines.

Flywheel inertia has a natural relation to outer diameter and the mass at the outer edge plus of course the rotation speed. There is not much of a practical limit of the diameter for a high inertia in a stationary engine. But a car must carry the weight all the service life creating an accumulating transport cost. An improvement of the torque pulsation effect is to use double disk flywheel inertia. It is now common to design the single flywheel into two disks. One is stiff on the crankshaft as before, but the second half is centred, but fully loose to rotate back and forth, together with the main flywheel body. The two disks have an outer flange holding a spiral shaped tangential spring and often
a volume of silicon liquid as a grease around the edge. The spring and shear in the liquid is adding damping to any relative motion between the two flywheels.

This picture is a double mass flywheel in illustrates the function of angular motion damping. The disk has the two parts with a red arrow on the disk fixed to the crankshaft and the blue arrow on the disk which has both a silicon oil damping and a pair of springs working tangentially. As the loose half (blue arrow) is moving with the crank shaft, the reaction of the crankshaft (red arrow) is well reduced thanks to damping in the spring and shearing damping oil in between.

4.11 THE FREE ROTATING STEEL RING IN A SILICONE LIQUID DAMPER

The above arrangement in a car engine flywheel as DMF would be too complicated in most large diesel engines. They have traditionally instead the silicon oil damping in the shape of an extra cylindrically shaped container almost filled with a solid steel ring. The small radial gap is filled with silicone liquid. When the crankshaft and container is moving with angular torque fluctuations, the inertia of the steel ring will not react quickly enough but cause a tangential shear movement in the silicon. This adds a significant damping to the crankshaft.
The EDG larger engines basic working design is now a logic development of what has been shown above. The engines are developed along two main paths. One is adding a straight line of cylinders. When more and more load is needed, the number of cylinders reaches at first 8 and the bore and stroke is increased until a suitable load can be obtained. Hospitals and computer server halls EDG often start below 1000 kWe and common 8 large bore straight inline diesels are here common. We will start looking at the torque pulsation properties of 8 and 12 cylinder engines. (See Figure 15 below.)

So far a flat design of the crankshaft was a simple solution to connect pistons. With more cylinders, the torque pulsation to handle now starts to reduce to a reasonable level. But it is still a challenge for an eight in line cylinder torque mitigation design. The crankshaft design now often includes a crankshaft with for instance 120 degree angular distance. All these variations are made looking for the optimal smooth flow of torque into the coupling. The polar moment developed for a 16 cylinder 2500 kW EDG is still typically a peak torque twice as large as the mean value.

In principle, most designs are based on the principle that an optimally smooth torque will be created if the two four stroke turns of 720 degrees are split in 8 sectors with each pair of pistons separated only by 60 degrees. All pistons add up in a continuous train of pulsations for the full power. Some EDG engines have a slightly different design where the cylinder bank angle is reduced to 50°.

The rotating system, the driveline, has the same method to take care of the pulsation as the previous individual engines described above. But important to notice, thanks to the increased number of cylinders, the need for mitigation is less, since the pulsation peaks are lower. A flywheel is still used in the drive end and the internal torque damping rotor with silicon is added as a separate unit in the free end. The main reason for a silicone torsional damper is also to deaden all resonance ringing in the crankshaft. These will otherwise soon lead to starting cracks and resulting fatigue breakdowns.

The flywheel is the last important item in this survey which is special to diesel engines. It serves in fact three functions. The first is adding a polar inertia to the driveline. To create as much inertia from a given mass as possible, the outer edge of the wheel is a massive ring. Secondly it has a gear for the starter motor function.

The third function is to give an easy flange for the coupling to the load. The outer side of the flywheel often has a ring of rubber elements. It forms a soft, often called resilient, coupling to the generator rotor s coupling half. The polar moment of inertia of the diesel engine and the polar moment of inertia of the generator rotor are two masses in a
torsional resonance system. The rubber elements in the coupling are a soft spring and create a soft connection.

Flywheel

Figure 13

The lowest resonance between the two rotor bodies will create a good isolation effect for torque pulsation frequencies above it. For a practical example, consider a case with the torque excitation frequency near 65 Hz. Typical for a low speed EDG with an inline 8 cylinder engine, the first mode resonance in the driveline should be at best below 40 Hz from this point of view. It will protect the rather complex generator rotor from excitation at in the range of some 50 Hz and upwards. This range is important, since it can easily force rotor parts in the rotor cables and exciter rotor and electronics to reach vibration level causing fatigue cracks.

These two pictures show the two sides of the typical rubber element based coupling. One side is the black large flywheel behind and the blue generator coupling hub in front. The other picture is the generator coupling hub seen from the other side 14 pcs rubber elements.

Figure 14a Figure 14b

As described above, the 8 cylinder crankshaft can stay flat, but for the 12 cylinder engine the angle per cylinder cranks is normally 60 degrees. For practical reasons, there is another complication as well. The number of cranks should both be related to 60 or 120 degrees and also be much fewer than 12. Using a V configuration with pair wise pistons means that each pair is using just six cranks and each crank is at angle steps of 60 or 120 degrees. Small variations of this are also in the market.
The designer will choose the good distribution along the crankshaft for minimum forces during the work.

This figure above still shows an inline both 8 and 12 cylinder engine torque. The smallest V engine for an EDG often has as many as 16 or 20 cylinders. The torque pulsation is often a multiple of half the speed, for instance 10 times. This is due to the V arrangement of cylinders in two banks rows and cylinders in pair on each crank and various angles in the cranks along the crankshaft.

The very important issue is to keep in mind that the force which is trying to twist the driveline and the stationary engine parts and the steel frame below is for most EDG in the order of twice the nominal full power torque. This is always a much more powerful force than any unbalance. To take care proper of this and to mitigate its effects on all main and secondary components is the central issue for all EDG designers. It will of course also reflect directly to the demands for the procurement organization to investigate and secure guarantees for a satisfactory result broadly around this impulse.
5 Measurements and Frequency Analysis

This report is not intended to be another basic training in general proactive maintenance of plant rotating equipment. For all basic diagnostic work it is suitable to use the same vibration route collector systems which are already used in a power plant. A machine is described with its properties in a database and a route of systematic measurements is used during the test runs. Points are the four main end bearings of the diesel engine and the generator with additional structure points on important piping, coolers, exhaust pipes, supportive pumps, starting air compressors and so on. All machinery and all static equipment should be treated as any other important machine in the plant with regard to frequency range, time for a measurement, etc. The analyst who is well experienced and well equipped for all other machinery will also have the tools and the experience to handle the EDG sets properly.

In special investigation situations when the EDG is brand new or a diagnostic trending has indicated an upcoming suspected fault, it may be a good support to do vector measurements to find the mode shapes of parts and equipment. Also for this work, the standard analyst in a plant already has years of experience. The historical review has shown that the diagnostic work has normally not been a problem to reveal essential problems. Instead, the management attention to the findings and the implementation has not always been efficient and optimal.
6 Historical Mitigation Actions

6.1 WHEN THE AGGREGATE BASE (FRAME OR FLOOR BELOW) IS TOO WEAK

Assuming that a free standing two bearing generator is mounted on a weak base, the first order motion mode and the orders related to the number of cylinders firing with one crankshaft revolution are due to a torque reaction, which in turn will cause the softest part to deflect a bit more than the stiffer. This will cause a radial misalignment between the engine and the generator. In this case, the first order motion would be most prevalent. The vibration may sometimes have a component of the number of rubber elements in the coupling times the rotation speed frequency. As a result of this, the rubber elements are often working too much and increase in temperature locally. As a result the rubber will age faster, harden and crack prematurely.

This condition cannot be an issue with a close coupled aggregate where there is a direct stiff, often conically shaped structure fixing the generator to the engine as on many smaller EDG designs.

6.2 A STATIC STIFFNESS CHECK SIDEWAYS

A first check of the aggregate at rest should include the alignment. This is a conventional check if the engine crankshaft is well concentric with the generator. The value of parallel and angle tolerance in the vertical and horizontal direction is checked at the coupling.

A simple but valuable check is to check if the top of the engine and top of the generator are fully stationary to each other sidewise, when the load in operation is varied from idle to full.

A good simple method to find out the real static transverse stiffness is to attach a 2 ton chain puller on the engine top. Use the puller and pull sidewise relative to the generator top in series with a scale to indicate the applied force. The deflection is measured using a lateral dial gauge fixed to the base.

This aspect of two weak base frame applies also to a EDG with separately installed engine and generator. Sometimes the two-bearing generator has a structural steel base which is mounted on points to the floor. Soft pads which do not increase the stiffness of the base enough are used. The torque reaction for load can even deflect and twist a weak base visibly. This deflection can result in severe misalignment. The vibration is due to the couplings radial rubber element deflection over a complete turn.

A judgment of acceptable deflection can in the first estimation be commented by the EDG supplier. A comparison to the expected moment applied to the generator at full load is a good guide. The simple static stiffness of the parts must be well above the expected transfer of moment at load.

An infrared temperature trend measurement at the rubber elements during load over for instance half an hour will be a good support. The cheapest point meter is very useful.
If there is a vibration measurement system available, a simple lateral bump test will identify where the lowest frequency resonances are situated in relation to the strongest fundamental torque pulsation is expected.

The necessary stiffness can be expressed as a minimum resonance frequency. As an example for an inline engine, the torque pulsation is often a multiple of half the speed, for instance somewhere in the range 40 to 70 Hz. All facts should be documented with all other knowledge of the EDG set. The investigation methods are intended to be used as primary steps to diagnose excessive vibrations.

6.3 METHODS TO STIFFEN AN EDG BASE

If the EDG set continues to exhibit vibration after a good alignment, and the relative motion at top sidewise (engine/generator) is significant, the base is likely not strong enough to hold the torque power reaction and simply needs strengthening. If this is a new situation for the EDG in question, cracks or loosening fixation should be investigated. The cracks do not have to be severe; it may only have tuned a twisting resonance to reach an excitation frequency and actually related to the base frame twisting resonance.

6.3.1 Boxing up the Frame Coupling Part

Adding steel to form a twisting stiff section between the engine and the generator is a proven method. One method of such strengthening is to weld plates on top and bottom of the steel frame across the width of the base from at least 700 mm forward of the rear engine supports to at least the same distance behind the generator feet closest to the engine. If the flatness of the frame cannot be properly checked after welding, drilling holes for bolts and adding conical shear locking pins work just as well, especially in cramped on site conditions. The important goal is to form a completely full twisting stiff box sectional shape of the previously open frame often made from two parallel I-beams.

6.3.2 Increasing the Frame Sides

Another example is to add thick steel plates or parts of U- or I-beam parts along the outer sides of the I-beams. They should be long enough to overlap pass the coupling parts at least twice the engine/generator feet distance or other width increasing steel flanges on top and bottom. This can be needed if the flywheel and oil pan of the engine is located low down between the sides of the I-beams.

6.3.3 Adding an Internal X-shaped Cross

A bottom X-cross at the skids lower side has also been successful where also the cross point has been supported sideways up to the upper frame flange level (Figure 19). To reduce the thermal distortion to a minimum, hundreds of the smallest possible and well spread spot-welds and keeping the steel just hand warm is effective.

It is important to keep in mind that since the source of the torque or dynamic moment force related to the number of cylinders firing with one crankshaft revolution is a fully
natural part of the engines normal work. The aspect of the EDG design to avoid the relatively high force level is normally well taken care of by the EDG designer. If the EDG in question has a history of good low vibrations, the practical reason for loosening the manufactured stiffness should at first be found and corrected.

6.4 KEEPING AN OPEN MIND FOR A POSSIBLE INNER FAULT PRESENT

The possibility that the shown high vibration may be caused by not easily visible broken joint such as for instance screws cracking due to fatigue or simply loosening nuts or screws should be kept in mind. Inner malfunction of the engine due to for instance a broken camshaft sleeve bearing is possible. Scoring of cylinders due to chemical reaction between cooling water and oil additives can be initial reasons for abnormal irregular firing torque. The diagnostic work should not be limited to just total vibration broad band levels, but also many extra systematically distributed points on the engine using spectrum analysis. The ISO Standard acceptance limits can only be a general support. Detailed knowledge of all of the aggregate systems and broad general diesel engine experience is the important base for all diagnostic work.

6.5 MITIGATION BALANCING ACTIONS, BASIC BALANCING PROCEDURE

6.5.1 Identify a real Need for trim Balancing on Site

The balancing of a conventional single rotor has logical steps to comply with the ISO 1940/1 Standard to a precision which is related to a maximum residual eccentricity.

A crankshaft has for each crank an added large mass with the purpose to counteract the complete movement and torque from the piston or piston attached at that crank. The external balancing goal is to make sure that the total sum of these added parts and forces in total meet the ISO 1940/1 requirements for the complete engine.

A rotor built from parts (added flywheel, added exciter, etc) has also the assembly error to take into account. If the rotor has any axial length, it will require a dynamic balancing procedure in two planes. Even when parts are made almost equal in mass, the final fixation to the crank shaft has tolerances which can add up to important size.

A valid reason to trim the residual unbalance in an engine can be present when the actual service stand has a local structure or global resonance which is considered too expensive or toocumbersome to correct.

When revision work includes a workshop balancing of the crank shaft or the complete engine rotor assembly with torsion damper and flywheel, it is usually balanced in a workshop balancing machine. The used drive in the balancing machine is often a cardan shaft with various hubs and disks to fit the driving system. It is essential that the balancing quality of the drive is proven to have a residual unbalance well below the limits for the rotor. The radial tolerance of the assembly fit is equally important. The traditional index test is here important to be performed and documented.

A similar aspect of final balancing quality is the final revision test driving a water brake system in the workshop. The cardan shaft is now rather rugged since it must be able to handle the full load with good margin. Its balancing quality and its tolerance in attachment to the diesel engine flywheel must be measured and the index test is a quick and good method.
Any index test means that the assembly is operated two times. The second run with the drive shafts turned half a turn. The vectorial difference will directly measure the drive shafts unbalance in two opposed directions. So the total change is twice the error in a simple way to express it.

6.5.2 Making Pistons and Rods weigh the Same, Trimming Crankshaft

A diesel engine rotor has special issues related to the inertial forces from the rods and pistons as well as the rather large mass of the torsional damper rotor which normally only rests in the silicon fluid damping with a shear motion on the container circumference filled with oil.

Each crank has counter weight segments applied to balance the local piston following well established calculations in the engine development phase. If there are signs of unbalance in a diesel on site there is most probably an error in assembly of the rotor parts or unwanted resonance amplification in a given direction. A large portion of the NPP engines has such a situation. For the crankshaft alone a precision of G6.3 is acceptable.

To minimize the added unbalances, pistons and rods should be trimmed to the equal masses. Rods are weighed on two scales so that their ends are the same in the group. This is the same procedure as steam turbine blades are made equal in moment weight.

6.5.3 Field Balancing of the Engine or the Generator, Verify the Need

Vector Method

Trim balancing of the diesel or the generator is using traditional methods for balancing on site. At best, a vector meter (an instrument measuring the 1x speed level and angle) is applied with the help of a reference sensor to get a pulse when chosen reference is passing. The vibration is measured at both ends of the machine to balance in two radial directions, normally vertical and horizontal. A test weight can be added in one plane in the flywheel and one plane in the torsional damper at the free end if the balancing was not made before. The sensitivity is measured or an old figure is used and a balancing weight is calculated and installed.

Experimenting by moving the Added Rotor Part in Steps, also Look for the Need

Balancing is not a very high tech procedure and can also be made successfully when no vector instrument is available. History has shown that smart mechanics have found ways to balance. If the unbalance is mainly a result of bad radial fit in the coupling to the generator, this rotor can be moved stepwise in 90 degree angles to find a minimum of 1xRPM vibration and a final trim can be made adding washers to the coupling. A technical judgment is needed to make sure that the balancing is the correct move. There is maybe only a steel fragment left in a facing surface to correct at first, which has caused an improper run out.

The Two or Three Point Method

Adding a test weight at one position in a first test run and then the same in the opposite direction can be used for a simple graphic sketch to find the needed trim weight. This is often called the two-point method and it has a more conservative approach in the three point version with 120 degrees between three test run positions.
Shop Balancing, stiff or soft Machine

The balancing quality from the production depends on the actual quality used in the factory. Modern balancing machines are of two principal types. One has in the two main bearings a radially very stiff design where the 1x speed force is measured inside the bearing. It is electronically converted to a balancing mass to add or to remove in the opposite direction. The other type of balancing machine has essentially very soft bearings and the actual vibration is measured. The instrument is a similar method to the on-site vector filter method for balancing. The extra support of a built-in calculation of the balancing weight is added.

Cardan Shaft Error in the Balancing Machine, Index Test

Driving heavy rotors such as the crank shaft, the torsion damper and the generator rotor flywheel and coupling hubs and exciter rotors is often made with a cardan shaft with its couplings. Even if the cardan is well balanced, it may have a fit tolerance adding an unwanted unbalance. The added errors from this drive help can and must be measured in the balancing procedure with a so called index test. The rotor is turned half a turn in the cardan shaft coupling and the two vectorial readings are compared. This will show the cardan error pointing outwards in two 180 degree fashions. The balancing can be aimed to the zero point between them or the cardan can at first be improved. Since the accuracy of balancing is most often measured in microns, already very small errors in cardans will add a large error to the balancing result of the main rotor.

An Important Check of the Balancing Machine as a Shop Witness

All balancing machines have a calibration routine and a regularly updated protocol is kept for each machine. The second big source of balancing errors is a personal mistake by the operator. With large rotor costs are at stake, it is always a very good check up by adding a simple know test weight like for instance 10 grams at known radius during four test runs in both planes if used. The vectorial readings are noted and the accuracy of the settings is directly verified.

Mandrel Tolerance Check and Documentation

Rotor parts are not only driven by a cardan, but also often need a help shaft (often called mandrel) to rotate. The mandrel itself and the fit to the object to balance must be perfect and known to its error. A good way to document and correct for the mandrel error as well as the cardan error is to always make an index test. Two test runs with the balancing machine external rotating parts turned half a turn between two test runs.

Turbo Balancing

The turbo will never be a balancing issue in the workshop due to the very high demands of the balancing precision. But the basic method is identical to any balancing procedure.

6.5.4 Secondary Help Systems Performance

Engine and generators stator parts with many types of attached piping for fluids and gases, wiring, turbo, electrical cabinets, heat exchangers and coolers are examples. Many practical cases during the large number of accumulated service years have proven that designs which only focus a strong and sturdy looking fixation often crack.
Items can always be fixed well and secure against using brute force. The necessary designer’s attention to apply to create good systematic mitigation of vibration is often too small. Soft and carefully damped but rugged installations is a well recognized method and the result is a long term safe well functional component.

Examples:

Figure 18
Pump with still low levels after years with a seismically soft foundation on four metal wire cushions and piping through rubber compensators. Compensator flanges firmly secured on both sides. Levels are lower than ISO 10816-5

Figure 19
Cracked pump foot after only days of service with high vibration levels but fully acceptable according to ISO 8528-9 and ISO 3838.
7 Applicable ISO Standards

For all general rotating machinery, a rough first judgment from measurements of total vibration levels is backed by ISO Standards such as ISO 10816. Most common types of machinery have separate parts in this ISO Standard.

The further diagnostic work to find and mitigate the reasons for vibration causes is supported by ISO Standards of applicable methods as well as accreditation of the technician’s competence levels. This is found in ISO 17359, 13373, 13379, 13381 (Methods, Competence, etc.), ISO 18436 Part II (Accreditation). Many active industrial plant operators have found this usage of proactive maintenance to save large unwanted costs as well as keeping the availability of the production very high. For the Scandinavian countries and most EU countries, the competence of the staff working with diagnosis of machines is normally based on a long practical experience in a maintenance workshop. Most activities are however based on the ISO Standards for vibration levels on non diesel machinery with the exception for piston compressors.

7.1 ISO STANDARDS FOR DIESEL ENGINE VIBRATIONS

All diesel engine driven generator sets are technically very complex. A simple broadband reading at the main bearings is next to impossible to use as a reliable diagnostic support for safe long term EDG operation. Evaluating the results from matching the large number of faults documented in this investigation, compared to advice given by existing diesel engine standards, clearly shows that very small number of the faults would have been detected in development and thus avoided.

Any Standard can indeed be used and from the advice given, the limits stated may allow trouble free operation with certainty but never with any responsibility. The sources of the brave statements of high allowed vibration levels tend to be suppliers, sales persons and unfortunately various management oriented persons. Workshop and operation oriented staff may at times have a say for the meeting minutes, but far too rarely a real part in important mitigation decisions.

A more clear result from the history collection is that machines made from common steel, designed by common development engineers have essentially the same or just a bit better reliability as the large number of more simple machine which just rotates without internal combustion. What is normally sensitive to vibrations is not less sensitive just because it is installed in an EDG environment. The well established levels for judgement for traditional machines will fill a good purpose in both.

From the rather impressive accumulated number of operating years from the current nuclear power stations, although a very limited number of different makes of EDGs, it seems that a rather conservative increase of the normal rotating machine limits is that best way to go. What should be a limiting factor to accept high levels is also the extreme trust which is placed on the emergency nature of reliability of start and accept full load for an unknown length of necessary service.

When operators try to endure higher vibration levels, there is a clear risk, that the important and even necessary functional reliability will not be reached. Instead of spending high repair costs and loosing EDG goodwill and allow risks for secondary plant damage, it seems to be a much more efficient path to mitigate the vibrations in a systematic manner.
Examples of good procedures are to only acquire already proven designs and immediately perform necessary mitigation modifications when errors are found. This may include tuning of resonances and adding effective, well known damping modifications.
8 Recommended Requirements

8.1 WHY EDG RELIABILITY IS RELATED TO ASPECTS OF VIBRATIONS

The EDG set is today used in a wider area than in just a nuclear power plant. It has become a vital part of important parts of society such as hospitals, power stations and IT centres. We hear too often that a hospital EDG did not start properly at the onset of a blackout. The traditional installations with “reliability as good as seems needed” are no longer acceptable. The EDG set reliability in a broad sense must have a good margin above just enough, so that a fault which prevents the proper function has the smallest possible risk to happen. Experiences related to vibration related issues from more than 100 years of EDG accumulated operation, has been investigated for this report. It can be summarized to state that requirements concerning vibration aspects for the procurement of new EDGs will with high certainty positively increase the EDG actual long term reliability. All these requirements can also be applied to further maintenance of existing EDGs.

8.1.1 Documentation

Fundamental principle of a project handling is that all the related aspects of vibration which is calculated, measured, verified and accepted by the parties shall have a document following ISO 9000 and applicable standards. All results, such as calculations, balancing work, alignment work and resonance investigations shall be documented completely and handed over to the end user when it has been made without delay. The documentation of each item under this heading is a part of the delivery of the products. A simple motivation of this is that the status of the machine as it develops over time must at any moment be possible to follow.

All documents and protocols shall be signed with full readable responsible person name and date of work shall be noted on the documents.

8.1.2 Qualifying Standard Equipments

When a product, either small or comprising many parts, in the project is handled by many different persons, make sure it is specified so that there is no chance that any person in the chain can misunderstand anything. Use for instance metric units and ISO Standards when applicable and verify that the supplier is doing that as well. If the equipment to be eventually purchased has been sold to anyone before, make sure that at least some of the present end users are visited by technically competent persons who would use and maintain the equipment and request a feasibility study.

8.1.3 Qualifying Prototype Equipments

Do not even consider to include a prototype in the EDG project. If it turns out to become necessary, it must be motivated thoroughly. Show and document every uncertainty in all unproven aspects. Invite the supplier to convince internal end users and maintenance staff to scrutinize the details. If there are any uncertainties, just leave this supplier or rephrase requirements thoroughly.
8.1.4 Units
Again, cannot be repeated enough, all units are to be Metric according to the SI-System. Angle is to be measured as degrees with 360° for a complete turn. Where applicable, concerning aspects of vibrations, angle is measured against the direction of normal rotation. This means that when you observe a rotor marked with angles and it slowly turns in the main direction of operation, the numbers on the rotor are increasing in order. Starting point of angles and direction to be clearly indicated on parts and documented. No Standard from outside EU / Sweden should be allowed.

8.2 RECOMMENDED LEVEL REQUIREMENTS AND BALANCING QUALITY

8.2.1 General
Rotor parts such as a single crankshaft or if applicable two parts of a crankshaft shall each be balanced both statically and dynamically one class better than the complete rotor. For combined crankshafts, the meeting surfaces shall be documented before and straightness after assembly shall be documented. The complete crankshaft shall then be trim balanced again dynamically.

The generator rotor shall be balanced alone without eventual exciter part and finally with the exciter installed. The run out of the meeting surfaces shall be documented.

Balancing shall be performed both for individual rotor parts (such as, but not limited to, shafts, impellers, coupling and coupling half, motor rotor, motor fans, axial thrust discs etc) and for the complete assembled delivered rotor. When rotor parts (such as impellers, couplings and axial thrust discs) are balanced, the final assembly trim balancing should be performed for the rotor parts step by step when assembling the complete rotor. Balancing with half key convention according to ISO 8821 shall be performed.

Any excessive unbalance found when a part is added to the rotor, shall be trimmed at its respective two planes.

8.2.2 Balancing Quality Grade for all Rotor and Rotor Parts
- Rotor complete according to ISO 1940/1 G2.5 (for normal service speed)
- Rotor parts one grade better, which is to ISO 1940/1 G1
- For the diesel engine crankshaft alone G6.3 is acceptable.

8.2.3 Secondary Parts such as Pumps and Fans near or on the EDG Unit
For pumps and other aggregate components (e.g. hydraulic coupling and motors) with higher speeds where the shaft is not any longer to be considered to be stiff, also balancing according to ISO 11342, low speed as well as high speed balancing, shall be performed. The balancing condition shall be fulfilled during the whole operating range in the balancing machine including full speed. The residual unbalance criteria shall be applied.
8.3 PERFORMANCE REPEATABILITY

When applicable for the design, and when an over speed test is applied, the repeatability (stability at repeated reaching of full speed) shall be verified.

8.3.1 Verification of Balancing Procedures, Mandrels, Etc

An index test shall be included in the balancing procedure for all rotors showing the change of unbalance from turning the cardan/help shaft 180 degrees (one full half turn). This will ensure there are no built-in faults in the individual rotor part balancing status due to eccentricity and/or unbalance in the cardan shaft and/or other help shafts (mandrels). The result of this test shall be recorded together with the normal balancing, reporting the balancing and checking procedure step by step. Reading in grams, balancing radii and degrees for each plane before and after the turning of the cardan shall be done. Resulting error in uncertainty of final balancing result shall be calculated and reported. Angle convention used shall be clearly documented.

8.3.2 Witness Points, Documentation, Handing over of Protocols

Balancing of the rotors is a witness point where the end user may attend for the first rotor and can choose to attend any following rotors. Balancing records from balancing in the balancing machine shall be given to the end user directly after balancing step by step without delay.

The final check and documentation of the complete rotor residual unbalance when coupling half is installed shall use planes which are used for balancing, such as motor core ends. This applies also for individual rotors inside hydraulic couplings.

8.3.3 Balancing Machine Calibration and User Settings Verification

Calibration of the Balancing Machine Instrument

The balancing machine is an instrument with a need for calibration which is normally performed once a year. Using the machine includes a range of settings by the operator which directly influences the measurements. The annual calibration certificate is to be included in the protocol.

User Settings

The current user settings can be adjusted wrong to cause incorrect readings. This is more common than an aged or wrong calibration of the instrument electronics. A complete balancing protocol will, in addition to the instrument calibration, verify that the operator settings are made without a mistake. Upon request from the end user, a known test weight should be inserted and the change in the vector readings recorded (grams/degrees) and compared to actual unbalance introduced. If the instrument can read for instance a 25 g change and the angle, this should correspond very well to the inserted test weight. Accuracy should be judged towards required residual unbalance requirements for the rotor in question.

8.3.4 Recording of Balancing Weights, Screws or Grinding

When correction of unbalance is made by inserting removable plugs or screws, the plan of the inserted weights shall be a part of the protocol. All facts on this plan shall be able
to reinstall weight if removed by mistake. Size, location, date and person I clear text who

8.3.5 Key Convention ISO 8821

When keys are used for torque transmission in a hub, the balancing shall be done with half keys, ISO 8821. When key is installed, no part of the key is allowed to protrude.

8.4 PUMPS AND FANS AND SIMILAR ROTORS

For rotor shafts which are impractical to rotate as a complete assembly in the balancing machine, dial gauge runout measurements can replace balancing of the shaft. Shafts shall normally be straight within 0.03 mm TIR (Total Indicator Reading (diameter)) as checked mid span with shaft resting at intended bearing positions. Runout checks shall also include fits to parts such as the impeller and couplings.

8.5 ELECTRICAL AND AIR MOTORS

Balancing of motors is normally not a witness point. But all other requirements described do apply. The final check and documentation of the complete motor rotor residual unbalance, when coupling half is installed, shall use planes which are used for balancing, such as motor core ends.

8.6 GEARBOX AND HYDRAULIC COUPLING

Rotor parts to be balanced to G1 for the respective working speed/speed range.

The rotor to be balanced step by step during assembly until the complete rotor is balanced as a complete rotor. Although not used in EDG sets, this may illustrate how to handle complex rotors which maybe will be included in the future EDG development.

For the normal variable speed hydraulic coupling, the balancing shall be performed first on the secondary shaft, then the primary rotor is assembled part by part (gear, axial thrust disk, hydraulic coupling rotor etc.) to a complete stage by balancing step by step with secondary rotor stationary inside. An alternate method can be applied after acceptance from end user if the supplier can show that the change in balance quality during disassembly and assembly can be maintained. Balancing quality for the complete hydraulic coupling rotor with secondary and primary shaft together ISO 1940/1 G2.5 with respect to highest service speed for each rotor.

The final check and documentation of the complete variable speed coupling rotors residual unbalances, when also coupling halves are installed, shall use planes which are used for balancing, such as gearwheel end planes, hydraulic coupling rotor planes.

8.7 STRAIGHTNESS, ECCENTRICITY/OUT OF TRUE, ALL ROTOR PARTS

It is recognized that the good balancing results achieved in the balancing machine could be easily destroyed during dismantling and reassembly of the rotor. The out of true and run out from mounting shall be verified with run out indication, be minimized where applicable and reported. The impact on the final residual unbalance on the final balancing state of the complete rotor shall be analyzed and documented.
8.8 RUNOUT
Maximum runout value for shafts is 0.015 mm TIR (Total Indicator Reading (diam.)).

Maximum runout for a complete rotor is 0.030 mm TIR.

8.9 BALANCING POSSIBILITIES ON SITE
The machines shall be fitted with balancing planes which are effective for balancing and easily accessible on site on a fully installed machine, unless technically impossible. It should be possible to easily insert and secure weights without need of machining on site (preferably by access to threaded holes, slots or similar). Hatches with simple standard tools access shall be included whenever possible. This is important for both ends of the diesel engine of the EDG and the generator.

8.10 ALIGNMENT LIMITS (MOTOR, GEARBOX, DRIVEN MACHINE)
Limits apply for cold and warm machine. A normal requirement for alignment quality of standard machinery is max 0.05 mm parallel error and max 0.05 mm per 100 mm angular error. This is with compensation for warm machine in operation when applicable. Such warm machine correction shall be either calculated or measured and documented on site.

8.11 FLATNESS OF MEETING SURFACES
Before placing the machine on the intended space, the flatness of the two meeting surfaces shall be checked and if not flat with accuracy that will not disturb the machine part internal alignment, reported and corrected.

8.12 SHIM CONSIDERATIONS
Alignment report shall include shims thickness and positions and a soft foot check.
Number of shims shall be limited to a minimum. Shims shall cover the joining surfaces. Shim material shall be either brass or stainless steel.

8.13 RESONANCES
8.13.1 General
The complete machine, i.e. diesel engine and generator as one aggregate, comprising the shafts and the machine structure, all attached equipment, all attached piping, fully installed on site, shall be free from resonances that can amplify the natural excitation forces typical for the machine in question to reach destructive vibration levels.

Parts subject to excitation difficult to tune away, must have the vibration mitigated to an acceptable level. Normally be equipped with damping elements and fulfil conservative demands of maximum vibrations levels.

Separate machines in the EDG system such as starting air compressors and cooling fans have the same requirements.

All piping from and to the aggregate with soft compensators shall be well secured on both sides to eliminate risk for looseness.
For applicable designs, a stiff rotor shall have all lateral critical speeds above the operating speed. A weak rotor has one or more critical speeds below the operating speed. Documentation such as calculations and measurements shall be handed over without conditions.

8.13.2 Resonance requirements in Detail

Resonance freedom in general (lateral, torsional and foundation with the applicable degrees of freedom) shall apply in the range +/- 20% from 1xRPM, 2xRPM and 1xBP, firing torque pulsation frequency as well as gear mesh and 100Hz. (BP = Blade/Vane Passage frequencies).

For machines with variable speed, this shall apply for the whole allowed operating range.

The first lateral critical speed for stiff rotors (for applicable designs) shall also be higher than 125% of each rotor's normal speed at full load. Example: Generator rotor.

The lateral critical speed for weak rotors shall have a separation margin of plus 25% and minus 15% from the operating speed.

8.13.3 Calculations

All calculation results shall be reported with realistic accuracy and uncertainties in input data and the result and estimated tolerance shall be clearly shown. The complete aggregate including driver, generator, common frame, exhaust system, room floor, etc. are to be included in the calculations.

8.13.4 Verifications

The important resonances shall be reported and verified with calculations as well as with vibration measurements as a function of speed (if applicable) for the whole operating power range on the fully installed machine on site.

8.13.5 Specific EDG resonance Considerations

Seismic and Short Circuit Demands

The main bodies in the EDG are the diesel engine, the generator and the common steel frame. Due to demands to sustain seismic and generator short circuit events, the aggregate might have special arrangements in the building. All parts related to the vibration aspects of the EDG shall be included in the requirements.

Verification of all important Resonances

The six first degrees of freedom resonances if an engine is installed on resilient spring elements on the frame shall be calculated and tested on final site.

None of the above resonances is allowed to be excited by 1xRPM, 2xRPM, firing order torque pulsation frequency as well as 100 Hz with good margins with age considering the gradually stiffer coupling rubber elements.
8.14 VIBRATION LEVELS ON INSTALLED MACHINE DURING OPERATION

8.14.1 General
Final evaluation of vibration levels shall be done on the fully installed machine and supporting machines (starting air compressor and ventilation fans as examples) on site, where all allowed operating conditions shall be tested. Maximum permissible vibration levels shall apply for all allowed continuous operation load condition/ranges.

Bearing vibration in mm/s rms (vibration vector 1xRPM, 2xRPM as well as overall level up to at least 2000 Hz) should be measured at full speed on site at all main bearings in three directions, horizontal, vertical and axial from idle to full load and during a load which has stabilized thermally. A physically long diesel engine may in addition have a measurement mid span. All points shall be chosen to have a direct path to the bearing. If a bearing is not visible, the drawings should be checked to find a path as direct as possible.

Each turbo and connected coolers shall be measured. Representative points should be marked to allow a repeated measurement. Simple stairs to allow easy measurements is recommended.

8.14.2 Vibration Measurements on Site are performed by the End User
If the diesel engine is the prime mover for a fire fighting pump unit with an angle gearbox and vertical shaft and impeller in the water channel out of easy reach, also the lowest pump bearing in the water is included in the measurement.

Evaluation of the results also by the manufacturer and compared to contract agreements shall be documented. Long term monitoring shall at best be made each test run but minimum 3 to 4 times a year.

8.14.3 Bearing Vibration Limits
Measurements and requirements regarding bearing housing vibrations shall normally be according to ISO 10816 part 3, Zone A for newly installed machine. Measurements shall be taken in 3 directions on each bearing housing (2 radial and one axial direction). The same criteria shall apply for all 3 measurement directions.

The frequency range should cover from 2 Hz up to at least 2x vane passing frequency, however minimum cover up to 2000 Hz. Both measurements of vibration velocity in mm/s rms and vibration displacement in microns rms shall be performed.

For machines with gears, the measurements shall cover minimum 1x gear mesh frequency. The attachment method of sensors shall reflect the necessary frequency range for each point.

8.14.4 Vibration Limits For all Normal Rotating Machines
Maximum bearing vibrations normally according to ISO 10816-3, Zone A; – however maximum 2.8 mm/s rms.

El. motors: above 300kW and rigid support Max: 2.3 mm/s rms
El. motors: above 300kW and flexible support Max: 2.8 mm/s rms
Pumps/fans: with rated power below 15 kW

- and rigid support Max: 2.3 mm/s rms
- and flexible support Max: 2.8 mm/s rms

Hydraulic couplings: Radially Max 2.8 mm/s rms, Axially Max 4.5 mm/s rms

Diesel Engine and Air Compressor to Maximum 6 mm/s rms

The criterion for displacement microns rms is according to ISO 10816-3, Zone A.

The same requirements apply for all 3 measurement directions.

Modern main components of the EDG, engine and generator have additional parts installed on the sides and the top. A supporting measurement point at both engine ends on top may help verifying the good condition of the engine.

8.14.5 Torque Vibration

The EDG set torsional resonance properties are calculated earlier. Earliest during the FAT (Factory Acceptance Test) test and latest a repeat during the SAT (Site Acceptance test), the torsional resonances shall be checked with a suitable method and torsional vibrations shall also be measured. This can be made during rotation with a tachometer pulse sensor with many pulses from the shaft per turn (such as a pulse striped tape) and the actual torque can be measured using strain gauges. The necessary component properties (inertia and stiffness) shall be given to the enduser in the written documentation.

8.14.6 Long term Monitoring

Moving in torsion by the torsional damper and the flywheel should be as a minimum monitoring watched by the operator during test runs with a strobe flashlight. If the rotating surface has been given a pattern, it will be easy to notice if the movement is acceptable. This is not used as an absolute measurement as such, but has proven to be a good warning for detecting a developing fault in its early stages.

8.15 PIPE AND COMPONENT VIBRATIONS

8.15.1 General

Secondary vibrations in pipes (fuel, lubrication, water exhaust), pumps (all in the EDG), coolers (air, ventilation, water) with media under pressure), measuring units (such as Pt100 and pressure and flow gauges, cable channels and bundles), which can cause fire or other damage, should be judged according to the specification below. This applies for connecting pipe systems in nearby stationary plants containing steam, water, gas or oil which due to a cracking pipe or leaking coupling can cause operation disturbances, fire or other damage. Components joining the pipes are normally subject to harder judgment. Allowed limits may only be exceeded for short time periods – a few minutes in connection with changes of operational conditions.
8.15.2 Measuring
The overall vibration level in the frequency range 1 – 2000 Hz should be measured as vibration velocity in mm/s rms, and vibration displacement in microns peak.

8.15.3 Limits
If the vibration level is above 7 mm/s rms or above 150 microns peak, at any point and in any direction, step should be taken by the supplier to mitigate the vibrations.

8.15.4 Exceeded Limits
If, for some point however, it is not practically possible to reduce the vibration level, the risk for fatigue damages should be evaluated. It should then be verified that the risk for fatigue is very low and the procedure, calculations and verifying stress measurements, shall be documented and handed over without delay.

8.15.5 Excessive Levels
Vibration at any point of above which exceeds 20 mm/s rms shall not be left in operation more than minutes before shutting down. If the supplier has many identical EDG sets in operation with higher vibrations without any problems or wear, this can be brought to prove and be a base for a technical negotiation. Actual supporting measurements with strain gages showing a good margin to fatigue cracks or risk of loosening is also a possible path. The primary path is to evaluate a mitigation activity.

8.15.6 Spectrum
For all levels where the measured levels are near or above the limits, an additional spectrum shall be measured with good resolution and with least 20 seconds average time to 2000 Hz.

A good procedure is to take spectra from all points measured. The diagnosis of the spectra should be carried out in cooperation with the parties and the result of this be documented.

Presenting the spectra with common scales and a listing of important peak frequencies is recommended.

8.16 VIBRATIONS OF NON-ROTATING MACHINES (STANDING STILL VIBRATIONS)
For parallel mounting of two or more machines with antifriction bearings, on a generally common foundation of with common pipe arrangement, where one machine is standing still while another is in operation, the following should apply:

The bearing vibrations of the standing (not running) machine shall be below 0.5 mm/s rms

It is important that the vibrations, transmitted from the machine in operation to the machine standing still, is minimized to prevent bearing damages.

When units are stationary at the EDG set, it is important to verify that the vibrations do not represent an ongoing wear of stationary rotors such as false brinelling effects. Such
rotors should be installed with proper damping or locking arrangements to prevent failure when not in operation in its intended function.

8.16.1 Remaining and General

The measurement results from a FAT; Factory Acceptance Test, will be used as a supporting assessment of delivery preparedness and release from factory only. Supplier has the full responsibility to decide if the equipment is ready. Final evaluation and judgment of the machine will be performed with measurements and tests on installed machine on site, SAT; Site Acceptance Test, when all allowed operation conditions can be tested in a satisfactory continuity.

The contractor is responsible for all measurements, that they are performed and reported to the enduser. The enduser has full right to participate and also to make own measurements for verifying the result. The measurement result from balancing, any resonance tests and vibration measurements from shop test shall be handed over to the customer without delay.
9 Conclusions

Listings of thousands of fault reports in the four NPPs show that a solid system is used to detect and report all serious deviations from normal. The safety of the NPPs is not reduced when these systems are in operation. The real problems are found on the real world handling of the primary design faults which cause expenses and the lacking proactive maintenance approach on operating the EDGs over the observed years.

The care of the EDG is almost impossible to perform when the set is standing still. The person doing the test runs can normally do just monitoring the EDG performance from that point of view. From the limited capacity in the short testing time and normally also limited knowledge in EDG maintenance aspects there are good ways available to improve the EDG reliability using persons from the workshop staff. However, the expenses in manpower doing it in a real low cost proactive maintenance approach is extremely low when compared to what is spent on late detected problems and what these have caused in indirect costs like NPP load reductions.

Due to the very high demand for safe reliable operation, the whole EDG set should be given an extra ordinary care to reach low vibrations with today’s well known methods and staff time should be spent on all test runs with routine vibration measurements if it is not already established.
10 Suggestions for further Work

The main issues found adequate in this investigation with improvement potential are

1. Initiation and implementation of a user community among all EDG set users regardless of industry, country, installation age and make. There are such groups for specific makes, but not for sharing vibration related issues in a broad active experience sharing mode. Many NPPs have it in a sharing of general process experiences. WANO peer reviews are the main proactive way of sharing experience.

2. Improving and using already used damping methods in for instance cars and airplanes are well suitable to mitigate EDG vibrations. Examples are wire mesh cushions used to mitigate piping and added component vibrations. There is a need for dimensioning advice, rules and recommendations. Refer to figure 18 for a good example.

3. The many applications of pipes for various fluids and gases are at suitable points broken and joined with compensators of various types. The reasons are for example to allow free movement from thermal expansion reasons, tuning of resonances and allowing safe movements in the event of an earthquake. There are many different types of such soft pipe joints. There is a need for dimensioning advice, rules and recommendations.
DIESEL GENERATOR SET VIBRATIONS

Emergency diesel generator, EDG, sets are a vital part of a nuclear power plant, to safeguard emergency electricity supply to the plant in case of loss of the external grid.

This report summarizes operational experience in the area of vibrations from EDG sets in Swedish and Finnish nuclear power plants. Procurement, installation, commissioning and maintenance of EDGs is included. Guidelines for requirements for procurement are also suggested.