USAAVSCOM-TR-87-C-2

AD-A219 535

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Preprint for the
NASA/Army Rotorcraft Technology Conference
held at NASA Ames Research Center
Moffett Field, California, March 17-19, 1987

NASA

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SUMMARY

Helicopter cabin interiors require noise treatment which is expensive and adds weight. The gears inside the main power transmission are major sources of cabin noise. This paper describes work conducted by the NASA Lewis Research Center in measuring cabin interior noise and in relating the noise spectrum to the gear vibration of the Army's OH-58 helicopter. Flight test data indicate that the planetary gear train is a major source of cabin noise and that other low frequency sources are present that could dominate the cabin noise. Companion vibration measurements were made in a transmission test stand, revealing that the single largest contributor to the transmission vibration was the spiral bevel gear mesh. Our current understanding of the nature and causes of gear and transmission noise is discussed. The authors believe that the kinematical errors of the gear mesh have a strong influence on the noise. This paper summarizes completed NASA/Army sponsored research that applies to transmission noise reduction. The continuing research program is also reviewed.

INTRODUCTION

Helicopter interior noise and vibration are of concern because of passenger comfort and the effect on pilot and crew efficiency. The military is most concerned with pilot workload and efficiency, while the commercial arena is interested to attract passengers who are expecting jet smooth and quiet transportation with the convenience of a vertical takeoff from congested areas. In current helicopters the excessive interior noise causes annoyance, disrupts crew performance and requires ear protection equipment to be worn (fig. 1). Most experts agree that the major source of the annoying noise in the cabin originates from the gearing in the main transmission which is commonly mounted to the cabin ceiling. The sound and vibration energy is propagated through the structure or through the air directly to excite the cabin walls.

In the past, a major goal of transmission design was to reduce the weight, and as weight decreased, the noise has increased (ref. 1). This may be due to the increased flexibility of the transmission housing that accompanies a weight reduction. Also, the noise increases with the power and size of the helicopter.

The objective of this paper is to identify the applicable tools and techniques that have been developed during the years of NASA/Army cooperation and to present them in one place. A second objective is to present some conclusions based on the relevant work of the past in summary form. The third objective is to describe the NASA/Army transmission noise program so that industry, government, and universties can work together to achieve quieter helicopter transmissions.

This paper will present and discuss noise and vibration measurements taken on the U.S. Army OH-58 helicopter transmission. Measurements were taken in the NASA Lewis Transmission Laboratory, and in flight at the Ohio National Guard Facility at Akron-Canton Airport. Our current understanding of the nature and causes of gear and transmission noise is discussed, followed by a summary of the past work sponsored by the Army Propulsion Directorate and NASA Lewis that is applicable to the noise and vibration problem. Now there is a focussed attention on helicopter noise; current activity and plans for future work on helicopter noise are presented.

OH-58 HELICOPTER & TRANSMISION

The OH-58C Helicopter is the Army's Light Scout/Attack helicopter, which has a single two-bladed rotor and is powered by a 236 kW (317 rated output shp) gas turbine engine. The gross vehicle weight is 14.2 kN (3200 lb). The main transmission has a reduction ratio of 17.44:1, dry weight of 0.467 kN (105 lb), and is engine output rated for 201 kW (270 hp) continuous duty. The Army began receiving the OH-58's from Bell Helicopter Company in 1969. The OH-58 is a derivative of the Bell Model 206. The most recent Army upgrade of this helicopter is the OH-58 D model, rated at 339 kW (455 hp) at the main rotor. The commercial family of 206 versions has several models.

The Noise Problem

Historically, helicopters have been plagued by internal noise problems. Noise levels range from 100-120 dBa in the cabin. The sound can be from many sources, such as the transmission gear noise, the turbine engine compressor and exhaust noise, the rotor blades, and air turbulence. The transmission is a particularly troublesome source and is believed to be the main source of annoying noise in the helicopter cabin. The noise from the transmission enters the cabin following two paths, structure borne radiation and direct radiation (fig. 2). The magnitude of the direct radiation is a function of the acoustic power radiated from the transmission case, transmitted acoustically to the cabin outer walls, and transferred through to the cabin. Of course if there are any small openings in the wall between the transmission compartment and the cabin the sound will directly enter the cabin. The structure borne path is particularly hard to block because the transmission case and its mounts are an integral part of the lift-load bearing path. The transmission mounts must be strong and rigid: strong enough to support the entire helicopter by transferring the lift-load from the rotor blades to the air frame; and rigid enough for stable control of the helicopter. The stiff mounts pass the gear vibrations exceedingly well to the airframe, and the sound transmits to the cabin directly.

OH-58 Investigations

The measurement experience reported here was limited to the OH-58 helicopter (fig. 3). Vibration measurements of the transmission were previously reported for the OH-58 helicopter (refs. 2 and 3) and for two larger sized helicopters (refs. 4 and 5). The in-flight measurements of cabin noise were performed in a National Guard helicopter at the Akron-Canton Airport. Measurements of the in-flight vibration and noise are presented here for the first time.

OH-58 Transmission.

The test transmission is described in reference 2 and is shown in figure 4. It is rated for use where the engine output is 201 kw (270 hp) continuous duty and 236 kw (317 hp) at takeoff for 5 min. The input shaft, turning at 6180 rpm, drives a 19 tooth spiral bevel pinion meshing with a 71 tooth bevel gear. The input shaft is mounted on triplex ball bearings and one roller bearing. The 71 tooth bevel gear shaft is mounted on duplex ball bearings and one roller bearing. The bevel gear shaft drives a floating sun-gear which has 27 teeth. The power is taken out through the planet carrier. There are three planet gears of 35 teeth which are mounted on spherical roller bearings. The ring gear (99 teeth) is splined to the top case and therefore is stationary. The overall gear reduction ratio is 17.44:1.

NASA LEWIS TEST STAND

Figure 5 shows the NASA 500 hp helicopter transmission test stand, which was used to run the self-excited vibration tests (ref. 3). The test stand operates on the "four-square" or torque regenerative principle, where mechanical power is recirculated around the closed loop of gears and shafting, passing through the test transmission. A 149 kW (200 hp) variable speed dc motor is used to power the test stand and control the speed. Since the torque and power are recirculated around the loop, only the losses due to friction have to be replenished.

An 11 kW (15 hp) variable speed dc motor driving against a magnetic particle clutch is used to set the torque in the test stand. The output of the clutch does not turn continuously, but only exerts a torque through the speed reducer gearbox and chain drive to the large sprocket on the differential gear unit. The large sprocket is the first input to the differential. The second input is from the upper shaft which passes concentrically through the hollow upper gear shaft in the closing-end gearbox. The output shaft from the differential gear unit is the previously mentioned hollow upper gear shaft of the closing-end gearbox. The torque in the loop is adjusted by changing the electrical field strength at the magnetic particle clutch. The input and output shafts to the test transmission are equipped with speed sensors, torque meters, and slip rings.

NOISE AND VIBRATION MEASUREMENTS

The transmission was instrumented with accelerometers with a flat frequency response up to 10 kHZ, installed in the test stand and operated at 6060 rpm and load of 224 kW (300 hp). After reaching an equilibrium operating temperature of approximately 93 °C (200 °F), the accelerometer signals were recorded on a 14 channel FM tape recorder and later processed with a digital signal processor to get the vibration spectra.

Test Cell Measurements

Vibration spectra have been extensively measured for the OH-58 transmission in test rigs at the Corpus Christi Army Depot and at NASA Lewis (refs. 2 and 3). Measurements were made for a matrix of test conditions and thus it was determined that transmission speed had a significant effect while torque had a small effect on vibration amplitude. The highest magnitude of vibration consistently occurred at the spiral bevel gear mesh frequency for a variety of accelerometer locations. A typical spectrum is shown in figure 6, where the accelerometer was located in the plane of the planetary gear stage, just above the split-line between the top and bottom halves of the transmission housing.

In-flight Measurements

In-flight measurements were made in an Ohio National Guard OH-58 helicopter at the Akron-Canton Airport. Data records were recorded on FM magnetic tape and later analyzed using a spectrum analyzer. Microphone and accelerometer transducers were used. One of the accelerometers was placed near the split line of the transmission upper and lower housing in the approximate location that was used to obtain the result shown in figure 6. The objective of placing the accelerometer was to have a comparison with data collected in the test cell. A second accelerometer was placed on the transmission support base at the cabin roof. The purpose of this was to characterize the structure borne vibration by measuring the vibrations on the airframe at a point in the path of propagation. Figure 7 shows the results from spectrum analysis of the accelerometer measurements. Microphones were placed head-high in the vicinity of the copilot station and the aft cabin. The objective was to measure the noise perceived by passengers and, with the accelerometer signals in hand, thereby determine the severity of the noise components due to the transmission. Figure 8 shows the results from spectrum analysis of the noise measurements.

DISCUSSION OF RESULTS

In general, the vibration spectra contain many discrete frequencies where there is significant concentrated vibrational energy. The frequencies are identified with the gear mesh frequencies and the higher harmonics at integer multiples of the mesh frequencies. From the measurements in the test cell it was found that the single largest contributor to the transmission housing vibration was the spiral bevel gear mesh (fig. 6). The flight data were consistent with these findings (fig. 7(a)), except for some additional vibrational contributions, which have not yet been identified.

Analysis of the flight and test rig data indicates that the highest amplitude of transmission vibration occurs at the bevel gear clash frequency. However, for transducer locations other than directly on the transmission, the flight data presented some different conclusions regarding the effect of transmission vibration on the cabin noise. Flight data (fig. 7(b)) from accelerometers on the transmission mount at the cabin attachment point show that the vibration at the bevel and planetary gear mesh frequencies are equal to one another and to the peak amplitude of the bevel gear vibration measured on the transmission case (fig. 7(a)). It now becomes apparent that the transfer function from the housing to the mounting point has increased the relative significance the planetary gear vibration. This could be due to structural resonance or to vibration from the planetary gear and bevel gear being transferred along different paths resulting in an apparent increase of the planetary gear vibration at the transmission/cabin interface. Based on this observation, one might expect the planetary and bevel gears to contribute equally to the cabin noise.

The noise generated in the cabin by the transmission vibration is a function of the transfer function between the transmission and the cabin interior and the acoustic efficiency of the process. Since the process is unknown at this time, it is necessary to rely on the data provided by the cabin microphones (fig. 8). The noise spectra from the microphone measurements show a trend of decreasing amplitude as the frequency increases. This is because the higher frequency noise waves are more easily absorbed and dissipated in the acoustic transmission process. The spectra at the two locations (figs. 8(a) and (b)) differ only slightly, possibly due to standing wave patterns in the cabin.

The transmission related noise in the cabin is dominated by the planetary gear mesh frequency. This indicates that reducing the vibration generated by the planetary gears could significantly decrease the cabin noise level. Attention should also be directed to identification and reduction of the noise source that exists at frequencies below 400 Hz. Therefore it apppears that the most troublesome noise in the cabin is the lowest frequency gear noise as well as other low frequency noise the source of which is unknown at this time. In the cabin, the bevel gear noise is significantly below the planetary gear noise. It may be concluded that if the large amplitude of vibration for the bevel gear had occurred at the lower frequency of the planetary gear mesh frequency there would be even higher level of transmission noise in the cabin.

NATURE AND CAUSES OF TRANSMISSION NOISE

Noise generated by gears is due to many mechanisms such as mechanical impact of gear teeth, ejection of air and oil from between the gear teeth, the time varying stiffness of the gear mesh, movement of the load on the gear tooth, and errors in gear tooth geometry (refs. 6 to 10). Many of these mechanisms are inherent to transmissions and their elimination as a noise source is impossible. It is believed that kinematic error is the most significant source of noise and vibration in gearboxes. Kinematic errors are particularly troublesome for spiral bevel gears. The spiral bevel gears in a helicopter transmit high power at high speed, so elimination of kinematic errors can reduce the noise and vibration.

Kinematic Errors

Kinematic errors are defined as the deviations from a constant rate of turning of the driven gear while the driving gear turns at a constant rate (fig. 9). If a set of gears transmits motion without kinematic errors then they are said to have conjugate motion (horizontal dashed line of fig. 9). Each parabolic curve shown in the figure represents the kinematic error for one gear tooth as it moves through mesh. Intersection of the parabolic curves is where the load is transferred to the next gear tooth. It is this varying gear ratio that provides a source of vibration (noise) excitation to the gear system.

Conjugate Motion

Conjugate motion results if the vector normal to the gear tooth surfaces at the point of contact passes through the pitch point during the process of meshing (fig. 10). This requirement is satisfied for spur gears of involute tooth profile under very light load; when the load is high, the elastic deflection of the gear teeth upsets the condition of conjugacy. The involute system of tooth shapes for spur and helical gears can be described by simple mathematical expressions. In contrast to spur gears, there is no equation for describing the surface of a spiral bevel gear tooth. The surface coordinates of the points on the tooth must be calculated, based on the generating motions of the grinding machine.

As currently manufactured, spiral bevel gears do not have conjugate action. Spiral bevel gears with conjugate action were examined many years ago. It was found that if the gears had line contact between the teeth then they were very sensitive to shaft misalignment. This resulted in very poor performance: for even slight misalignment, the tooth contact moved to the edge of the tooth, causing very high contact stress, noise and poor life. To compensate for this sensitivity, the gears had to be made with something called "mismatch", which is a crowning of the tooth profile in both the lengthwise and profilewise directions. This reduced the sensitivity to misalignments but it also compromised noise, because conjugate action was lost.

The process of grinding teeth on a spiral bevel gear is a function of many different settings on the gear grinding machine. Nominally similar gears may result from several different sets of machine settings. Usually the machine settings are chosen, the gear and pinion are made, and the gear and pinion are tested in a fixture to see what kind of contact pattern exists between the teeth. Then the machine settings are adjusted to improve the contact pattern between the teeth, and the gears are ground again. The process may have to be repeated several times.

It is possible to determine the contact pattern and kinematic errors, based on a given set of machine settings. This procedure is extremely complicated and must be done using a computer (refs. 11 and 12). The basis for the mathematical methods is vector analysis and differential geometry.

The way of visualizing how the meshing of spiral bevel gears with zero kinematical errors takes place is similar to the spur gear example (refs. 13 and 14). Recall from the spur gear example (fig. 10), that the surface normal at the contact point always passes through a fixed point in space--the pitch point. For the bevel gears, which can be imagined as two pitch cones rolling against one another at a pitch line (fig. 11), the normal to the tooth surfaces at the contact point should pass through a fixed line in space,--the pitch line.

If we require that the tooth surface normal vector always passes through the pitch line, and if we require that the tooth surface normal be constrained to move parallel to itself as the contact point shifts across the tooth, then the gears will have zero kinematic errors. The problem becomes one of how to achieve this type of motion by the intelligent selection of the machine settings for the gears to be manufactured. This has been accomplished through the kinematic modelling of the manufacturing process (ref. 15), which results in a set of nonlinear equations that must be solved simultaneously using numerical methods.

AVAILABLE SUPPORTING TECHNOLOGY

Helicopter cabin noise is significantly affected by the transmission and in particular the gears in the transmission. Gears are a source of high vibration and have been the subject of study for years in research investigations too numerous to mention here. There has also been a significant amount of NASA/Army sponsored research that is pertinent. For that work to be truly useful, it must be brought to the attention of gear and transmission designers and researchers. The work falls into the categories of Dynamic Load Analysis, Tooth Profile Modification, and Measurement Tools.

Dynamic Load Analysis

Dynamic load is defined as the load on the gear tooth as a function of time and position as the gear rotates. Dynamic loads are caused by the interaction of the mass of the gear and driven elements and the stiffness of the gear tooth. Usually the gear system is modelled with second order differential equations with time varying stiffness parameters. The stiffness changes because the number of gear teeth in mesh varies as the gears rotate. The average number of teeth in mesh is called the contact ratio. The Hamilton Standard division of United Technologies, under a NASA contract has developed two computer programs for the calculation of gear dynamic loads (refs. 16 and 17). Computer program GRDYNSNG is for a single pair of spur gears in mesh. The model includes two rotational degrees of freedom, and time varying tooth stiffness. Contact ratios between one and three are possible, and variations of the tooth from involute form are possible. An option permits a buttress form tooth, which has a lesser pressure angle on the drive side than the coast side. Computer program GRDYNMLT extends the capability of GRDYNSNG to include multiple gear mesh conditions such as present in a planetery gear stage.

The Cleveland State University has developed computer program PGT which calculates dynamic loads on planetary gear trains. The dynamic model has 9° of freedom, and is able to analyze a planetary gear train with three planet gears. A variable mesh stiffness is used, including the effects of planet phasing and location errors. The analysis can be used to study static and dynamic load sharing among planets, tooth errors and intentional profile modifications (ref. 18). The computer programs PGT and GRDYNMLT have been compared with each other and with experimental data from full scale transmission experiments (ref. 19).

The gear tooth stiffness that is modelled in most gear dynamic computer programs is static tooth stiffness. A more realistic stiffness model has the load moving across the profile of the tooth as the gears rotate; the effect becomes very significant at high speeds. The influence of moving load was investigated under a NASA sponsored grant at Michigan Technological University (ref. 20). The University of Cincinnati investigated the effect on involute and straight tooth forms (ref. 21) and developed a computer program to study the effects of parametric variations on gear dynamic load (refs. 22 to 23).

Computer program TELSGE was developed by Northwestern University, where the gear tooth stiffness was determined by finite elements. The model is for a simple spur gear mesh with 2° of freedom, and includes the effect of the thin film of lubricant between the gear teeth (ref. 24). The problem of dynamic loads in spiral bevel gears is extremely complex due to the tooth geometry and the additional degrees of freedom necessary for even a simple mesh of two gears. In reference 25 the concepts of reference 24 were extended to the case of spiral bevel gears using 12° of freedom. The study was for a particular pair of gears and the results are not generally applicable to all spiral bevel gear pairs. There is continuing work at Bolt Beranek and Newmann to investigate the noise generating mechanism for bevel gears (NASA contract NAS3-23703).

Tooth Profile Modification

Tooth profile modification, especially tip relief is a commonly used method to control the amplitude of dynamic load in gears. There is no concensus among researchers on what is the best or optimum shape of tooth profile modification. A study of the problem was conducted by Bolt Beranek and Newmann using a conventional modification consisting of linear tip relief, in comparison to a new profile that was determined on the basis of minimum vibration excitation (ref. 26). Significant differences in the dynamic forces transmitted by the teeth are predicted for the two cases. In contrast to all the methods reviewed so far, the Bolt Beranek and Newmann approach uses the frequency domain, rather than integrating equations of motion in the time domain.

The noise from spiral bevel gears is thought to come primarily from the kinematic errors that are inherent in the manufacture of the gear teeth. A study to determine a way to manufacture the gears eliminating the kinematic errors was conducted by Litvin at the University of Illinois (ref. 27). The result of the analysis is to provide new settings for the bevel gear grinder so that the gears are manufactured with a conjugate tooth shape.

Measurement Tools

The measurement of gear noise is usually performed using accelerometers placed on the gearbox housing. As the measurements have shown in this paper, the highest contributor to noise can best be found under realistic conditions of running the gearbox, while using a microphone. The conditions in a test cell are not conducive for exacting microphone measurements due to the presence of other machinery and accessories in the test cell that contribute to the noise. In addition, the noise field is complex and a single microphone in the test cell may not be definitive. The element of personal safety in the test cell often precludes manual movement of microphones during a test run. The problem of test cell measurements was studied by Flanagan and Atherton (ref. 28) and their solution was to use a robot-held acoustic intensity probe. Acoustic intensity measurements are made with two closely spaced microphones which measure the sound power passing a stationary point. The total sound power emanating from a source (the gearbox) can be obtained by measuring the intensity around the source and integrating over the surface enclosing the source. Since intensity is a vector quantity, sources external to the gearbox will make a net zero contribution. The advantage of the intensity method is that a specially treated anechoic chamber is not needed. The robotic acoustic intensity measurement system (RAIMS) is shown in figure 12.

Kinematic error has been explained previously. The theoretical aspect of kinematic error is well appreciated by gear theoreticians but there is currently no practical machine to measure the kinematic error of gears while the gears are loaded. A design study for such a machine was conducted by Houser (ref. 29). Until such a machine is built, progress in gear noise reduction will be limited because of our incomplete understanding of the influence of transmission errors and gear tooth flexability on noise.

These tools need to be exploited and further developed to improve their usefulness in specific application to solving noise and vibration problems in helicopter transmissions.

CURRENT RESEARCH APPROACH

There are three general fronts on which we can attack the helicopter cabin noise problem: by using acoustic treatments in the cabin, by using isolation methods, and by reducing the source of noise excitation (fig. 13). The acoustic treatment approach has generally worked in the past but at the expense of added weight as well as added cost. New methods such as advanced lightweight sound treatments, and optimum usage of those treatments should be investigated. The work should emphasize a minimum weight penalty for the necessary noise reduction. The isolation approach can be used to manage the energy paths of the vibration and noise and prevent them from efficiently passing the energy to the cabin interior. New approaches to isolation can result from structural modification with special attention to the acoustic/dynamic coupling. Vibration absorbers should also be investigated. These could be active or passive and placed anywhere in the energy path. Reduction of the noise by reducing the vibration at the gear mesh is attractive because it could have benefits of increased life and reliability as well. The gear mesh dynamics could be improved with new tooth forms for low noise. Increased damping mechanisms within the gearbox could absorb the energy being transmitted to the cabin. An improvement of the overall transmission system dynamics could be achieved with new design techniques for housings, bearings, gears, and shafts based on dynamic and vibrational criterion. Advanced bearing mounts, and damper pads could result in lower dynamic loads.

The role of NASA Lewis and Langley Research Centers will be to cooperatively research the cabin noise problem, to concentrate in the traditional areas of their respective expertise, and to provide the enabling technology to the industry for use in effectively reducing cabin noise.(fig. 14). Langley will concentrate in areas such as cabin noise characterization, structural modification and advanced treatments. NASA Lewis will investigate ways of quieting the gearbox. The gear mesh will be studied for ways to reduce dynamic loads with new tooth forms and tooth profile modification. Damping techniques, detuning, and system optimization will be investigated.

CONCLUDING REMARKS

This paper has described work conducted by NASA Lewis in measuring cabin interior noise and in relating the noise spectrum to the gear vibration for the Army's OH-58 helicopter. Noise and vibration data were collected and analyzed from flight tests and from ground-based tests in a transmission test stand. Our current understanding of the nature and causes of gear and transmission noise was discussed. This paper summarized NASA/Army sponsored research that applies to transmission noise reduction. The continuing research program was also reviewed. The following remarks summarize the important conclusions.

1. During the last 20 yr helicopter cabin noise due to the gear train has continued to increase with the power-to-weight ratio of the transmissions. This has required that additional sound treatment material be added, causing a weight penalty to the total helicopter system.

2. A large portion of the cabin noise is contained in discrete frequencies associated with gear mesh behavior. It is believed that significant reductions of noise can be achieved if gear vibrations within the transmission are reduced. The single largest contributor to the transmission vibration is the spiral bevel gear mesh and the planetary gear train is a major source of cabin noise. The authors believe that the kinematical errors of the gear mesh have a strong influence on the noise.

3. Several analytical design tools have been developed that will be useful in reducing gear noise excitation, viz., minimum excitation gear profile design techniques and dynamic load analysis computer codes. Also, the robotic acoustic intensity system (RAIMS) for measuring noise in a noisy test cell environment will be a valuable tool for measuring and comparing the relative noisiness of advanced components and transmission systems compared to baseline technology levels. These tools need to be exploited and further developed to improve their usefulness in specific applications.

4. NASA Lewis and Langley Research Centers with the cooperation of the collocated Army Research Centers are continuing to perform focussed research for reducing the cabin noise of future helicopters. This will be accomplished by developing the enabling technology for reduced gear noise, reduced noise propagation to the cabin and advanced acoustical treatments of the cabin interiors.

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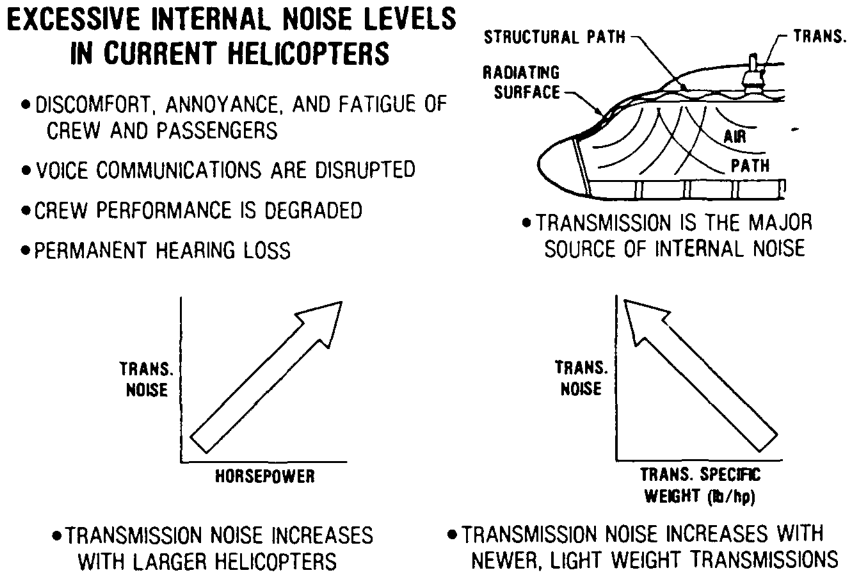


FIGURE 1. - THE HISTORIC TREND FOR THE PAST 20 YR WAS HIGHER POWER, GREATER POWER-TO-WEIGHT RATIO AND INCREASING TRANSMISSION NOISE.

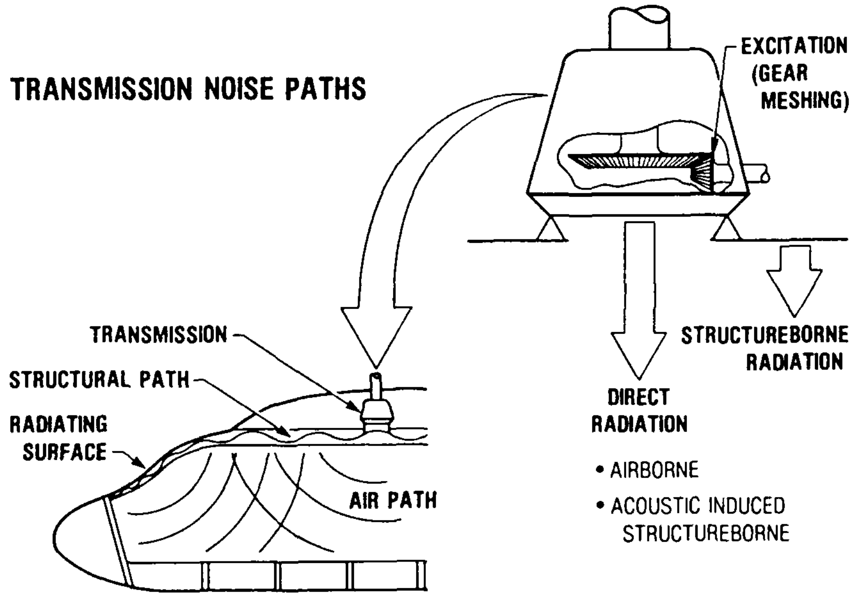


FIGURE 2. - THE NOISE FROM THE TRANSMISSION TRAVELS VIA STRUCTURAL-BORNE AND DIRECT AIR-BORNE PATHS TO THE CABIN WALLS AND IS RADIATED TO THE CABIN INTERIOR.

FIGURE 3. - THE OH 58 IS THE ARMY'S LIGHT OBSERVATION HELICOPTER. THE CIVIL VERSION (SHOWN) IS THE MODEL 206.

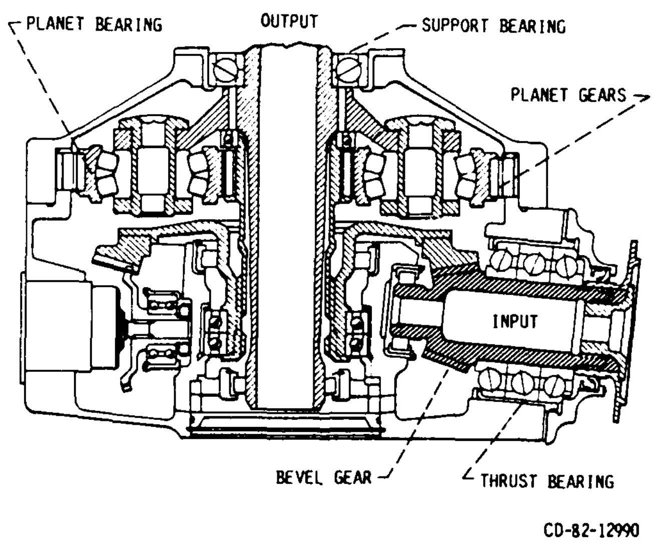


FIGURE 4. - OH 58 HELICOPTER TRANSMISSION.

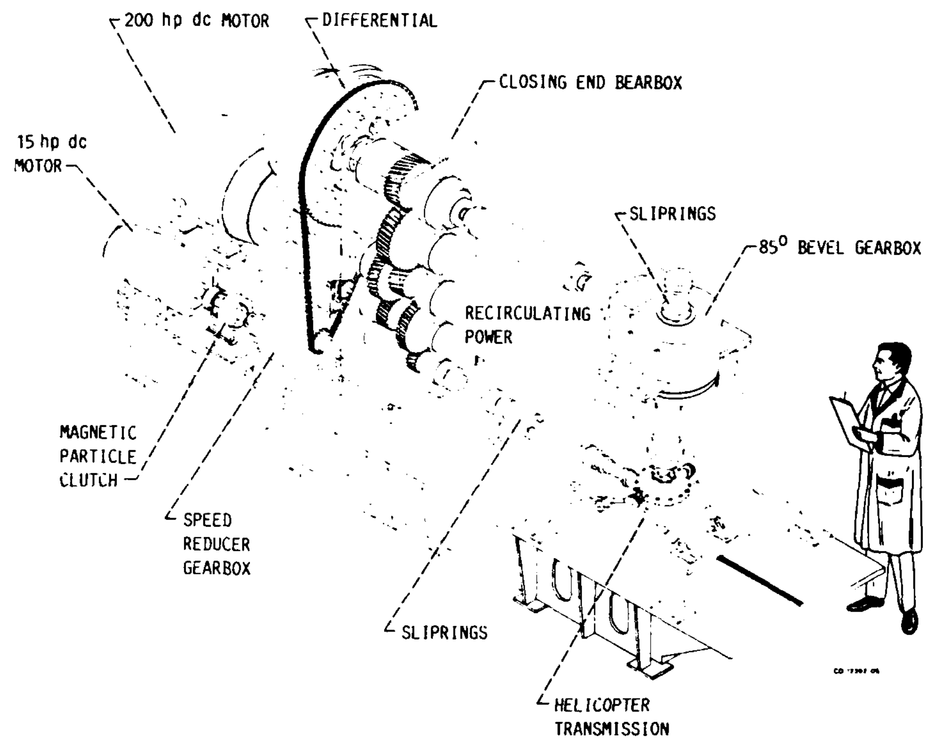


FIGURE 5. - THE NASA LEWIS 500 hp HELICOPTER TRANSMISSION TEST STAND.

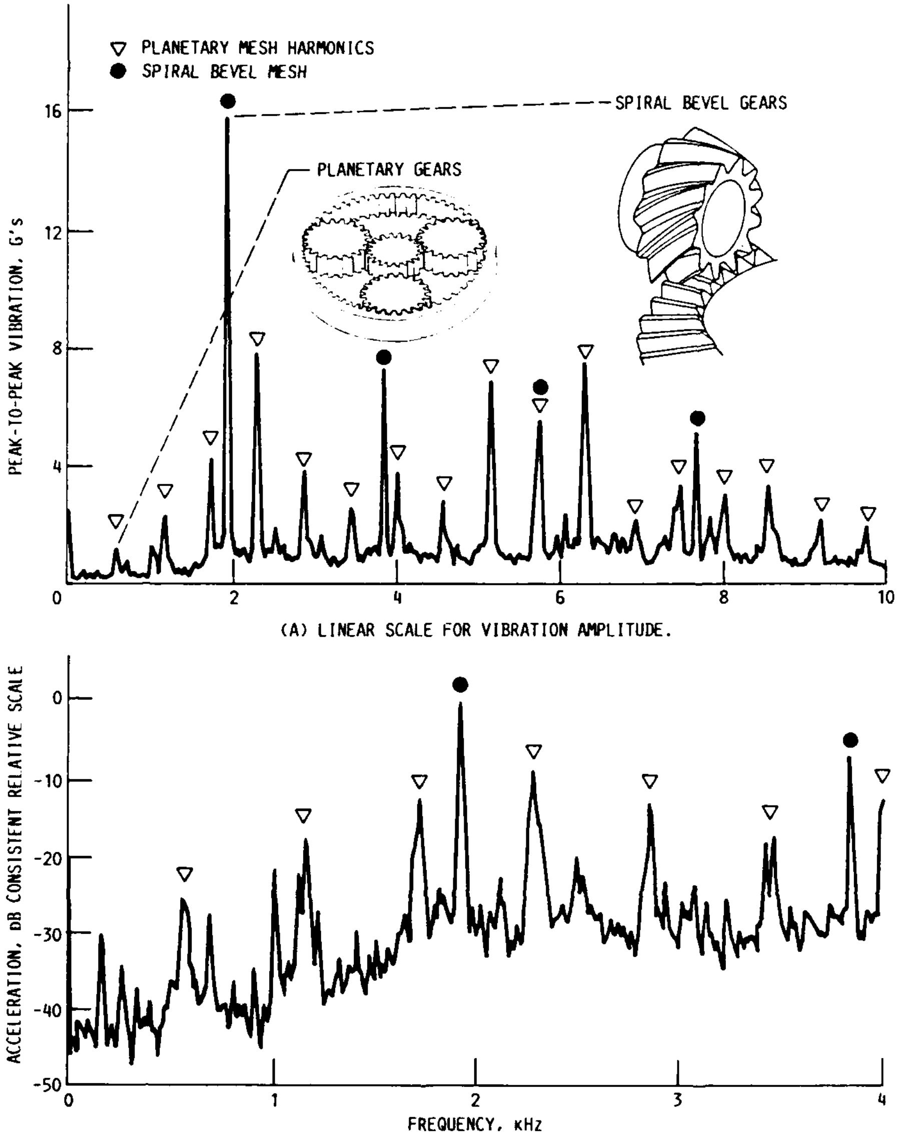
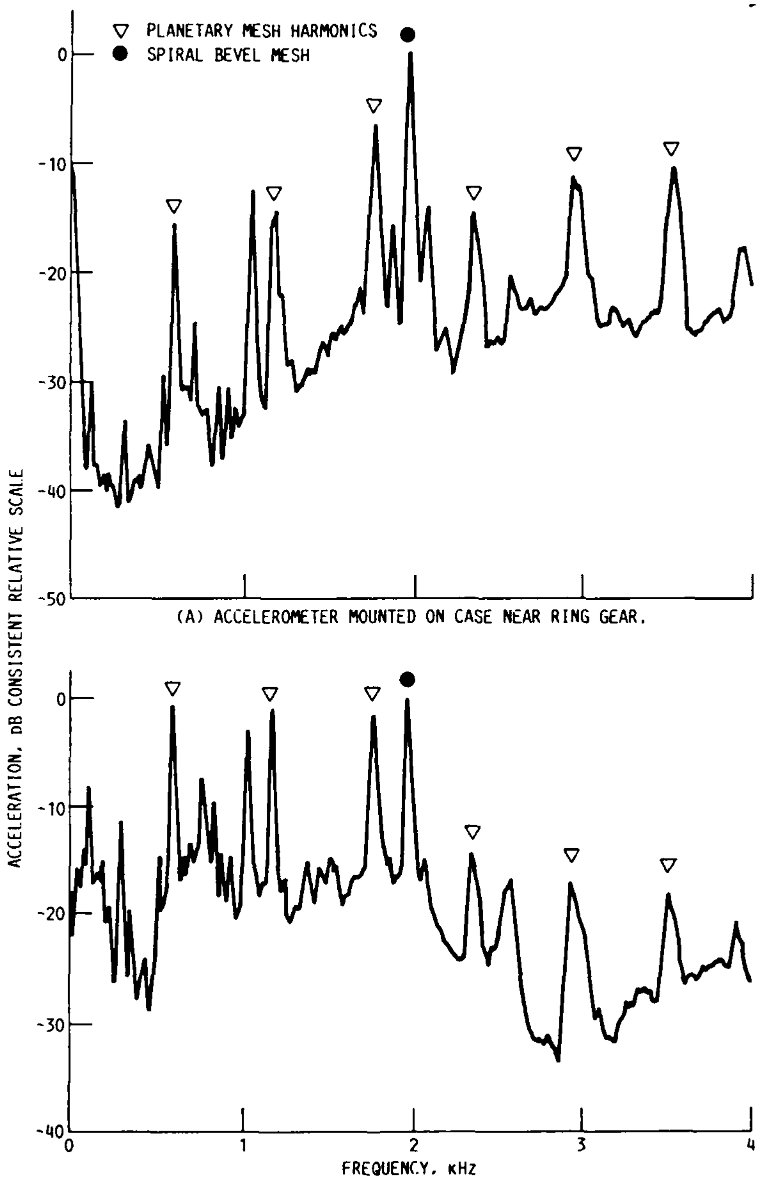
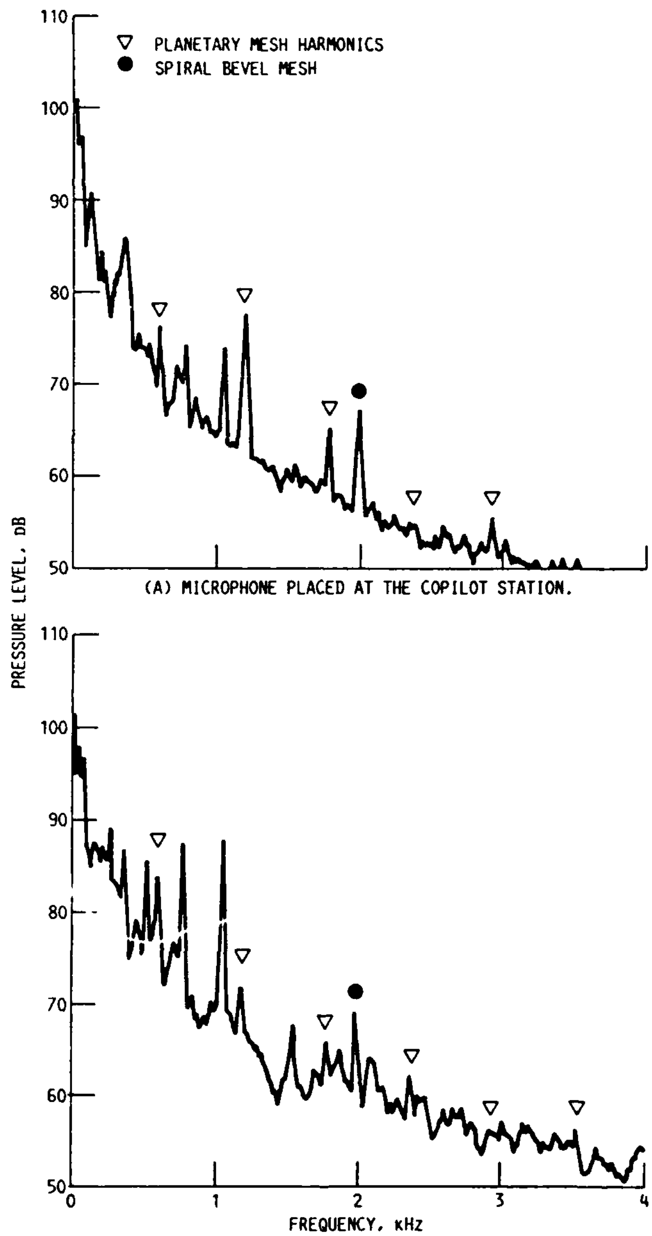


FIGURE 6. - VIBRATION SPECTRUM OF AMPLITUDE VERSUS FREQUENCY AS MEASURED IN NASA TEST STAND. ACCELEROMETER MOUNTED ON CASE NEAR RING GEAR. NOTE THAT VIBRATION ENERGY IS CONCENTRATED AT THE GEAR MESHING FREQUENCIES AND THEIR HIGHER HARMONICS.



(B) ACCELEROMETER MOUNTED ON TRANSMISSION SUPPORT BASE AT ATTACHMENT TO CABIN ROOF.

FIGURE 7. - VIBRATION SPECTRUM OF AMPLITUDE VERSUS FREQUENCY AS MEASURED IN FLIGHT. VIBRATION ENERGY IS CONCENTRATED AS A FEW DISCRETE FREQUENCIES AND THEIR HIGHER HARMONICS. NATURE OF THE SPECTRUM IS SIMILAR TO TEST CELL MEASURED DATA.



(B) MICROPHONE PLACED IN THE AFT CABIN.

FIGURE 8. - NOISE SPECTRUM OF SOUND PRESSURE LEVEL VERSUS FREQUENCY MEASURED DURING FLIGHT. PEAKS IN THE SPECTRUM HAVE BEEN IDENTIFIED AS TRANSMISSION ORIGINATED. THE HIGHER FREQUENCIES ARE ATTENUATED.

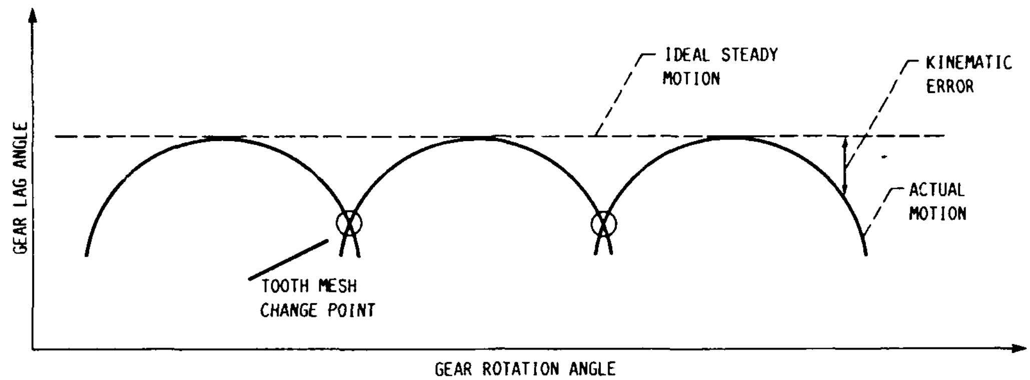


FIGURE 9. - TYPICAL TRANSMISSION ERRORS AS A FUNCTION OF GEAR ROTATIONAL ANGLE.

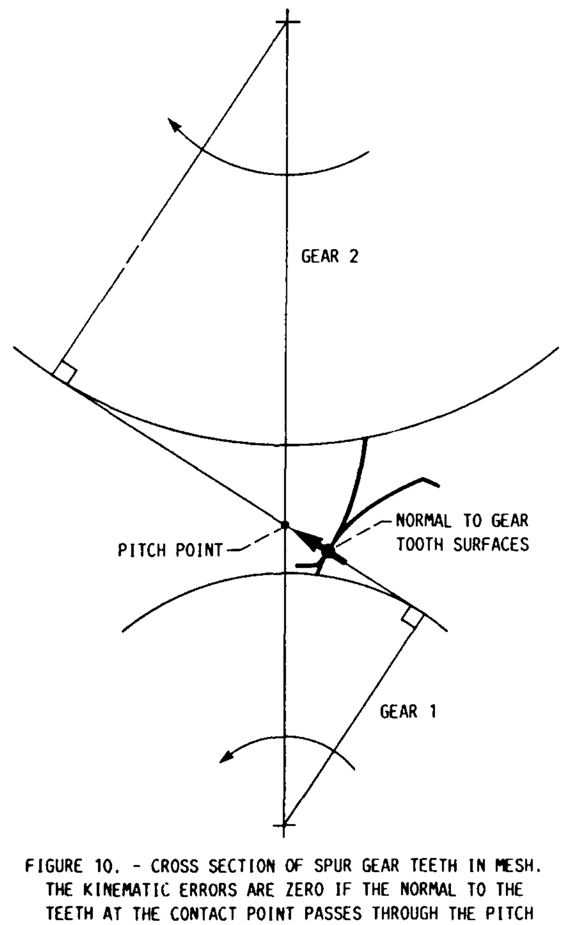


FIGURE 10. - CROSS SECTION OF SPUR GEAR TEETH IN MESH. THE KINEMATIC ERRORS ARE ZERO IF THE NORMAL TO THE TEETH AT THE CONTACT POINT PASSES THROUGH THE PITCH POINT AS THE GEARS ROTATE.

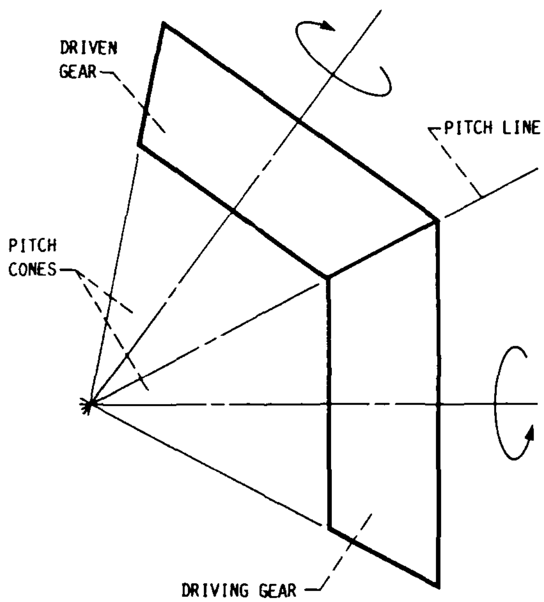


FIGURE 11. - BEVEL GEARS ARE REPRESENTED BY THEIR PITCH CONES IN ROLLING CONTACT. THE PITCH LINE IS THE CORRESPONDENT TO PITCH POINT FOR SPUR GEARS.

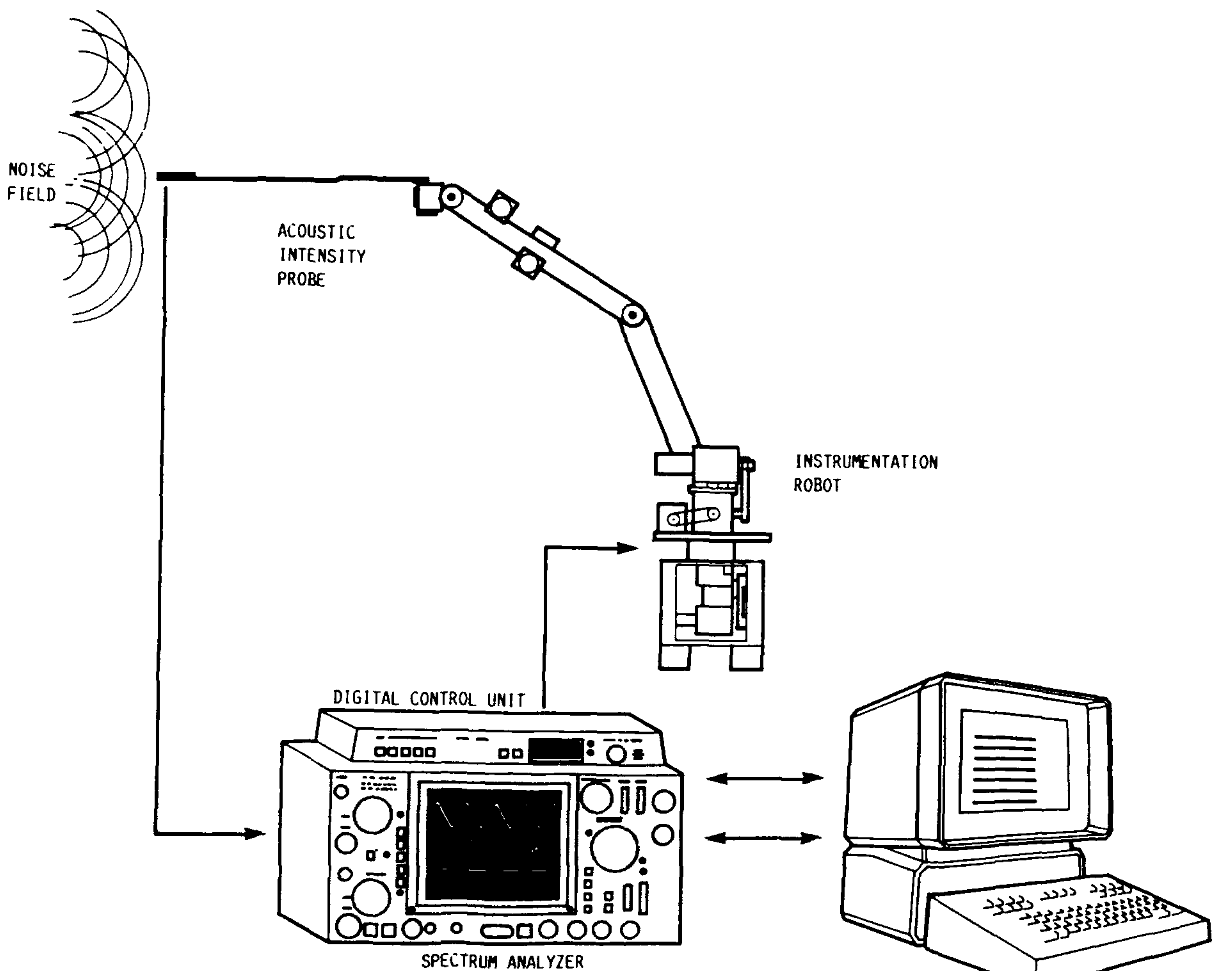


FIGURE 12. - SCHEMATIC OF RAIMS (ROBOTIC ACOUSTIC INTENSITY MEASUREMENT SYSTEM).

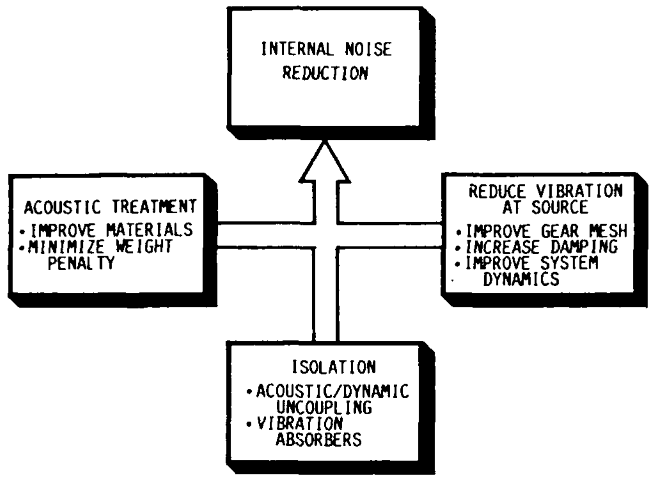


FIGURE 13. - NOISE REDUCTION CAN BE ACHIEVED BY USING ACOUSTIC TREATMENTS, ISOLATION TECHNIQUES, AND REDUCING THE SOURCE. LEWIS WILL CONCENTRATE ON REDUCING THE SOURCE.

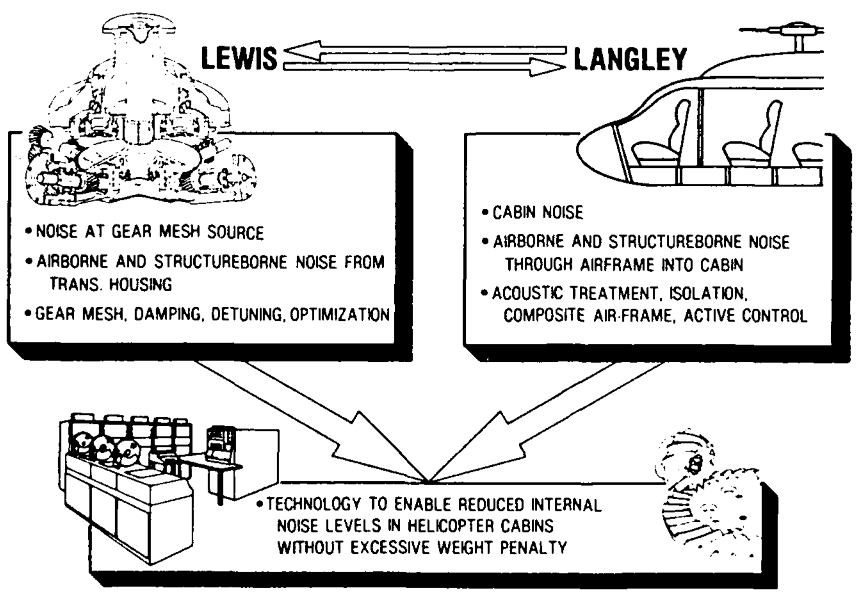


FIGURE 14. - THE ROLE OF LEWIS AND LANGLEY IS TO FOCUS ON THE AREAS OF THEIR EXPERTISE AND COOPERATIVELY PROVIDE ADVANCED TECHNOLOGY TO THE INDUSTRY FOR USE IN EFFECTIVELY REDUCING CABIN NOISE.