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In this report, the experimental work carried out prior the proposed main trials are described in detail. The report consists of two main sections. The first section deals with the bench tests carried out with the Central Control Unit CCU for the active fin stabilisation system. In section two, the measurements recorded at sea during the tuning experiments in December 1992 are described. An overall view of the LFE stabilisation work has been included at the beginning of this report.

1. An Overall View

LFE stabilisation, in essence, is an old concept. In the past, the idea has been explored under the name of stabilisation to the apparent vertical. This type of work came about in the literature mainly in the wake of the successful application of the active fin stabilisers. Despite the good overall performance of active stabiliser to counter-act ship roll motions, it has been suggested that the lateral acceleration is also of importance to the performance of a ship. In short, the desire to stabilise a ship to the apparent vertical or LFE has stem for the wish to provide a more stable platform in terms of lateral acceleration, which would enhance the comfort or the operability of a ship.

The LFE stabilisation work carried out at the Ship Science Department can be conveniently divided into two parts - a numerical and an experimental approach based on sea-trials. The numerical part involves the use of a sea-keeping program based on strip theories. The computer program has been modified to allow for LFE feedback which was compared with the normal roll feedback in a conventional controller. In the first instance, encouraging results were found with LFE stabilisation using fins (see Appendix B). It was found that good performance could only be achieved when the ship was tuned for LFE.

The idea was then applied using the rudder as the control surface. However, in this case, although some improvement was reported, this was found to be not as effective as rudder roll stabilisation (see Appendix C). Further work was done to explore the use of the rudder with filters, which would improve the frequency response of feedback signal (see Appendix D). Although substantial improvement was achieved, the results were still lower than a straight forward rudder roll stabilisation strategy.

On the sea-trials work, a number of adaptations have to be made to the active fin controller for this purpose. This consisted mainly of a signal conditioning device (LFE signal box) that can transmit a LFE signal compatible to the Central Control Unit CCU. A new Eprom chip was also needed in the CCU which would allow the experimenter to change the controller coefficients. This was provided by Brown Brother's Limited.

The design and development of the LFE signal box are documented in Appendix C and D. Its sole function is to transform a d.c. LFE signal into a synchro form for CCU. The LFE signal is provided by a high quality accelerometer. Based on the experiments described latter on in this report, the whole set-up for the trials have proven to be robust and reliable. A procedure for the main trials has also been developed.

Unfortunately, the date of the main trials has been postponed twice due to various reasons. At this stage, the concept of LFE stabilisation using the active fins has yet to be proved. However, all the necessary components are at in hand for the final stage of the investigation.

2. CCU Tests

As part of the preliminary experiments leading up to the main trials, tests were carried onboard a frigate when the ship was alongside. The main objectives of these tests were as follows:

(a) to make sure that all the equipment built are totally compatible with the fin controller hardware onboard ship,

(b) assessing the performance of all the equipment built for the LFE trials under conditions as close to the main trials as possible,

(c) set up an efficient procedure for the rigging up, operating and de-rigging of equipment, and

(d) derive the frequency responses of a set of tuning frequencies and the sensitivity of the overall gain K_4.

The idea behind these tests was similar to the forced roll trial conducted earlier (see Crossland 1992). However, instead of using the built-in Service Module (SM) of the CCU, an alternative input signal had to be provided for the present purpose. This input signal, of known frequency and amplitude, was generated by a mechanical oscillator (plate 1) with the accelerometer (plate 2) to be used for the main trials. This signal was then fed into the LFE signal box (plate 3) which converts a d.c. signal into a synchro form required by the CCU. To allow for the selection of LFE or roll control signal, a turn-key switch box (plate 4) was used to interface between these two signals and the CCU. The demand signal for the fins provided by the CCU would control the fin movements, which in turn would affect the ship motion. The overall signal path is shown in diagram 1. In the present case, the signal path up to point D only is of interest.

After some initial trial and error, objective (a) was accomplished. All the settings were thoroughly validated by the subsequent testings in objective (b) and (d). The problem of a stiff LFE response reported previously was not encountered, therefore the need of the modifications to the LFE signal box described in the last report (Appendix D) was vindicated. At one stage there were some difficulties in deciding the signal combination in interfacing the synchro signal to the CCU, as six combinations of a three phase signal were possible. Fortunately, with the help of reference (1), which was provided during the tests by the officer in charge, the correct wiring was established.

All the equipment was now ready to be used for objective (d). As with objective (c), a separate document has been produced (Tang 1993), which describes a procedure that can be followed for future LFE trials.

In the remaining sections of this part of the report, the results derived from the objective (d) experiments will be discussed. First of all, a set of controller parameters was required. To derive this set of controller values, the procedure described in Marshfield 1980 has been applied at a range of frequencies over the phase response curve derived from the forced roll tests (Crossland 1992). These different controller settings in effect provided different amounts of phase-lead to the system. These sets of control values (K_1 to K_3) have to be converted into a format compatible to the CCU. This conversion procedure is described in detail in appendix A, which was provided by Brown Brothers and Company Ltd. For the frigate employed for the experiments, these control parameters are given in Table 1. (Note that as the tuning frequency increases, the K_3 value increases with a corresponding decrease in K_1.) In deriving these values, only the phase-lag due to the fins were included. Therefore the frequencies are labelled nominal frequencies. It was found in these tests that the controller phase-lag was in fact appreciable. Unfortunately their magnitudes could not be calculated from the forced roll tests and were not available before these tests.

2.1 Amplitude and Phase Response

Due to the constraint in time, tests were only carried out at nominal frequencies of 0.7, 0.67, 0.6 and 0.54. Six oscillation frequencies were selected to cover the range of period of interest and the oscillation amplitude was set equivalent to about six degrees roll at the oscillator. The resulting fin demand was recorded by reading the fin output at the SM module (1002 for fin demand) whilst the phase between point A1 and C were derived using a HP frequency analyzer. As the fin demand was a digital output that changed rather rapidly in time, the accuracy of these readings would not be as high as could be expected. There was possibly a ten percent margin in the reading error. For the phase readings, due the low resolution at these relatively low frequencies, the accuracy would be expected to have a margin of 5 degrees. Nonetheless, within these limitations, the data derived did give a good qualitative and a quantitative measure of the parameters of interest. The overall gain K_4 was set to four, which was about a third of the Brown Brothers' value. A dummy speed of 15 knots was introduced. Without this, no fin demand could be generated by the CCU. (This was done by dialling in 3015, 2010 and 2005 in the Service Module at the beginning of the test. To cancel this request, dial in 2009.)

A validation test was first carried out to see if there was any difference between the responses derived from the original and the new Eprom for the LFE work. It was found that the results were practically the same and the error margins were typical of those mentioned above. In all the tests discussed in the following two sections, only the demanded fin angles were used. This was considered appropriate since from the forced roll trials, it was found that the demanded to actual fin angle ratio was the same in the range of frequency of interest. This was confirmed in some initial testings, in which the fins were engaged. During the time when the fins were actually used, which lasted almost about three hours in total, very smooth fin motions were found (unfortunately, no time-histories were not recorded) As useful information could be derived without actually engaging the fins, it was decided to disengage the fins in case of a mishap.

The phase and amplitude response of the tuning frequency tests are given in figure 1 and 2, using the original Brown Brothers' values as a reference (see Table 1). It can be seen from fig.1 that the fin demand rises sharply at high frequency oscillation in Brown Brothers' case. This is in keeping with the remark that 'the CCU does not like high frequency '. This trend is also exhibited in the case with nominal frequency 0.54. This suggests that poor high frequency performance would also be expected from these K values and this setting probably marks the lower limit to the tuning frequency. It should be noted that the fin demand in the Brown Brothers' case varies more sharply compared with the other three nominal frequencies, hence more sensitive to frequency changes of waves in the seaway. The effective overall gains among the four nominal frequencies are quite different, with the 0.54 case having the highest overall gain.

Turning to fig.2, it is apparent that the different amounts of phase-lead are introduced by different tuning frequencies and the differences in phase-lead are fairly well maintained across the range of excitation frequencies. Therefore, in future work, tests would be needed only at one tuning frequency. The frequency response at intermediate frequencies can be estimated. The Brown Brothers' response is very different from the rest, showing a fairly sharp rise towards the high frequency. Near the natural roll frequency of 0.6 Hz, the response of the 0.67 case is the nearest to the Brown Brothers'. This would imply that for a roll dominant response near this frequency, the K settings of the 0.67 case should give comparable performance to the original Brown Brothers' case, which is for a pure roll signal.

Hence, from the tuning point of view, this would set the upper limit to the tuning frequency.

A comparison with the predicted values indicated that at the natural roll frequency, which was about ten seconds, a difference of about 20 degrees was found. Therefore, the phase-lag of the controller at this frequency would be about 20 degrees. From the previous forced roll trials, it was found that the fin servo and hydraulic mechanisms introduced about 18 degrees of phase-lag to the signal at this frequency. This means that the overall phase-lead provided by the Brown Brothers' controller settings would be about 45 degrees. Compared to a phase-lag of about 55 degrees of ship dynamic response, this phase-lead of about 45 degrees is acceptable according to Marshfield 1980.

In the actual trials, additional phase effects introduced in the filter of the LFE signal box could be significant. A few spot checks were carried out, which verified the predicted phase response in the filter design given in fig.3. It is shown in fig.3 that within the range of oscillation period of interest, the additional phase effect would be less than 10 degrees. This is considered satisfactory as the phase effect is also zero around the natural roll frequency. Less appreciable phase effects can be achieved, but this would be in the expense on the quality of the signal i.e. more noise in the signal or more of the higher frequency content which is not so agreeable to the controller.

2.2 Overall Gain Effect

It was anticipated that during the tuning stage of the main trials, the allocated time would be quite limited. Therefore it would be useful to know how the step changes in the overall gain K_4 would affect the fin response so that the desired K_4 could be arrived at more quickly. The tests carried out were similar to those in the last section. However, with the time available, only two oscillation periods were used, which were 10 and 7 seconds.

In fig.4, the oscillation period was 10 seconds. It can be seen that all the four different cases have fairly similar slopes, i.e the fin demand increases roughly by the same amount with increasing K_4. The average slope was about 2 degrees/ unit K_4. Suppose the r.m.s. fin angle was 10 degrees at the tuning stage of the main trials and the margin was supposed to be within 10%, i.e. the desired overall gain would produce a r.m.s. fin response between 9.0 to 1.1 degrees. Judging from the slope of 2 degrees/ unit K_4 above, the resolution in K_4 may be too coarse for the 10% margin proposed. Similar results are found in fig.5, where the oscillation frequency is 7 seconds. In both cases, the increase in fin demand varies linearly with increasing K_4.

In a few of these tests, the K_1 to K_3 were increased to twice or reduce to some fraction of their nominal values in Table 1 whilst keeping K_4 the same. It was found that the resulting fin demands were practically the same as varying the K_4 alone accordingly. Therefore, should any of the K_1 to K_3 values exceed the limit of 100, the maximum setting in the parameter card, the K_4 value can be used to compensate for this restriction. The linearity of the system at the input stage was also examined. This was done by increasing the amplitude of oscillation of the mechanical oscillator and recording the input demand at SM 1004. The results are plotted in fig.6, which show a linear response. However, there is a factor of 2 in the gain at the input stage before the signal was processed by the CCU. This factor was not documented in the references provided by the manufacturer. As the overall gain in the LFE signal box needed to be fine tuned at this stage to the level consistent with the gyro input, this unknown factor of 2 had created a great deal of confusion and uncertainty about the adjustments. However, this has now been resolved and the reading displayed at SM 1004 is only half the actual value.

2.3 Tests at HMS Sultan

To complement the work that had been carried out with the frigate, additional CCU tests were planned to be carried out at HMS Sultan. However, after some initial tests, it was found that the Eprom in this CCU was an old version, which could not accept the new Eprom. Nonetheless, some tests were carried out. These were done by altering the period parameter within the CCU, whilst the basic experimental procedure remained the same as those described above. The results are given in fig. 7 and 8, which show similar trends to those of the Brown Brothers' case in fig. 1 and 2. In particular, in fig.8, where the result from the frigate was also plotted, it is quite clear that the two CCU systems are practically the same. Therefore, providing that a new Eprom can be made available to Sultan, it would be much better to carry out the CCU tests there instead of the ship.

2.4 Final Comments

During the three day testings at Lancaster, a great deal of experience has been gained with the new equipment and the way in which one can proceed with the trials. Besides the data derived from these tests, a few additional comments should be offered.

(1) The sockets in the CCU are of a military standard. Despite the careful design, it is not easy to plug in the cables correctly to achieve the necessary contact. Therefore, it is a good practice to check thoroughly that whenever a new signal is plugged into the CCU, it is screwed into the socket perfectly.

(2) As was mentioned earlier, the fins were engaged at one stage during the tests. In two of these tests, vibrations were detected in the ship, which were due to the fins oscillating at high frequency. This was possibly an indication of the fins operating in an unstable region (controller-wise). No apparent reason could be attributed to the cause of this effect. In subsequent tests when the fins were turned off, it was found that the demanded signal had a higher harmonic component which was about three times the excitation frequency set at the oscillator and the amplitude about a third the corresponding value. However, the input signal from the oscillator was found to be purely sinusoidal. For some unknown reason, the CCU seemed to have corrupted the signal or some kind of interference had taken place. *Similar* misbehaviour of the CCU was mentioned by Chief Ottison. Therefore, one should be aware of this *strange* behaviour of the fins and always on the lookout for the presence of high frequency vibration. Under normal working conditions within the ship, where other machineries are running, this could be a difficult task. If the operator suspects that the fins are running strangely, the fins should be turned off at the SM with 2002. The Chief should be consulted if this behaviour persists. It has been found that the fins would return to normal when the CCU is switched on after the whole system has been powered off.

(3) At one stage, the LFE signal was recorded directly from the LFE signal box. However, for no apparent reason, the recording equipment interfered with the signal transmitted to the CCU. This could produce potentially dangerous consequences. Therefore, any measurements to be taken directly from the LFE signal box should be checked carefully for interference and if possible to be avoided.

3. LFE Experiments onboard a Frigate

In early December 1992, during a passage between two ports, LFE stabilisation was experimented onboard a frigate. These experiments were carried out on an opportunity basis and with a minimal disruption to the ship's program. Despite these restrictions, valuable experience has been gained with LFE stabilisation.

In these experiments, the motion and the fin responses were recorded when the ship was running under roll stabilised, LFE stabilised or unstabilised mode. The control-loop for the two modes of stabilisation is shown in diagram 2. Whilst the ship's gyro provided the roll signal, an accelerometer was placed in the hangar to provide the LFE input to the CCU. A three-component accelerometer was placed close to this LFE accelerometer to record the vertical, lateral i.e. LFE and longitudinal accelerations. The locations of these two accelerometers are given in Table 2. The demand and the actual fin motions were taken from two measurement points in the CCU. In total, seven channels of time-history signal were recorded:

(1) roll from the gyro
(2) fin demand from the CCU
(3) fin actual from the CCU
(4) LFE from the 3-component accelerometer
(5) longitudinal accelerations from 3-component accelerometer
(6) vertical accelerations from the 3-component accelerometer
(7) Roll/ LFE input signal in the CCU

Of prime interests are the roll, LFE and actual fin signals. The ship's particulars are given in Table 3.

During the passage, thirty-seven run records were taken, with a nominal run time of five minutes per run. The mean, the r.m.s., the maximum and the minimum value of each channel were the basic data derived. An example of these, based on R29 is given in Table 4. The additional channel labelled 8 is the sway acceleration, which is derived from the difference between the roll and LFE time-histories.

Particulars of each of the test run, such as the stabilisation mode, the gain settings and the ship speeds etc. are listed in table 5a and 5b. It can be seen that the ship was mostly running at speeds about 10 knots in bow/head seas. The sea states were mainly three and was in fact decreasing in strength towards the end of the passage. From these tables, the data analyses can be carried out in groups, enabling the effects of LFE stabilisation to be examined more closely.

The resulting r.m.s. values of the roll (1), fin demand (2), actual fin (3), LFE(4), vertical accelerations(6) and the sway accelerations (8) are presented in table 6a and 6 b. To give an overall picture of some of these data, the results for the roll, actual fin and LFE are plotted in fig. 9, 10 and 11 respectively. In general the maximum and minimum values are between 2 to 3 times the r.m.s. value. As over two third of these results were quite low in magnitude, hence of lower accuracy, R26 to R33 will be examined more closely in the discussion following.

The vertical acceleration data are shown in fig.12 with the LFE data. It is somewhat unexpected to find the vertical accelerations of an order lower than those of the LFE values, considering that the ship was running mainly in bow to head seas. In fig.13, the r.m.s. actual and demand fins are plotted, which shows a difference in magnitude of about 10% to 20%. Hence, the measured demand fin angles in the last section (CCU tests) would be lower by this margin. R.M.S. sway accelerations were derived from the difference between the roll time-history measurements and the LFE measurements, which took into the account of the phase effect. Another set of values, derived from the difference between the r.m.s. roll and r.m.s LFE values, were also calculated. These two sets of data are shown in fig.14. As is apparent from the graph, an under-estimation of the sway accelerations up to about 50% can be found if the phase effect is ignored. In general, the roll angle term accounts for 50 to 75% of the LFE value.

3.1 Results Discussions

For the present discussion, the results are divided into groups, which would help to focus on the more salient features particular their individual running conditions. As was mentioned earlier, two third of the results were quite low and their accuracy was not expected to be very high. Runs with r.m.s. roll angles below 1.0 degree would be considered low in the present context. The following discussion is an attempt to draw out some useful facts from the data.

3.1.1 Group 1

The first group of results comprises of R1 to R9, in which the original K_1 - K_3 roll gain settings were used both for the roll and LFE stabilisation. The overall gain K_4 used was about 50, 25 and 12% of the pre-set level of 15 installed for roll stabilisation. The low K_4 was to ensure a gradual build-up of LFE stabilisation initially, as this was the first attempt of this mode of operation. Furthermore, the wave excitation was quite low, therefore a lower than normal overall gain would produce higher motion responses, resulting in greater measurement accuracy. In so doing, it was assumed here implicitly that a corresponding increase in the overall gain would counter-act the motions induced at higher sea-states, which could be scaled up in a linear fashion in the broadest sense.

In this group, results from R2 and R3 form the best basis for comparisons as their roll values are the highest. It can be seen in Table 6a that by switching from roll to LFE, the fins as well as the motion r.m.s. values were reduced. This was also true in comparing R4 with R6. Comparing to the unstabilised case in R1, the roll and LFE responses were lower in either roll or LFE stabilised mode in R2 to R6. In R7 and R8, the overall gain was probably too high for the wave excitation for the present purpose and the resulting motion data could be ignored. R9, an unstabilised case, also indicated lower lateral motion responses. In short, based on these first few tests, LFE stabilisation seemed to have worked well with no detrimental effects and showed that some improvement over the conventional roll stabilised case could be made.

A closer look at the time-histories of some of the data will provide more details about this set of tests. In fig. 15, the fin time-histories of R2 and R3 are shown. It can be seen that the fin motions in the roll stabilised case R2 was very smooth. The fin oscillations also corresponded closely to the roll motion. However, in the LFE stabilised case R3, the fin oscillations were at much higher frequency. Despite this high fin frequency, no noticeable difference could be detected in the resulting roll motions comparing to the roll stabilised case R2. This high fin frequency response would be expected based on the results in fig. 1 - the original roll control coefficients were sensitive to high frequencies. In general, LFE signal would have a higher frequency content than the corresponding roll signal. Therefore, using the original roll controller coefficients, the fins would oscillate at higher frequency with LFE than roll.

In case of R7, R8 and even R9, more noise was present in the lateral motion time-history signals, indicating the lower quality of these data. This was also reflected in their low r.m.s. values.

3.1.2 Group 2

This group consists of R10 to R14, which was the first attempt to tune LFE for the fins. The nominal frequencies used (see Table 1 and 5a) were 0.67,0.54,0.6 and 0.64 respectively for these four runs. From the r.m.s. roll, fin and LFE data in Table 6a, it can be seen that R12 had the lowest fin and lateral motion responses. Also, comparing R12 to R6 of the last group, in which the r.m.s. roll were the same, but the r.m.s. fins and LFE were higher in R6, which was not tuned for LFE. Therefore, the optimum tuning frequency for LFE would be around the nominal frequency of 0.6.

Looking at the actual fin time-history of R10, which had the highest tuning frequency, the fins were responding to the acceleration effects of the LFE. This can be detected in the *dented* peaks in the time-history corresponding in particular to instances of high lateral motions (see fig.16). However, for some reason, this effect did not occur at the troughs of the time-history. (Perhaps the controller had some differences in gain in the positive and negative signals.) As the tuning frequency lowered, this over-sensitivity to the acceleration effects disappeared (see fig. 16). Also, from these two figures, the fins in R29 were working at a lower frequency comparing to R10. These data have also been examined in the frequency domain using FFT. However, due to the short run-time, the resolution was not fine enough for comparison. An example of this is given in fig. 18.

3.1.3 Group 3

This group is formed by R15 to R18, which are records of the normal operation of the fin stabilisers. Looking at the r.m.s data from Table 6b, the r.m.s. values consisted of some of the better results. It was unfortunate that LFE could not have been be tried at this juncture as the ship was undergoing some other tests. With an overall gain K_4 of 15, the fins quite often reached the maximum amplitude limit.

3.1.4 Group 4

R19 to R25 make up this group for the discussion. This was the second attempt to tune LFE for the fins, to see if comparable results to group 2 above could be obtained or even improved. R19 and R20 were used as the reference for comparison, which had fairly high r.m.s. motion results. Unfortunately, the course was changed after these two runs, and for the first time the ship was running in following seas in these tests. The resulting motions were reduced quite significantly. The overall gain was set at 8 to compensate for the low motion response.

The nominal frequencies used were 0.64, 0.6, 0.58 and 0.54 for R21 to R24 respectively (Table 6b). Again results with the nominal frequency of 0.6 had the lowest r.m.s. lateral motions with the lowest fin activities. The time-history signals were very similar. Although the lateral motion in R25 for the roll stabilised case were lower than those of R22, the corresponding fin motions were in fact higher. These comparisons are necessarily qualitative as their numerical values are some of the lowest in the whole test program. Nonetheless, it again suggested that the nominal frequency of 0.6 was the minimum.

3.1.5 Group 5

Having located the *desired* nominal frequency, the next step was to make comparisons between roll and LFE stabilisation by matching the resulting r.m.s fin motions as closely as possible. This is the proposed method for the main trials, i.e. set the overall gain for LFE to a point where the r.m.s. fin values of the two modes of stabilisation are within 10% difference. R26 to R33 were the results obtained in trying out this approach.

Initially, an overall gain of 20 was used in R26 to explore the effect of 'excessive' gain. It is apparent from the fin time-history that the fins saturated more frequently as a result of a higher gain but other side-effects were not encountered. In R27, the recommended overall gain was used. R.m.s. values similar to R26 were evident. The fin stauration due to high gain can be seen in fig. 17 which shows the fin motions of R26 and R27.

The overall gain was reduced to 10 in R28 to bring about higher motion responses. At this point, LFE was used with an overall gain of 10 to see if the fin motions would match up with those of R28. It was found that a gain of 10 in R29 did not produce low enough fin motions for comparisons. Therefore the overall gain was increased to 13, but still the resulting r.m.s. fin motions was not close enough to the R28 case. In order to achieve the 10% margin, the overall gain for roll was reduced instead. In R31, the overall gain for roll stabilisation was decreased to 7. The resulting fin motions were now comparable to those of R29 and R30.

Further adjustments were made in R32 and R33. However, these two results should be left out from this discussion at this point. This is justified on the ground that by reducing the overall gain, the lateral motions should increase. But in these two cases, they both showed a reduction in the lateral motion with lower overall gain, or keeping the same lateral motion levels with reduction in fin activities. This might be brought about by a change in the sea conditions locally (compare R31 to R32).

Based on the results from R28 to R31 alone, it is difficult to say how well LFE stabilisation compare with the conventional roll stabilisation. If R29 was compared with R31, the conventional roll case seemed to fare better. However, if R30 was compared with R31, some slight advantage could be gained from LFE. These are based on the criterion of similar fin activities from the two modes. A higher sea-state would probably produce more of a difference between the two modes of operation. However, could it be better to make a comparison based on the r.m.s. roll motions as a matching criterion? From R30 and R31, the r.m.s. roll motions were the same. These were achieved with similar fin motions with a difference of about 5%, but resulting in about 9% improvement in LFE with LFE stabilisation. On the other hand, the r.m.s. roll motions in R29 (LFE stabilised) was 20% higher than those of R31 (Roll stabilised) and the corresponding LFE is greater by 13%, but the fin motions differed by only 3%.

3.1.6 Group 6

This group consists of R34 to R37. The major change in the running condition was the ship speed, which was increased to 23 kts (see Table 6b). The fin motions were reduced substantially as they are more effective at high speed. It is interesting to note that in R37, the overall gain was reduced in the hope to obtain high motion responses. However, as can be seen from a comparison with the results from R36, the resulting lateral motion had in fact reduced with the lower gain. Also , at this stage, the sea was decreasing in strength rapidly and therefore no further measurements were carried out.

3.2 Comments and Observations

(1) Upon close inspections of the LFE and roll time-histories from the measurements, it was found that roll stabilisation was more likely to introduce *disruptive* motion than LFE stabilisation. This can be explained as follows. Firstly, the LFE and roll time-histories were normalised by their r.m.s. values. From a plot of these two set of data, it can be seen that the roll and LFE results would correspond closely, i.e. the maximums and minimums are of more or less the same magnitude and they decrease or increase in direct proportion. This is the case for the unstabilised case, for example in fig. 19 for R19. The solid line is LFE and the dotted line is roll.

When the LFE magnitude is above a threshold value, which is likely to be at or above the peak values, an interruption to a person's postural stability would take place. This has brought about the conception of Motion Induced Interruption MII. When roll stabilisation is used, the overall motion will be reduced and MII is less likely to occur. However, when the motion is high enough, MII will also take place despite being stabilised for roll.

Looking at the roll and LFE time-histories of R18, under roll stabilisation in fig. 19. It can be seen that a few of the maximum/ minimum values of the two signals are not of the same order, i.e. roll motion is reduced by an amount significantly more or significantly less than the corresponding LFE. Hence, with roll stabilisation, LFE and roll are more likely to affected in an out-of-proportion manner. This may explain why some of some mariners, especially when roll stabilisers were first introduced, found it difficult to balance themselves when the ship was roll stabilised - the perceived angular motion (roll) could be significantly different from the acceleration (LFE) experienced. One of the conclusions in Warhurst et al 1969 : " lateral accelerations caused by roll-reducing devices may be more harmful to human performance than some greater amount of roll " could also find support in the way the above time-history data were examined. The disparity in the peak values of the two signals can again be found in fig. 20 for R28, another roll stabilised case.

Now turning to the LFE stabilised case in fig. 20, the correspondence at the peak values are *better*. This suggests that better postural stability performance could be expected from LFE stabilisation. Therefore in the main trials, these two-histories together presented in such a way should be looked at in relation to the video recording of the human factor experiments. The correlation between roll, LFE and postural stability due to this effect could then be established.

Based on the above observation, it is proposed here that in the unstabilised case, postural stability is lost mainly due to motions exceeding some threshold values, whilst under roll stabilisation, apart from this threshold exceedance effect, postural stability can also be lost due to the difference between the perceived angular motion and the acceleration actually experienced by a person. This could be an important factor in assessing the successful use of LFE stabilisation. Judging the performance of the two modes of stabilisation purely on r.m.s. terms may not show this difference in the disparity in motion reduction at specific time instances.

(2) Immediately after R26, a LFE stabilised mode was planned. When LFE mode was on, vibration was detected from the fin operation and the fins were switched off after a few cycles. It was found that the controller gains had not been changed from the roll stabilised mode and that the overall gain was 20. It appeared that this combination of controller coefficients with a high overall gain had probably put the system near the unstable region of the control system when LFE was used.

4. Conclusions

(1) All the equipment designed and built for the LFE trials have proven to be of a high standard. They have all performed well in the tuning trials and are now totally adjusted for the trial ship, ready for the main trials.

(2) Based on the experience from the tuning trials, a procedure for the LFE trials, specifically for the operation of the controller and the auxcillary equipment, has been proposed for the main trials. This procedure provides a safe and efficient routine for the operator.

(3) Based on the limited tests with the tuning procedure, it appears that there is an optimum point for LFE stabilisation. Some encouraging results were obtained when LFE stabilisation operated at this optium point. However, the criteria, on which the performance comparisons between the two mode of operations are made, may not be adequate.

(4) At this stage, there is not enough evidence to show that LFE stabilisation is *better* than roll stabilisation. However, at the same time, LFE stabilisation has not produced any undesirable effects if probably tuned for the ship. There is limited evidence to show that roll stabilisation could induce disruptive motions whilst LFE stabilisation appears to cause less disruptive motions.

References

(1) BRF6510(217) Central Control Unit (stabilisers)
NSN 2030-99-799-3637 MoD.

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DRA /TM(AWMH) 92336

Marshfield B. 1980
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Ship Science Contract Report to DRA Haslar

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Naval Ship Research and Development Centre Report 2828.

Table 1.
An example of K values over a range of frequencies for the trial

Nominal Frequency	K_1	K_2	K_3	K_4
Brown Brothers	25	80	84	15
(A) 0.7	0	80	28	-
(B) 0.67	6	80	25	-
(C) 0.64	9	80	23	-
(D) 0.62	12	80	20	-
(E) 0.60	14	80	17	-
(F) 0.58	19	80	12	-
(G) 0.56	25	80	12	-
(H) 0.54	34	80	9	-
(I) 0.52	51	80	6	-

K_4 for (A)-(I) are to be decided in the sea-trials
One third of the Brown Brothers' value is a good starting point

Table 2.
Accelerometers Locations

	X (m) (aft of midships)	Y (m) (from centreline)	Z (m) (above USK)
3-component accelerometer	21.44	0.35	10.30
LFE accelerometer	21.00	0.35	10.30

Table 3.
Ship Particulars

Length (m)	122.9
Beam (m)	14.91
Draft (m)	4.67
Displacement (ton)	4248.0
Trim (m)	0.60
VCG above USK (m)	6.05
LCG aft of midships (m)	1.90
Keel to VCB (m)	2.85
Keel to metacentre (m)	7.61
GM (fluid) (m)	1.35

Table 4.
An Example of the R29 Data

Channel	Parameter	Mean	R.M.S.	Max.	Min.
1	Roll (°)	-2.42	2.63	3.92	-9.24
2	Fin Demand (°)	-.14	6.95	18.23	-15.31
3	Fin Actual (°)	.01	7.77	17.19	-20.86
4	LFE (m/s^2)	-.13	.52	1.17	-1.43
5	Long[l]. acc. (m/s^2)	-.18	.04	-.06	-.25
6	Vert. acc. (m/s^2)	.05	.20	.59	-.53
7	CCU input ('v')	.00	.14	.35	-.33
8	'Sway' acc. (m/s^2)	.00	.18	.51	-.49

Channel denotes the A to D system channels used for acquiring except for channel 8 which is to denote the calculations based on channel 1 and 4.

Table 5a.
A List of Trial runs (7/12/92)

Ref.	TIME	K_1	K_2	K_3	K_4	Mode	Kts	θ μ	S.S.	comments
R1	20:35	-	-	-	-	U	10.7	315	3	auto-off
R2	20:50	25	80	84	4	Roll	10.6	315	3	"
R3	20:58	25	80	84	4	LFE	11.9	320	3	"
R4	21:17	25	80	84	2	Roll	11.6	320	3	"
R5	21:27	25	80	84	2	LFE	11.5	320	3	Aborted
R6	21:34	25	80	84	2	LFE	11.0	340	3	auto-off
R7	21:47	25	80	84	8	Roll	12.1	340	3	"
R8	21:55	25	80	84	8	LFE	12.1	340	3	"
R9	22:05	-	-	-	-	U	12.0	340	3	"
R10	22:15	6	80	25	4	LFE	11.9	340	3	"
R11	22:24	34	80	9	4	LFE	11.6	340	3	"
R12	22:31	16	80	17	4	LFE	11.6	340	3	"
R13	22:44	9	80	23	4	LFE	11.7	340	3	"

NOTE:

θ - wave angles, 0°: bow sea, 90°: starboard beam sea, 180°: following sea; 180°-360°: port side.

S.S - sea states

Table 5b.
A List of Trial Runs (8/12/92)

Ref.	TIME	K_1	K_2	K_3	K_4	Mode	Kts	θ	S.S.	comments
R14	09:21	25	80	84	15	Roll	19.0	-	3	aborted
R15	09:55	25	80	84	15	Roll	19.0	-	3	auto-on
R16	10:10	25	80	84	15	Roll	10.0	-	3	"
R17	10:45	25	80	84	15	Roll	10.0	-	3	"
R18	11:43	25	80	84	15	Roll	10.0	-	3	"
R19	11:50	-	-	-	-	U	10.0	180	3	"
R20	11:57	25	80	84	8	Roll	10.0	340	3	"
R21	12:20	9	80	23	8	LFE	10.7	230	3	auto-off
R22	12:27	16	80	17	8	LFE	10.7	230	3	"
R23	12:34	19	80	15	8	LFE	10.6	230	3	"
R24	12:40	34	80	9	8	LFE	10.2	230	3	"
R25	13:48	25	80	84	8	Roll	-	-	-	"
R26	14:50	25	80	84	20	Roll	9.3	340	3	"
R27	15:00	25	80	84	15	Roll	9.4	340	3	"
R28	15:10	25	80	84	10	Roll	9.4	340	3	"
R29	15:20	16	80	17	10	LFE	9.7	350	3	"
R30	15:25	16	80	17	13	LFE	9.7	340	3	"
R31	15:36	25	80	84	7	Roll	9.9	350	3	"
R32	15:43	25	80	84	4	Roll	9.6	350	3	"
R33	15:49	16	80	17	7	LFE	9.8	350	3	"
R34	16:11	25	80	84	4	Roll	9.8	350	3	"
R35	16:17	-	-	-	-	U	23.0	0	3	"
R36	16:24	16	80	17	7	LFE	23.0	0	3	"
R37	16:30	16	80	17	4	LFE	23.0	0	3	"

Table 6a.
R.M.S. Values of Selected Channels

	Roll 1 deg	fin demand 2 degs	fin actual 3 degs	LFE 4 ms^{-2}	vertical 6 ms^{-2}	longitudinal 8 ms^{-2}	Mode of stabilisation
R1	1.51	-	-	0.38	0.22	0.16	U
R2	1.24	2.81	3.14	0.35	0.26	0.17	Roll
R3	0.82	2.02	2.41	0.24	0.22	0.12	LFE
R4	1.29	2.04	1.53	0.32	0.23	0.14	Roll
R5	-	-	-	-	-	-	LFE
R6	1.16	1.03	1.26	0.27	0.22	0.1	LFE
R7	0.52	2.28	2.57	0.14	0.20	0.08	Roll
R8	0.59	2.96	3.79	0.15	0.20	0.08	LFE
R9	0.75	-	-	0.19	0.21	0.09	U
R10	0.99	1.25	1.43	0.25	0.18	0.10	LFE
R11	1.16	1.99	2.21	0.26	0.25	0.09	LFE
R12	0.81	1.09	1.25	0.19	0.22	0.08	LFE
R13	1.06	1.30	1.48	0.25	0.25	0.09	LFE

Table 6b.
R.M.S. Values of Selected Channels

	1	2	3	4	6	8	Mode
R14	1.69	4.91	5.51	0.39	0.30	0.17	Roll
R15	1.57	10.59	11.89	0.37	0.29	0.19	Roll
R16	2.17	14.13	15.58	0.46	0.28	0.22	Roll
R17	1.88	12.84	14.34	0.37	0.20	0.17	Roll
R18	1.06	8.17	9.18	0.27	0.26	0.12	Roll
R19	1.79	-	-	0.39	0.26	0.14	U
R20	1.78	7.28	8.16	0.45	0.31	0.24	Roll
R21	0.94	1.94	2.27	0.19	0.13	0.08	LFE
R22	0.72	1.79	2.09	0.15	0.11	0.08	LFE
R23	0.81	2.31	2.68	0.18	0.12	0.09	LFE
R24	0.74	2.55	2.91	0.16	0.11	0.08	LFE
R25	0.63	2.43	2.74	0.14	0.10	0.07	Roll
R26	2.01	15.80	17.67	0.42	0.25	0.22	Roll
R27	2.09	14.02	15.52	0.41	0.23	0.20	Roll
R28	2.14	10.19	11.34	0.44	0.26	0.22	Roll
R29	2.63	6.95	7.77	0.52	0.20	0.18	LFE
R30	2.15	7.47	8.38	0.42	0.21	0.17	LFE
R31	2.15	7.22	7.98	0.46	0.19	0.17	Roll
R32	2.13	4.06	4.51	0.44	0.18	0.17	Roll
R33	2.46	4.58	5.10	0.49	0.15	0.16	LFE
R34	0.76	3.51	3.82	0.17	0.07	0.09	Roll
R35	1.59	-	-	0.33	0.06	0.09	U
R36	1.10	1.76	1.94	0.21	0.06	0.08	LFE
R37	0.70	0.57	0.65	0.13	0.04	0.05	LFE

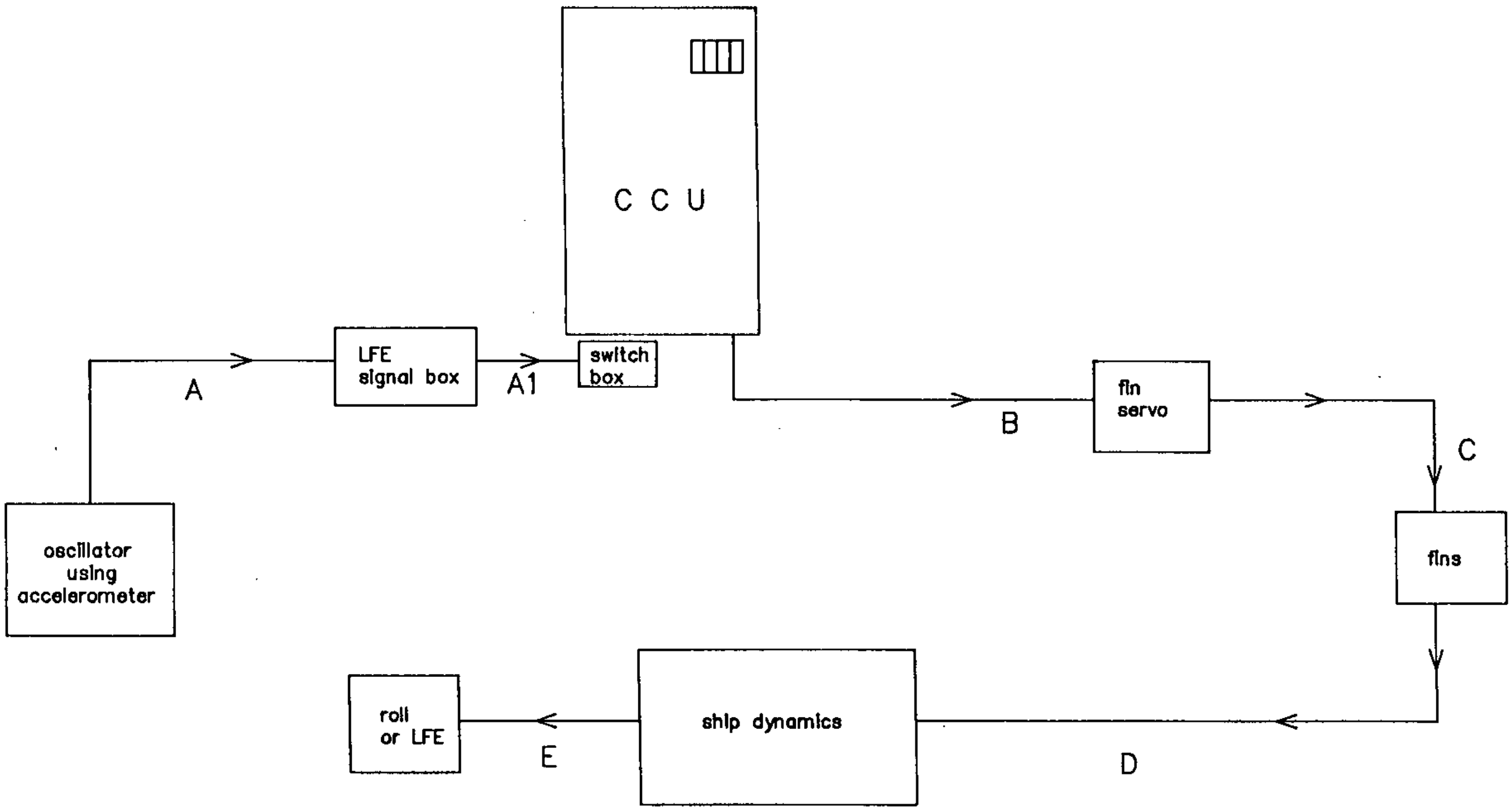


Diagram 1. Signal path for the CCU tests

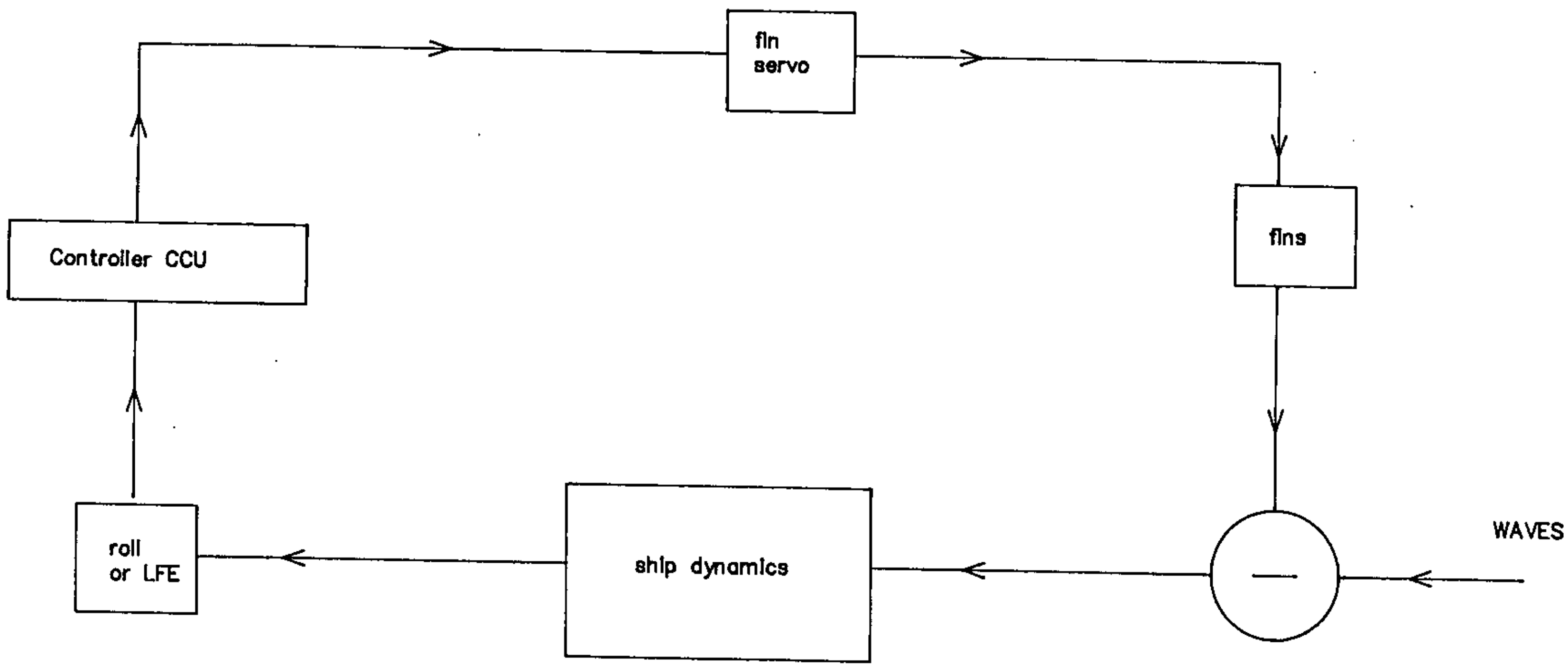


Diagram 2 Block Diagram for Roll or LFE fin stabilisation

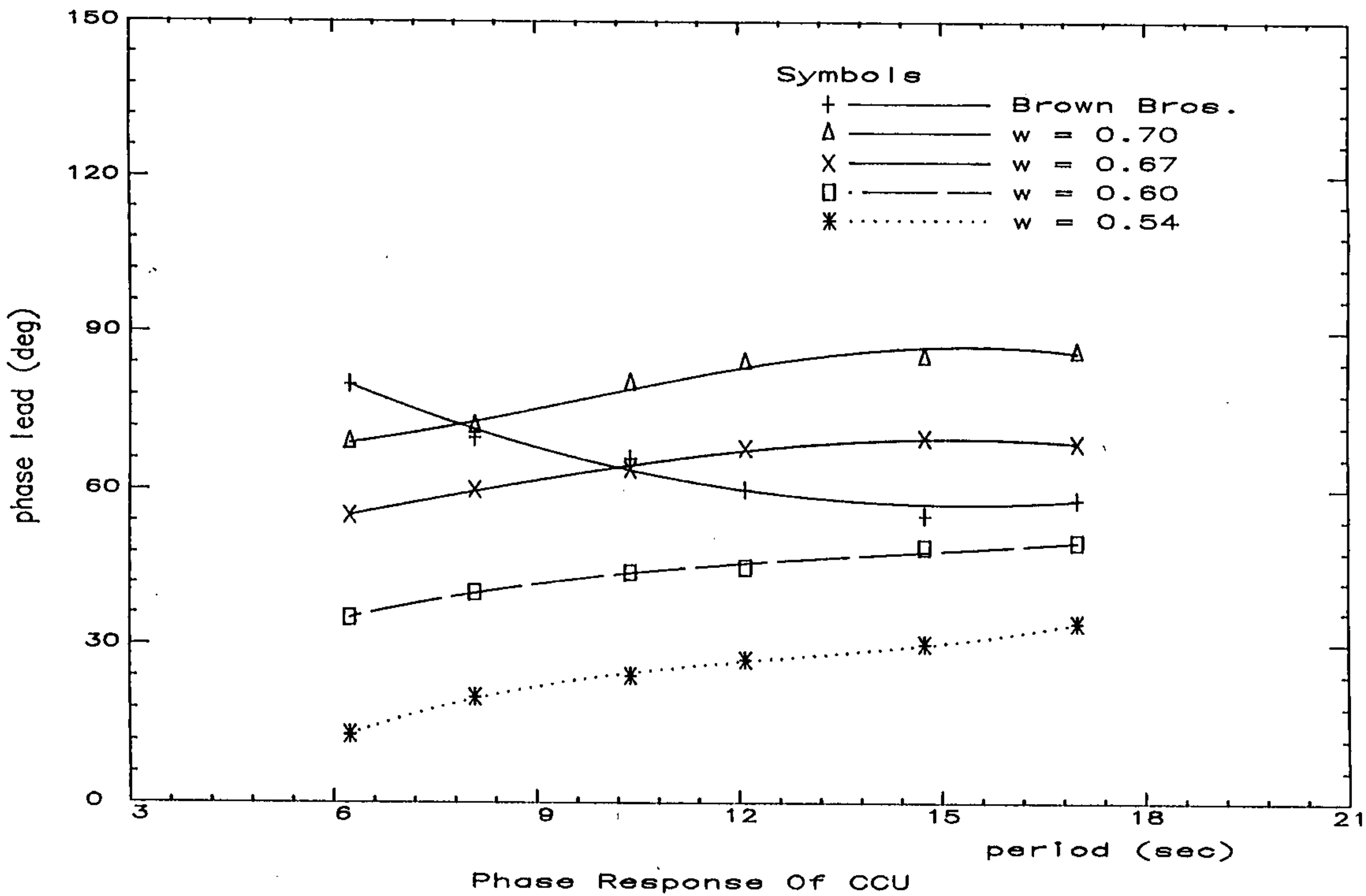

Symbols
Brown Bros.
w = 0.70
w = 0.67
w = 0.60
w = 0.54
150
120
90
60
30
0
3
6
9
12
15
18
21
phase lead (deg)
period (sec)
Phase Response Of CCU

FIG. 2

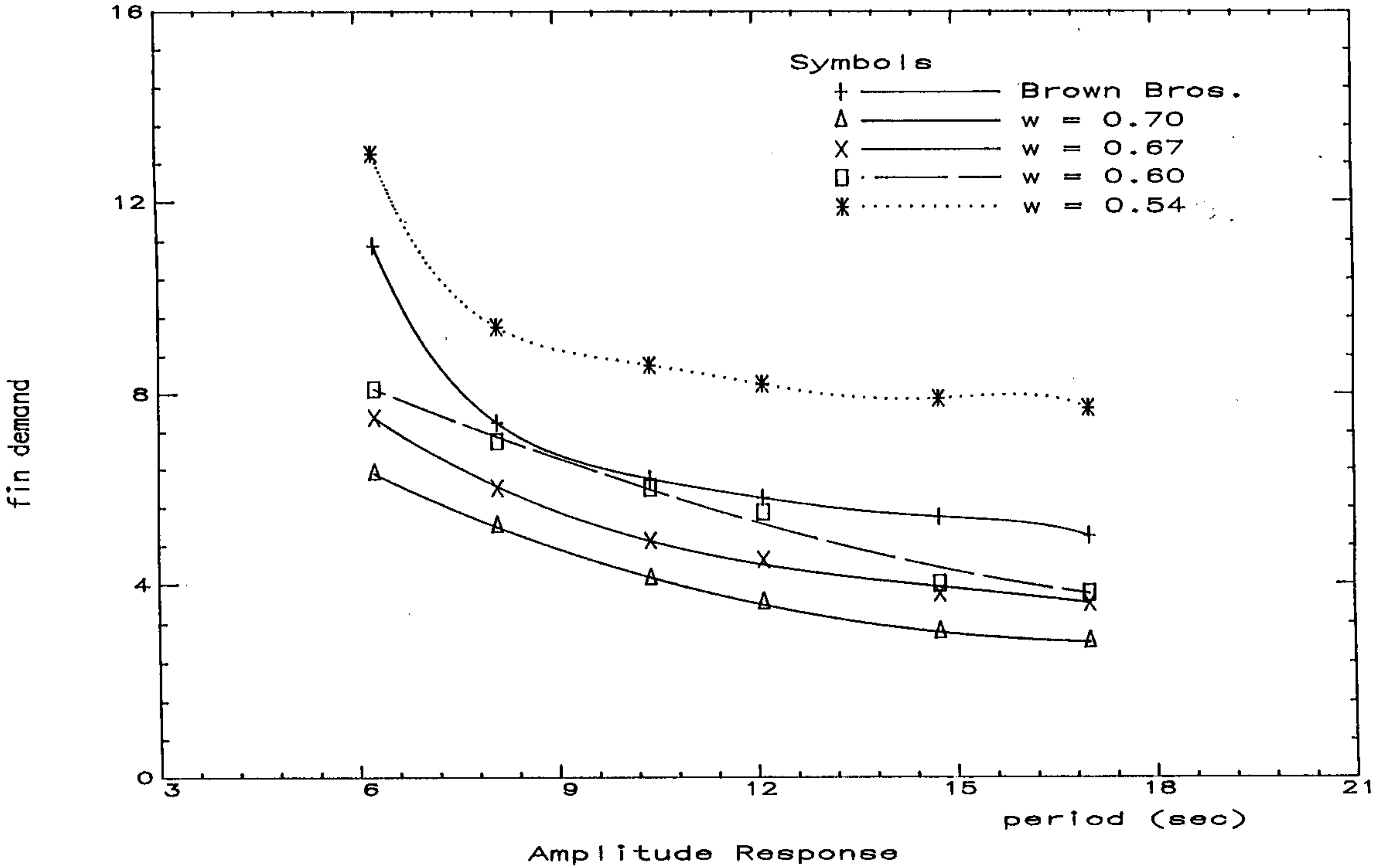
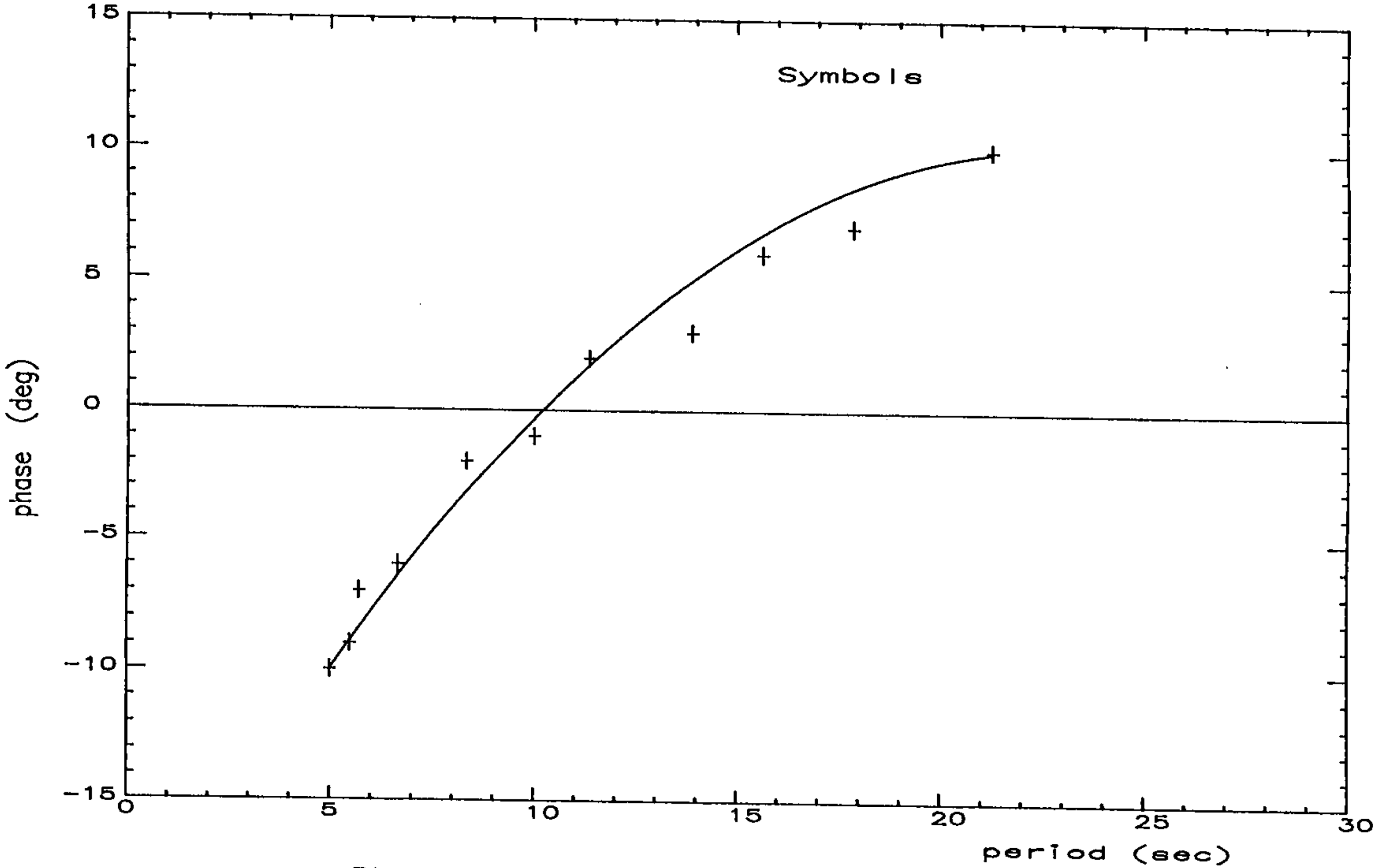

Symbols
Brown Bros.
w = 0.70
w = 0.67
w = 0.60
w = 0.54
fin demand
16
12
8
4
0
3
6
9
12
15
18
21
period (sec)
Amplitude Response

FIG. 1



Phase response of the LFE filter

FIG. 3

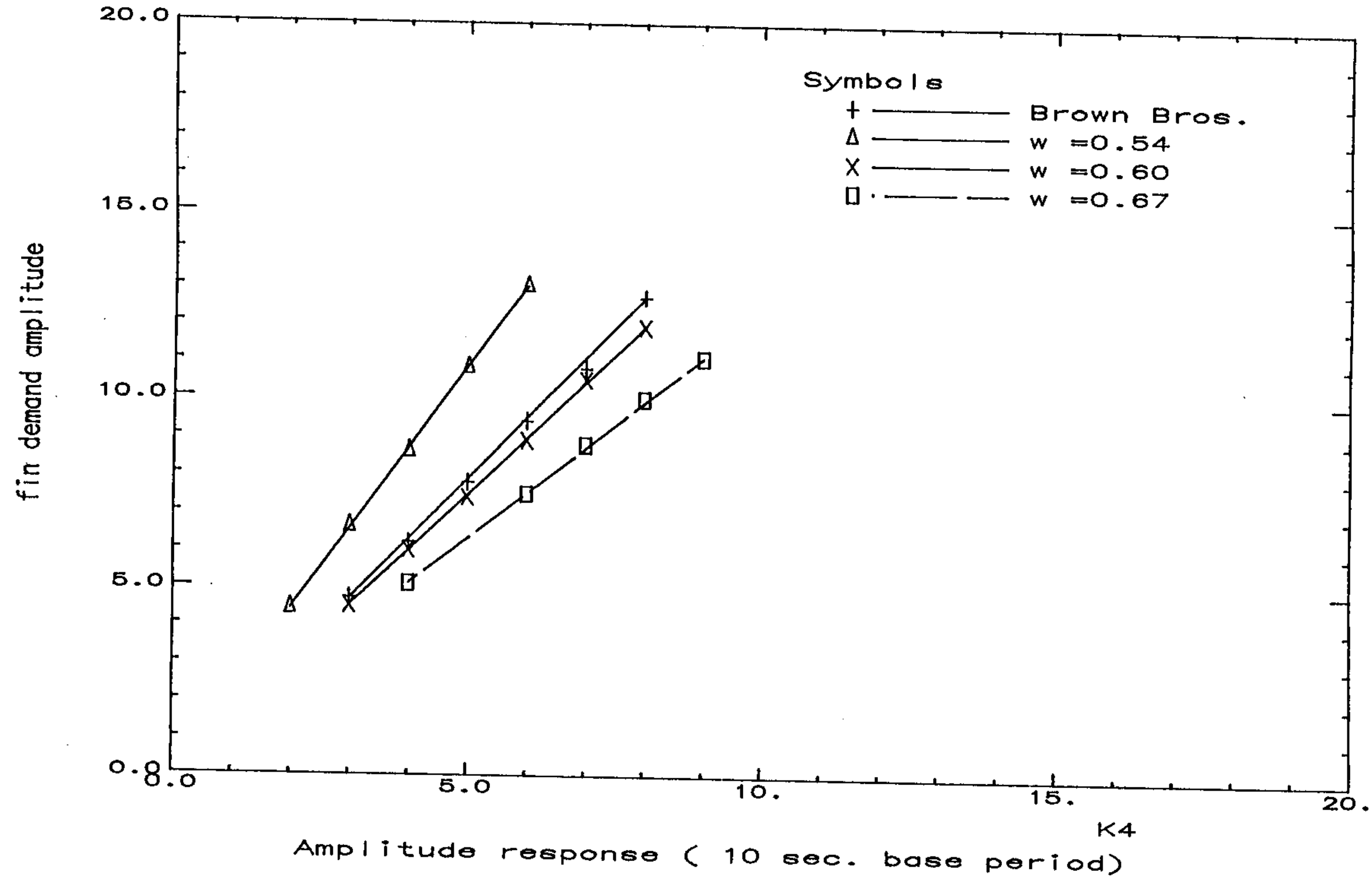

Symbols
+ Brown Bros.
Δ w =0.54
X w =0.60
□ w =0.67
20.0
15.0
10.0
5.0
0.0
0.0
5.0
10.
15.
20.
K4
fin demand amplitude
Amplitude response (10 sec. base period)

FIG. 4

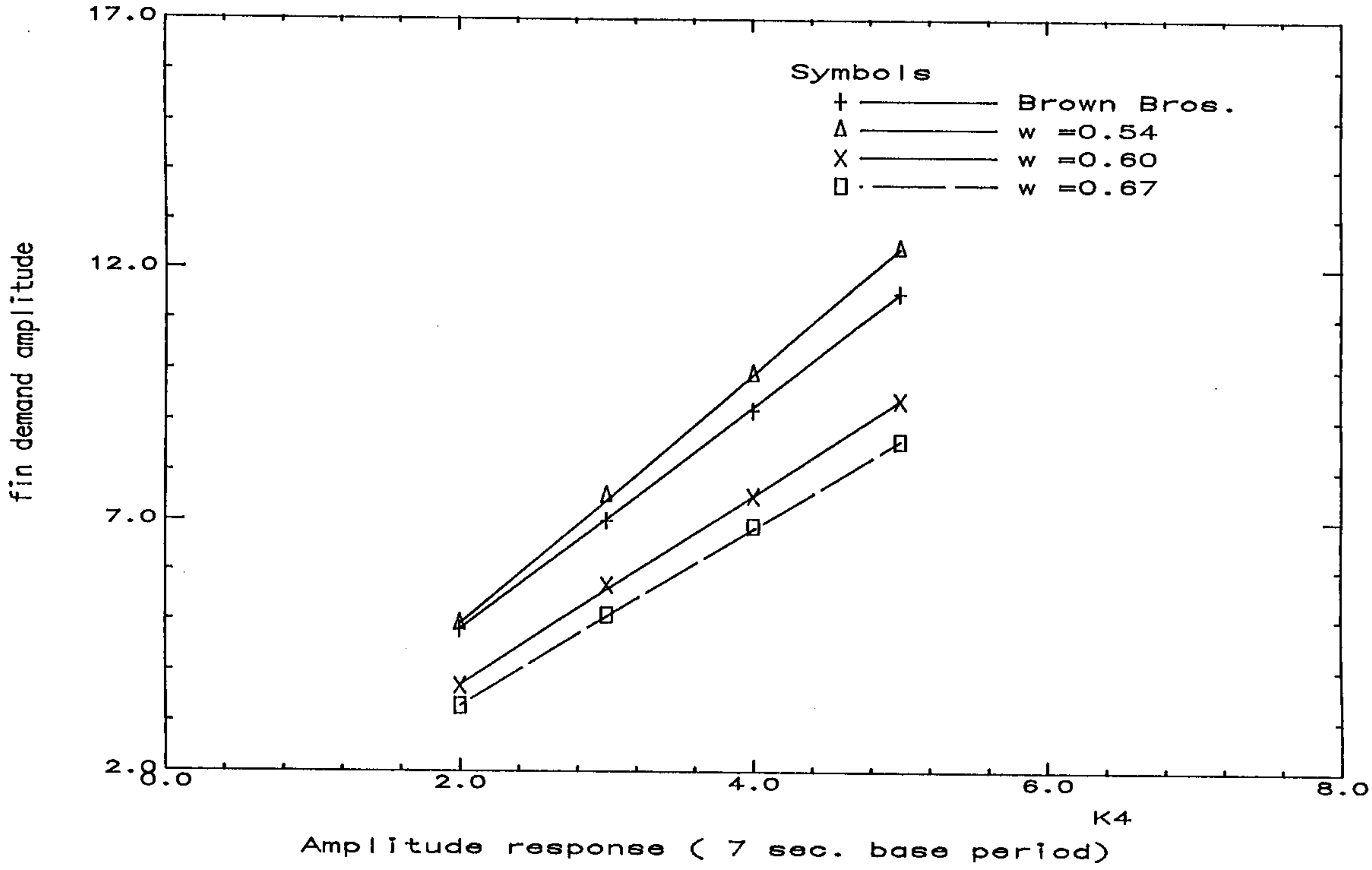

Symbols
Brown Bros.
w =0.54
w =0.60
w =0.67
17.0
12.0
7.0
2.8
0.0
2.0
4.0
6.0
8.0
K4
fin demand amplitude
Amplitude response (7 sec. base period)

FIG. 5

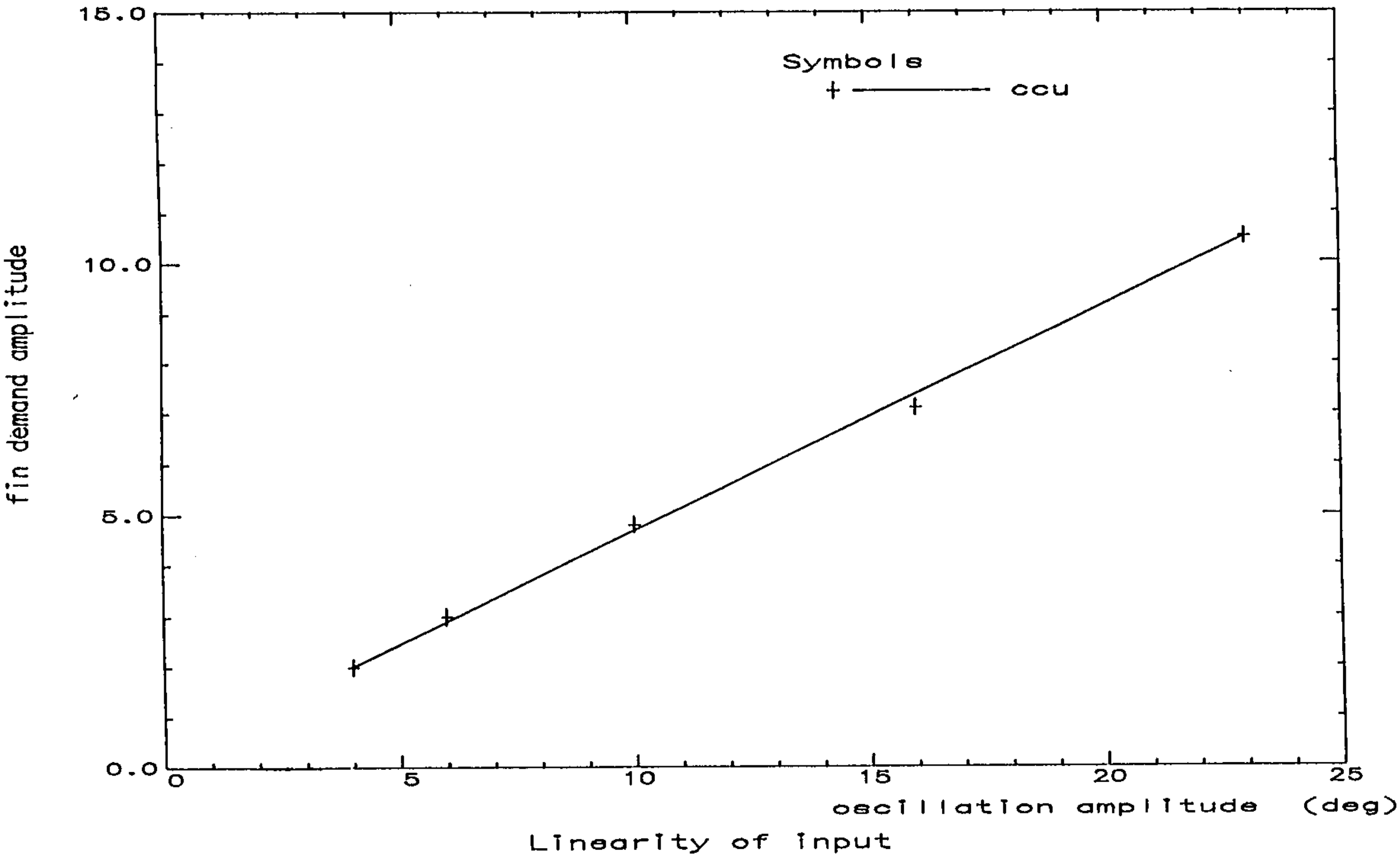
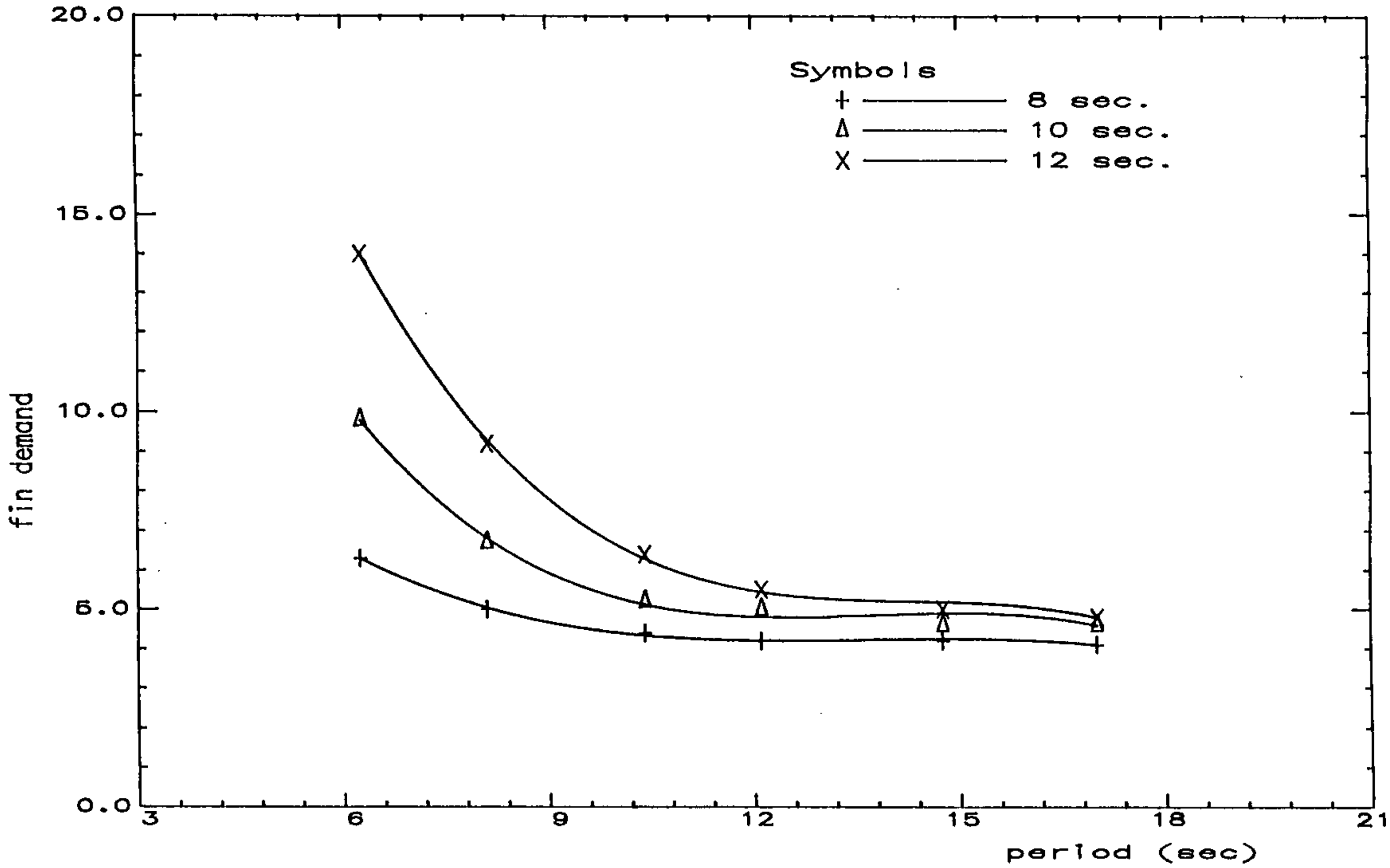

15.0
10.0
5.0
0.0
0
5
10
15
20
25
Symbols
ccu
fin demand amplitude
oscillation amplitude (deg)
Linearity of input

FIG. 6



Amplitude Response (HMS SULTAN)

FIG. 7

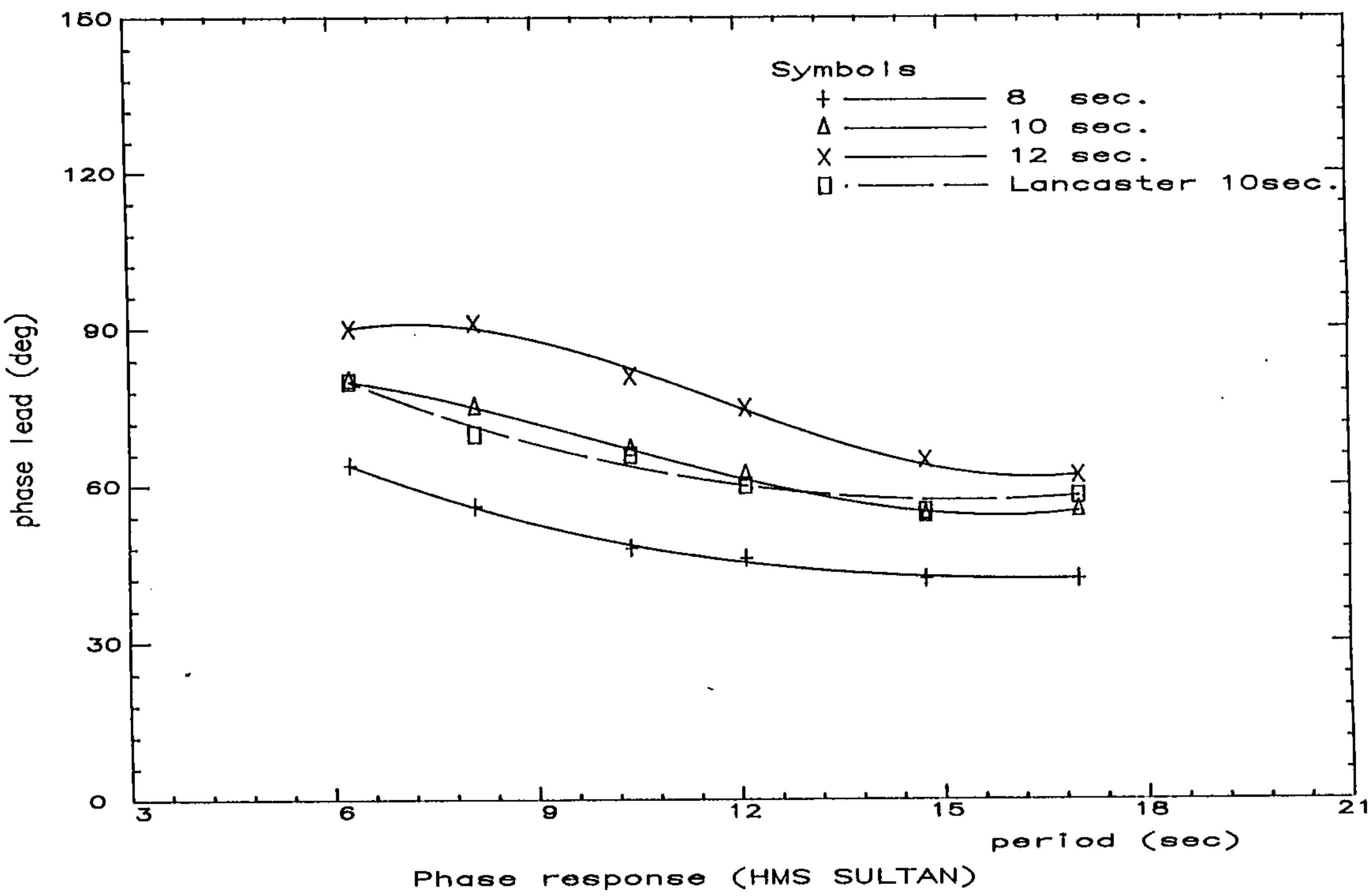
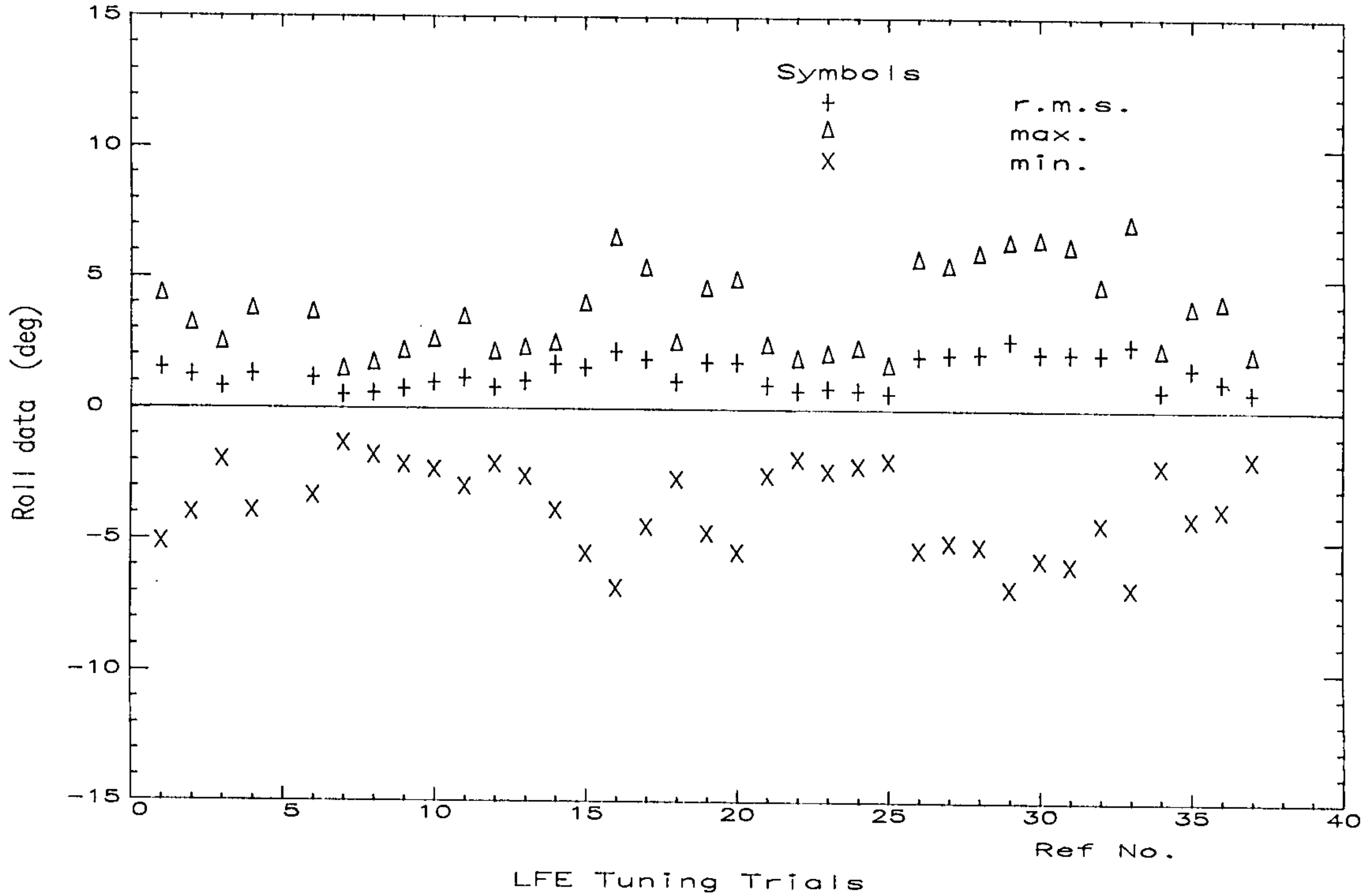

Symbols
+ 8 sec.
Δ 10 sec.
X 12 sec.
□ Lancaster 10sec.
phase lead (deg)
0
30
60
90
120
150
3
6
9
12
15
18
21
period (sec)
Phase response (HMS SULTAN)

FIG. 8



Symbols
+ r.m.s.
Δ max.
X min.
Roll data (deg)
15
10
5
0
-5
-10
-15
0
5
10
15
20
25
30
35
40
Ref No.
LFE Tuning Trials

FIG. 9

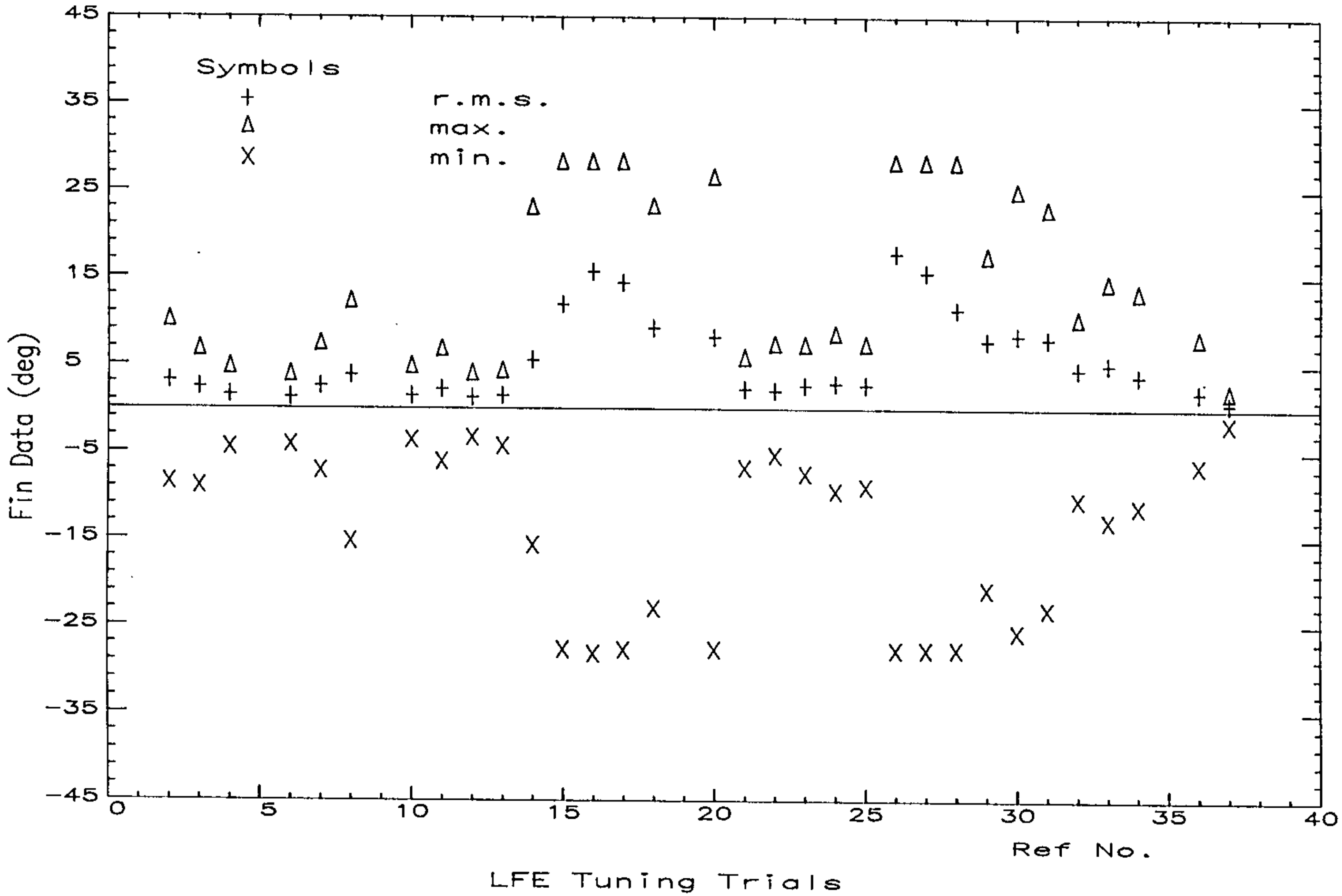

Symbols
+ r.m.s.
Δ max.
X min.
Fin Data (deg)
45
35
25
15
5
-5
-15
-25
-35
-45
0
5
10
15
20
25
30
35
40
Ref No.
LFE Tuning Trials

FIG. 10

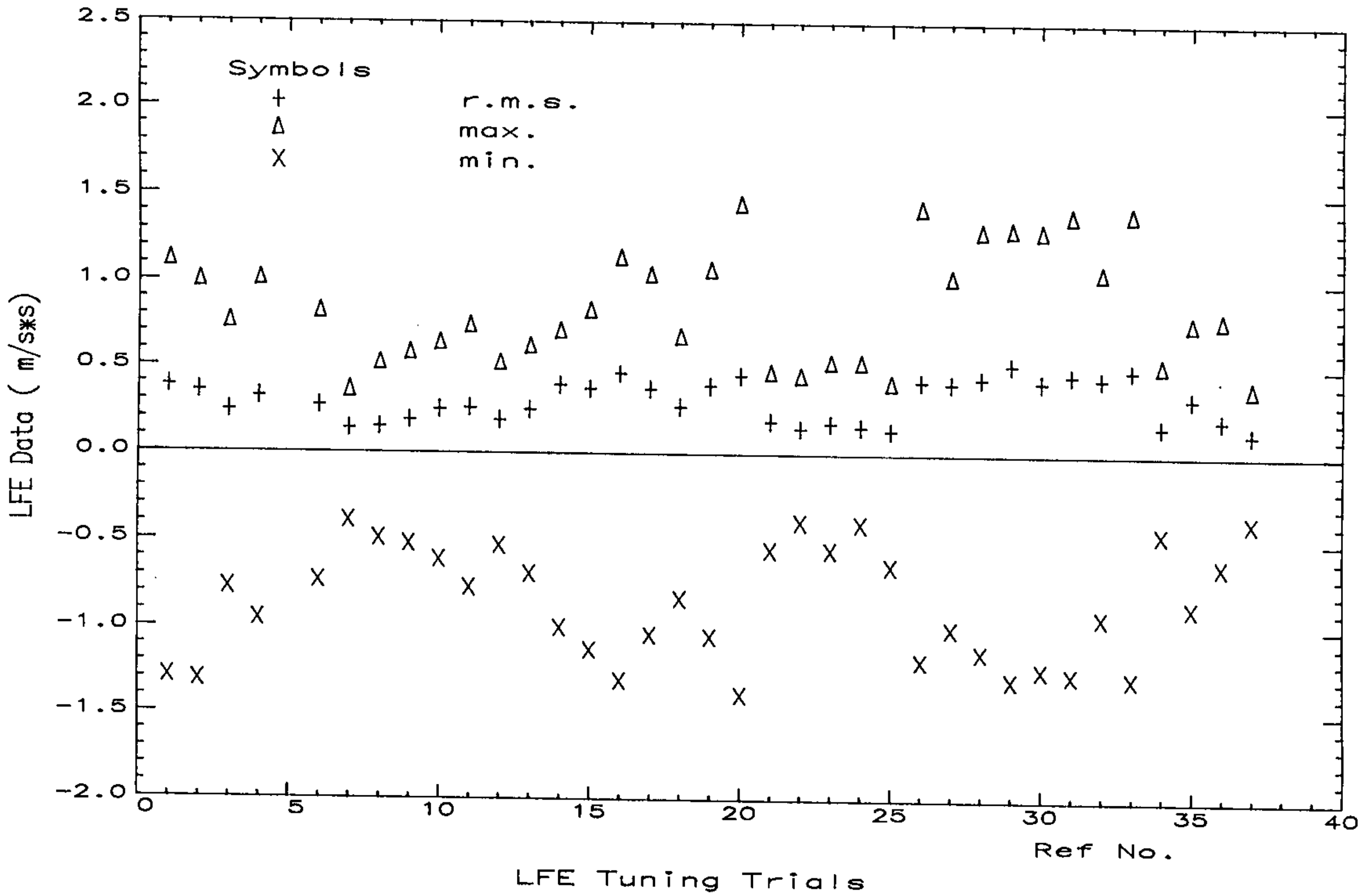
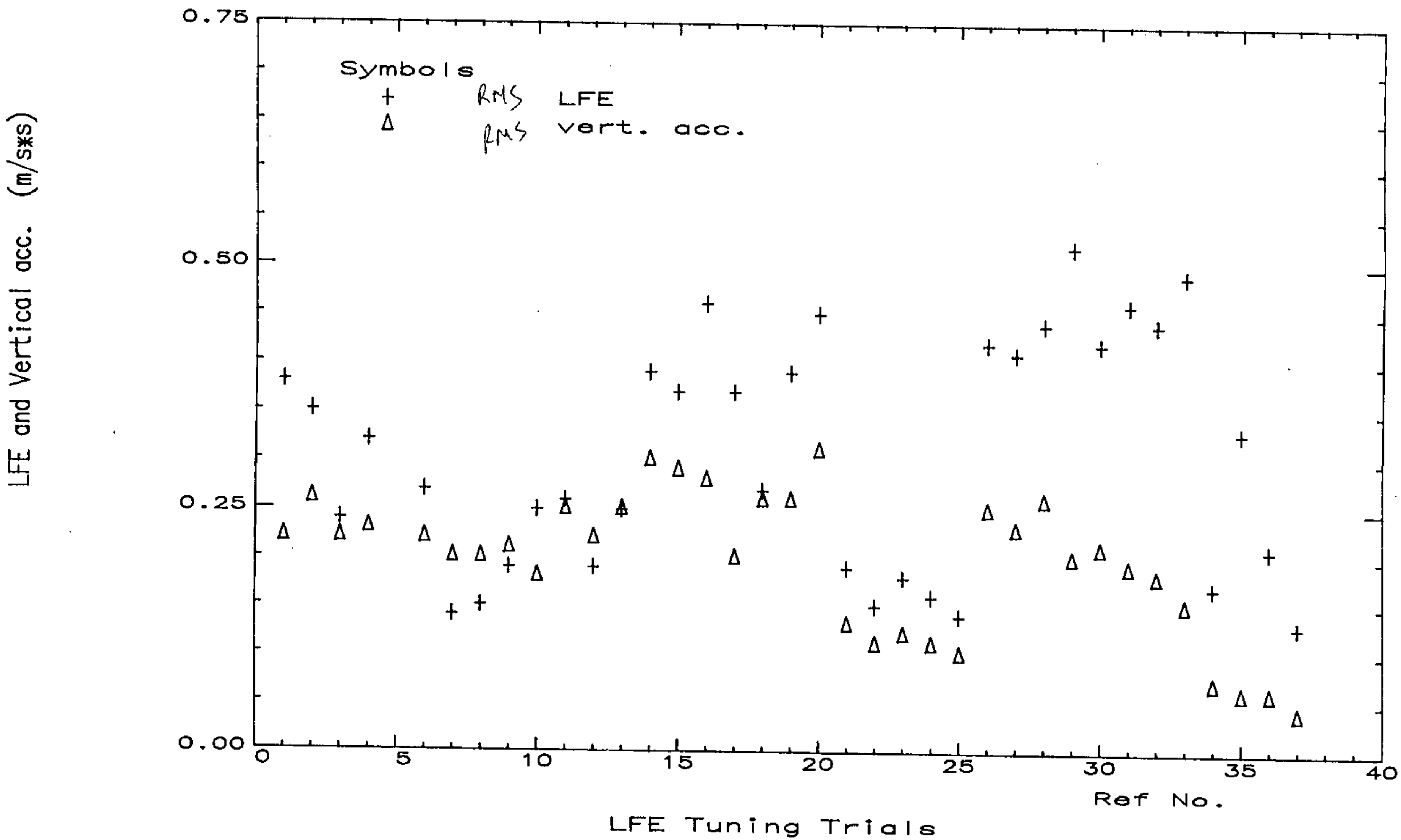

Symbols
+ r.m.s.
Δ max.
X min.
LFE Data (m/s*s)
Ref No.
LFE Tuning Trials
2.5
2.0
1.5
1.0
0.5
0.0
-0.5
-1.0
-1.5
-2.0
0
5
10
15
20
25
30
35
40

FIG. 11



Symbols
+ RMS LFE
Δ RMS vert. acc.
LFE and Vertical acc. (m/s*s)
0.75
0.50
0.25
0.00
0
5
10
15
20
25
30
35
40
Ref No.
LFE Tuning Trials

FIG. 12

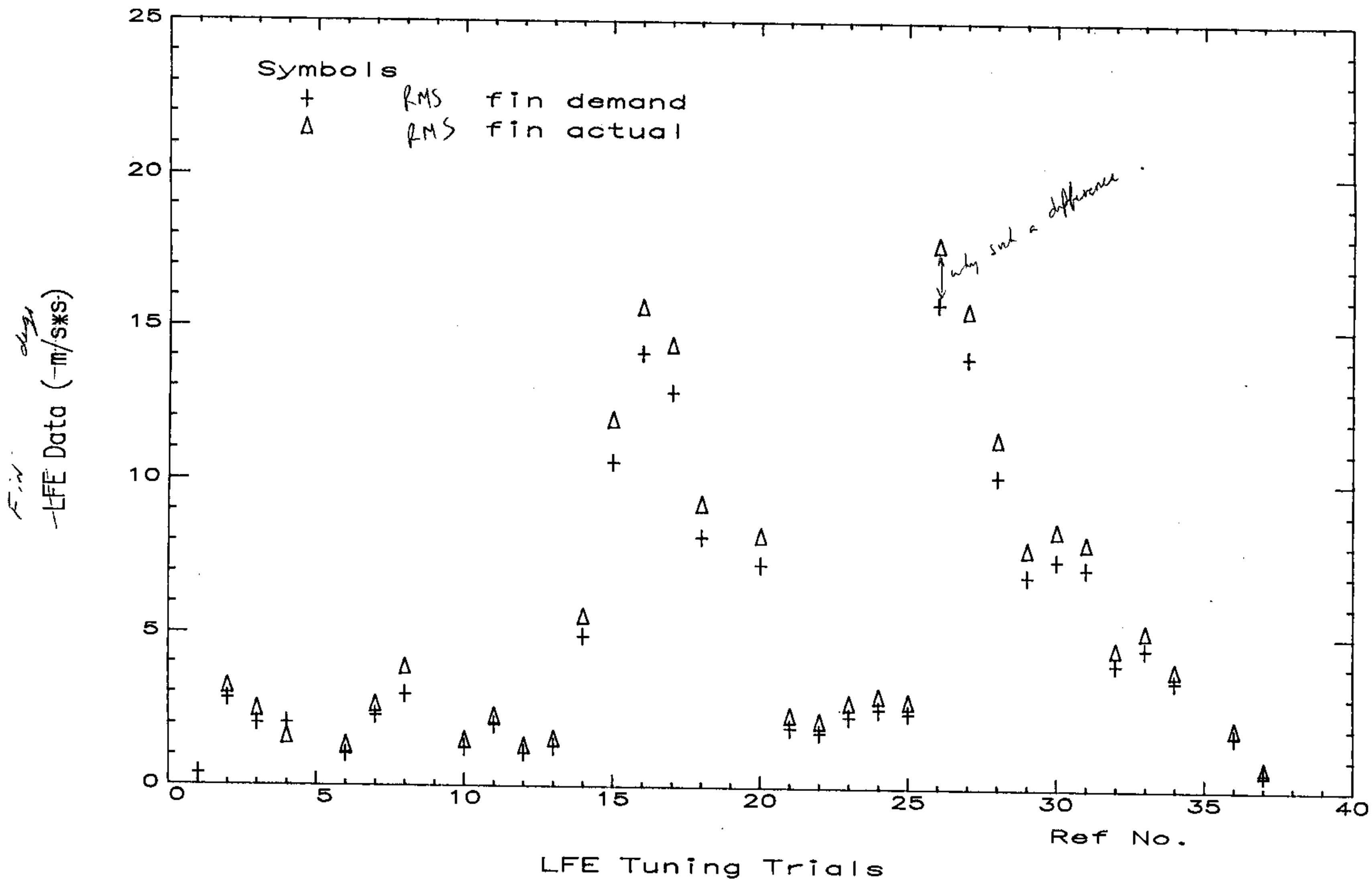
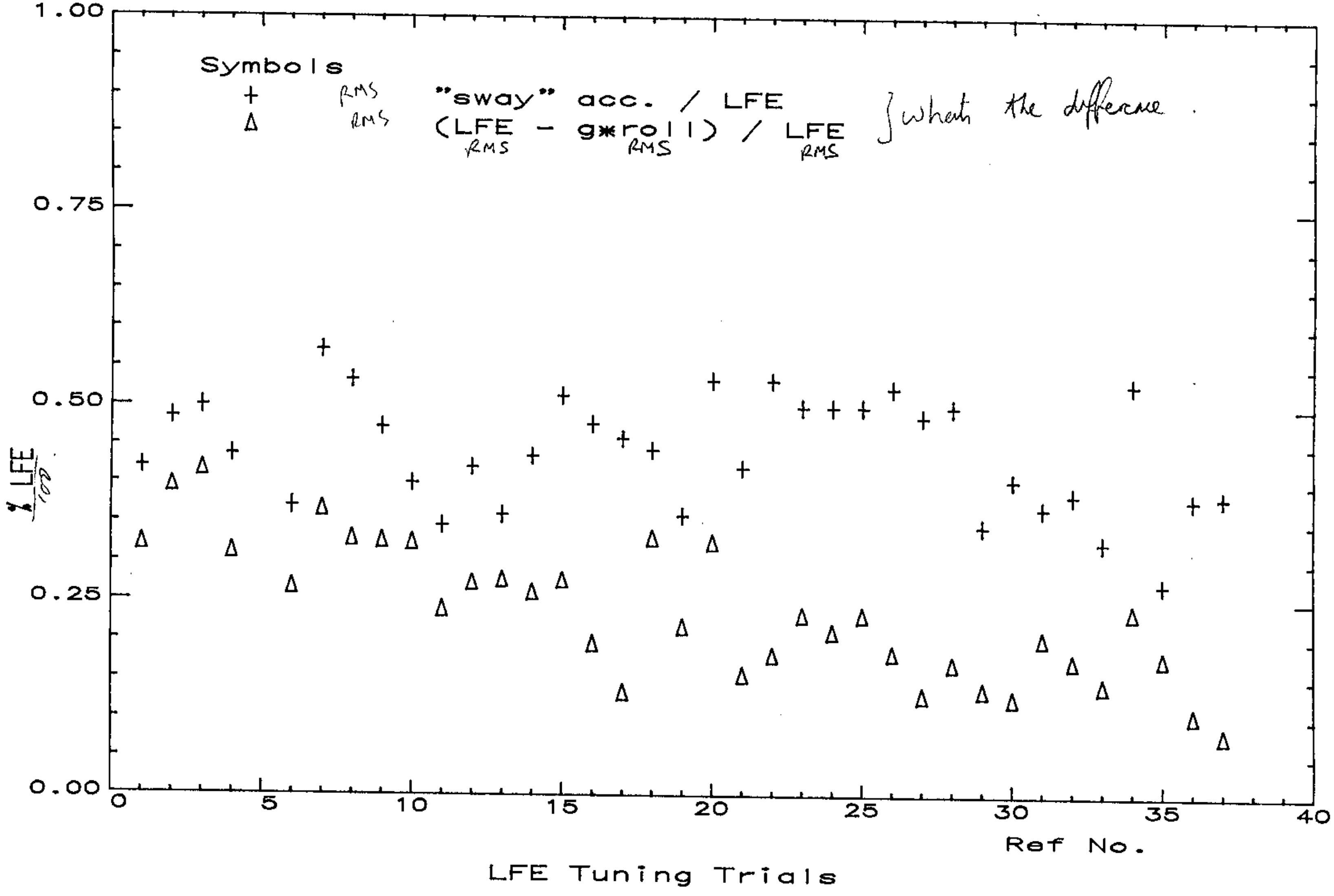
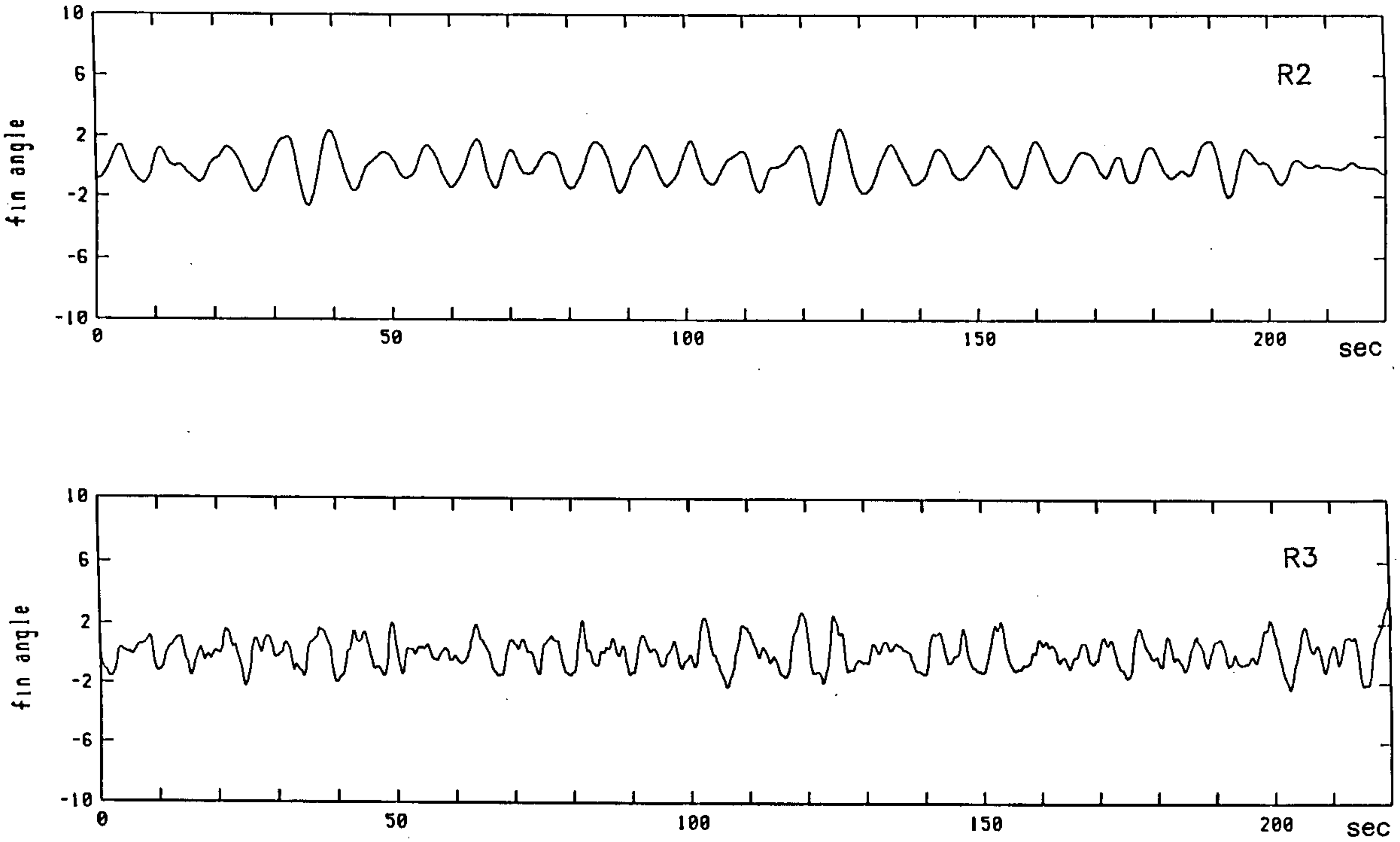

Symbols
+ RMS fin demand
Δ RMS fin actual
why such a difference
Fin degs
~~LFE~~ Data (~~m/s*s~~)
0
5
10
15
20
25
30
35
40
Ref No.
LFE Tuning Trials

FIG. 13

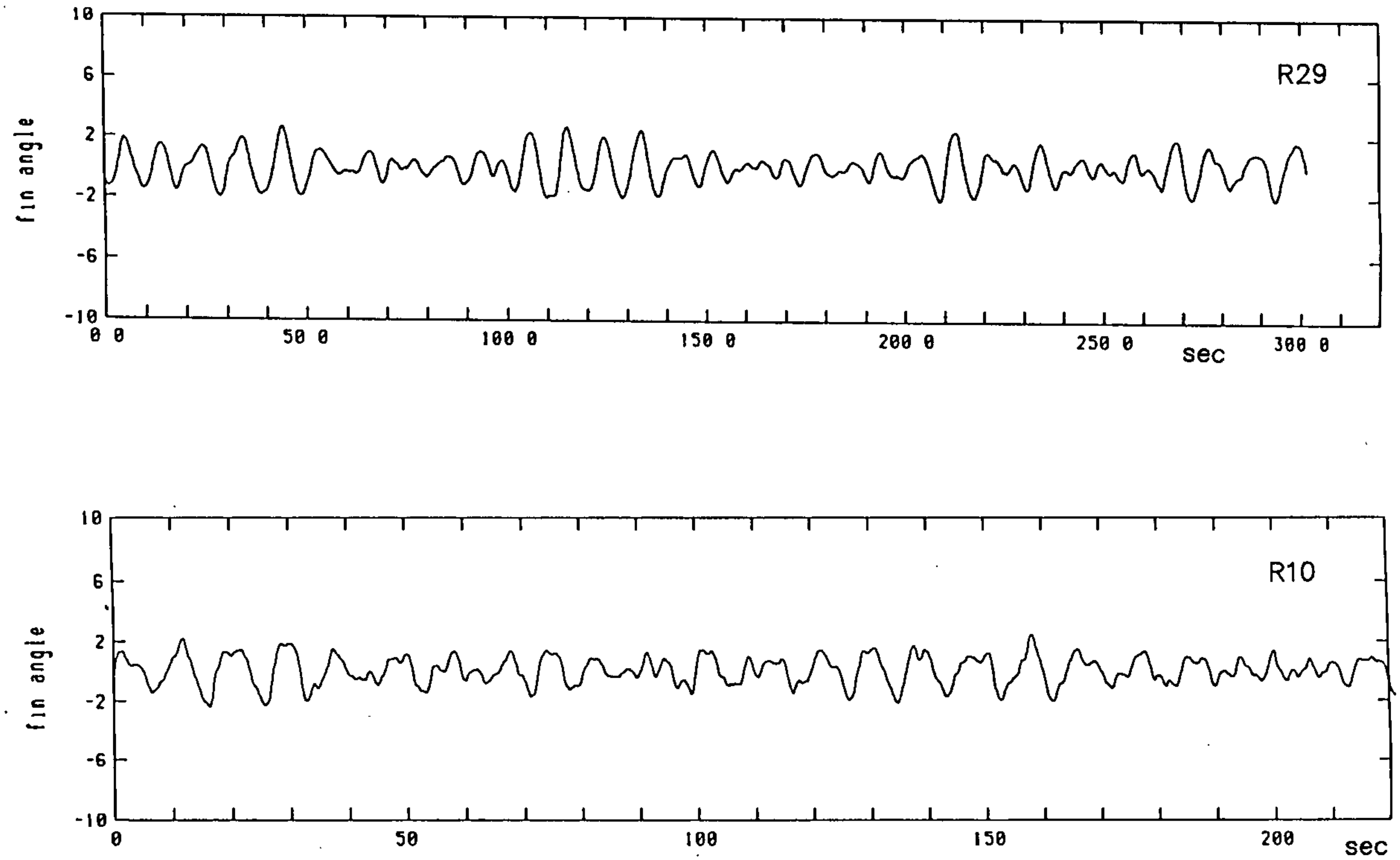


Symbols
+ RMS "sway" acc. / LFE
Δ RMS (LFE RMS - g*roll RMS) / LFE RMS
what the difference
1.00
0.75
0.50
0.25
0.00
% LFE/100
0
5
10
15
20
25
30
35
40
Ref No.
LFE Tuning Trials

FIG. 14

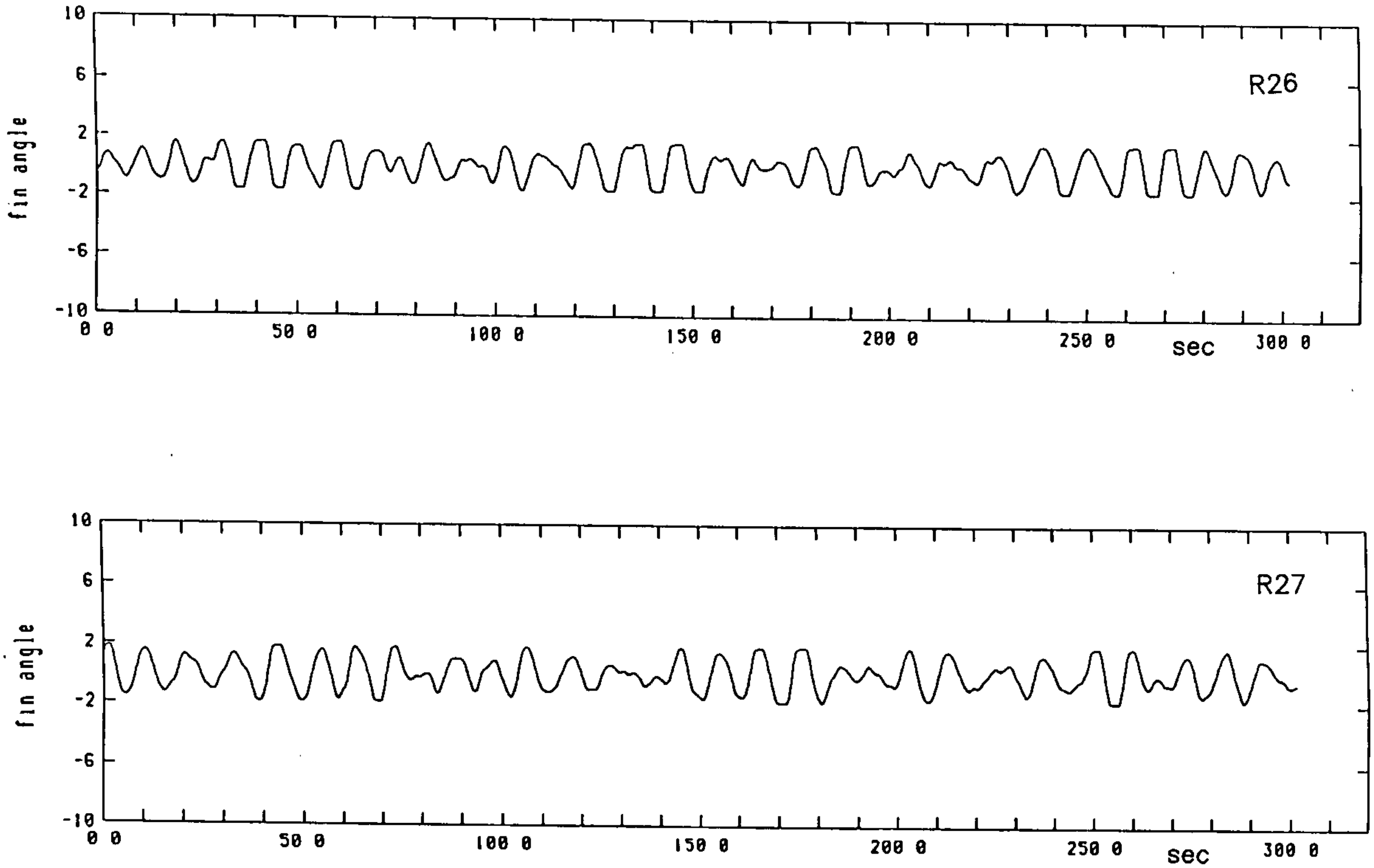


Fin motions of R2 and R3
FIG. 15



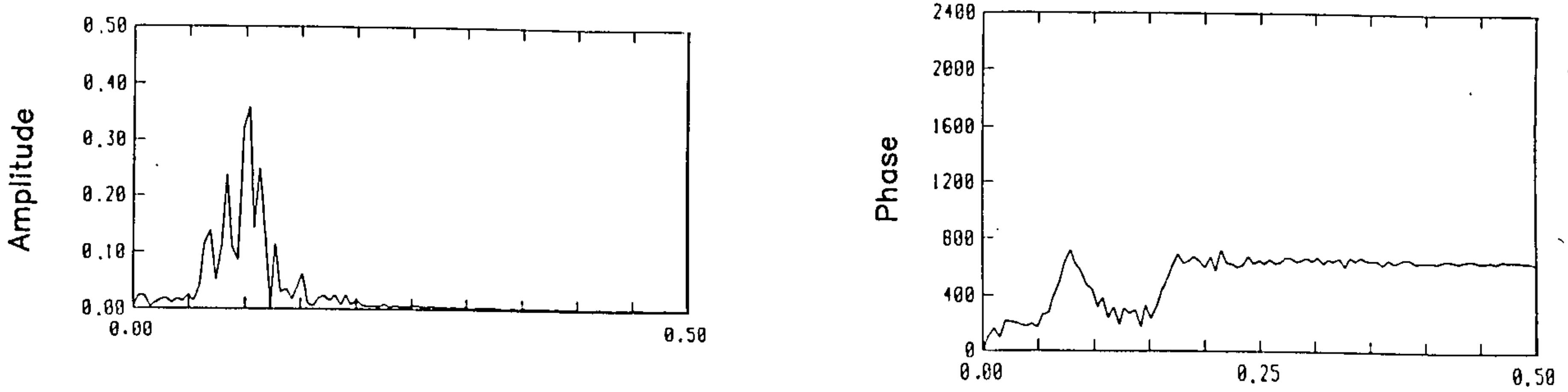
Fin motions of R29 and R10

FIG. 16



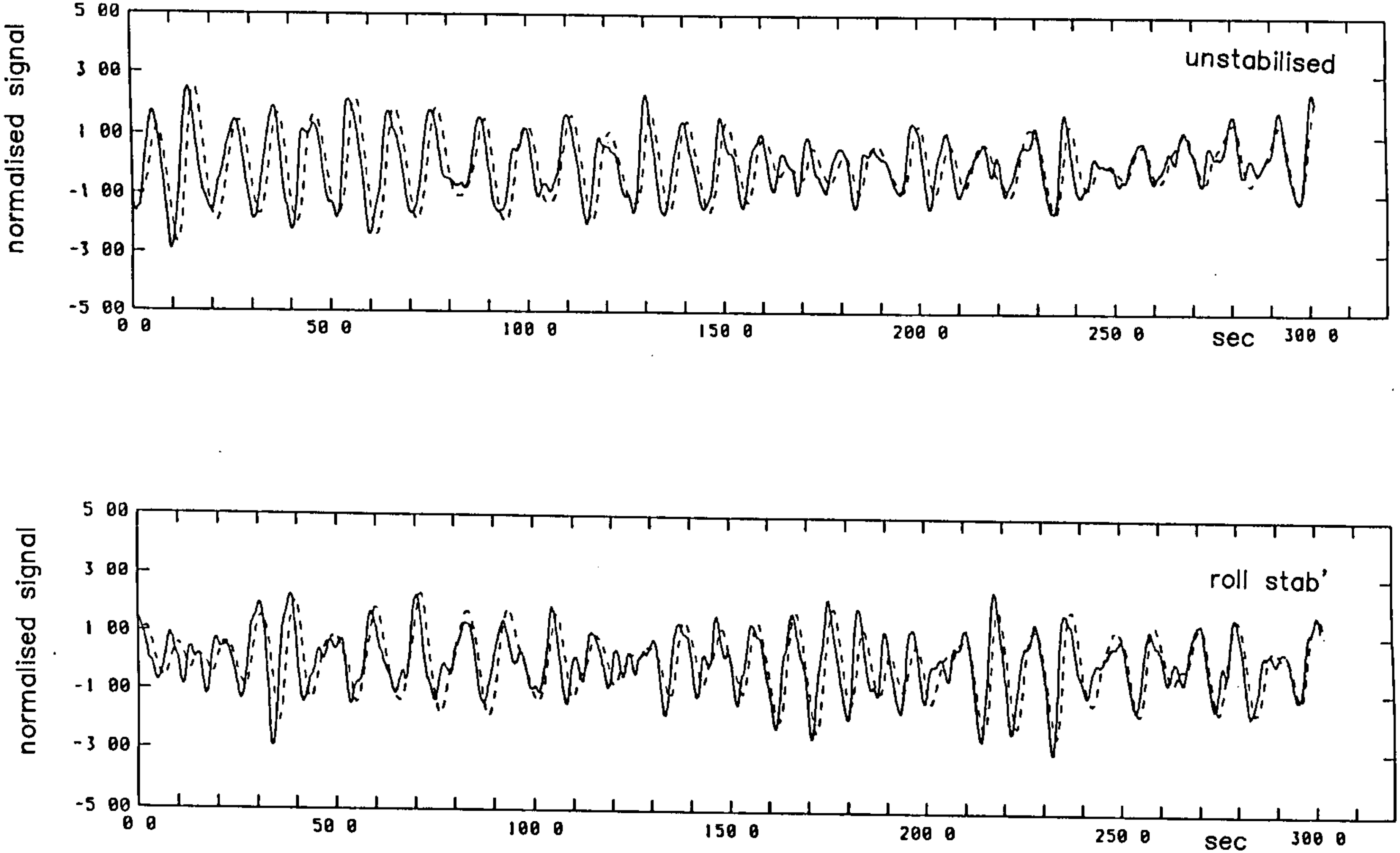
Fin motions of R26 and R27

FIG. 17



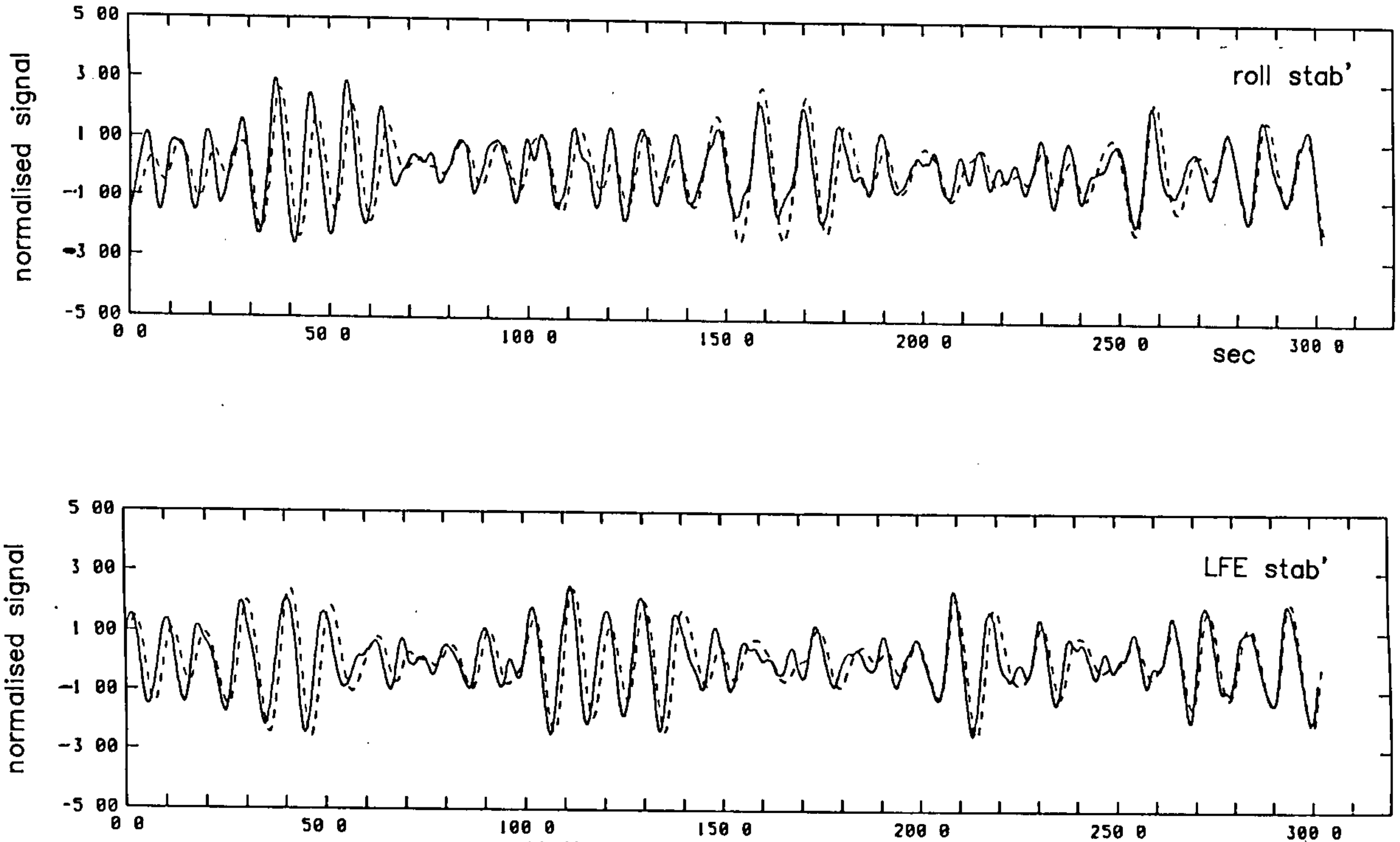
An example of the FFT output

FIG. 18



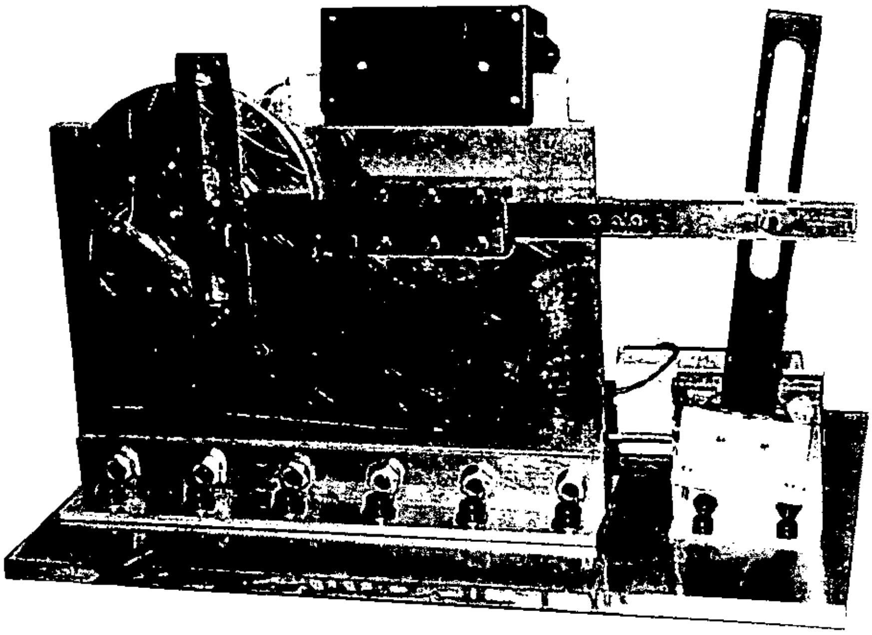
Motion comparisons for unstabilised and roll stabilised

FIG. 19



Motion comparisons for roll and LFE stabilised

FIG. 20



Mechanical Oscillator

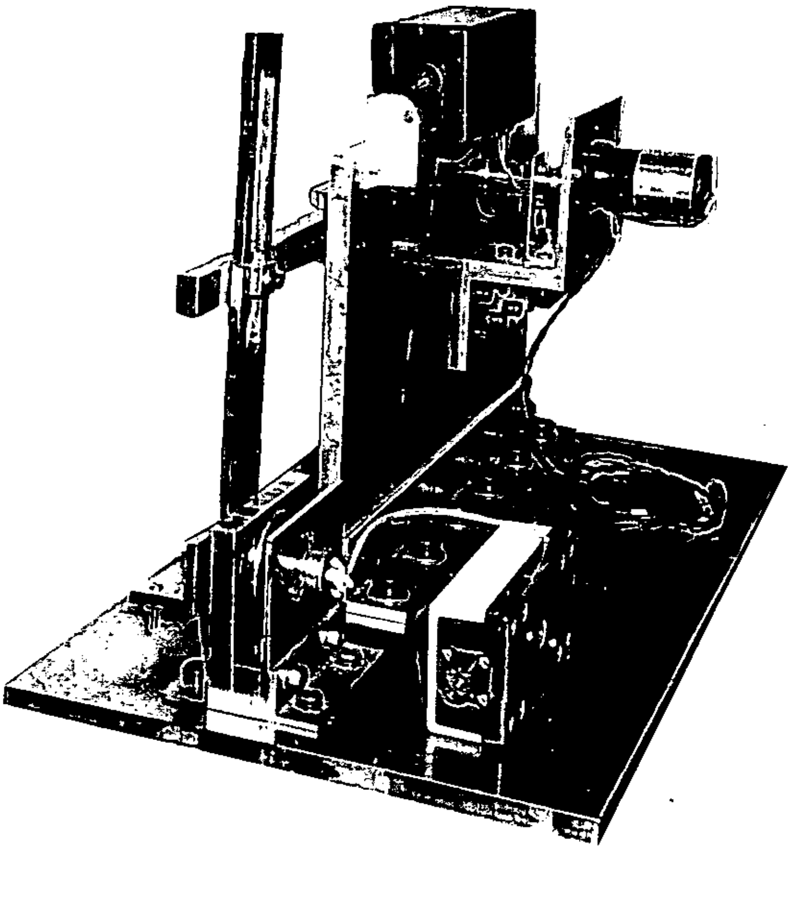
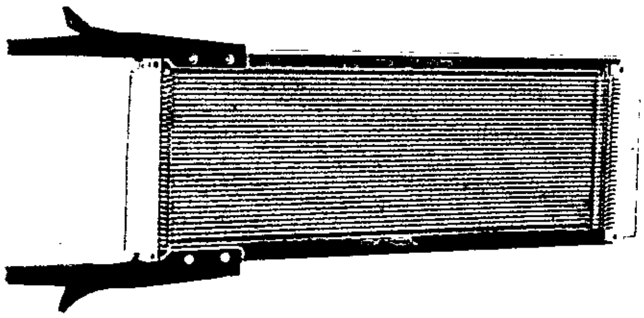
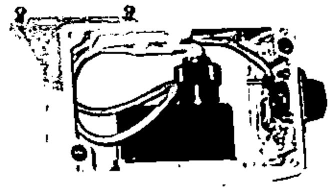


Plate 1



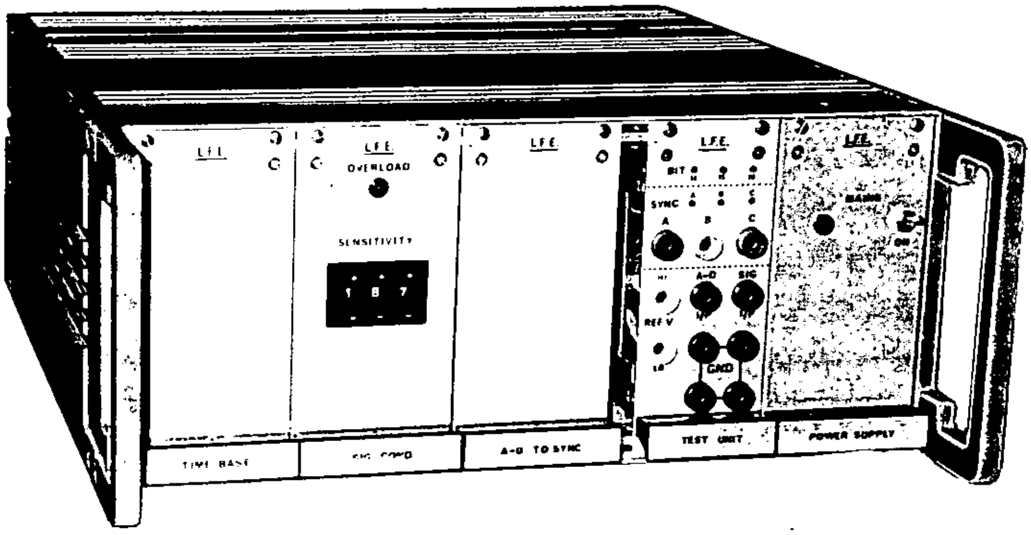
Extension Card

Eprom



LFE accelerometer
housed in a case

Plate 2



LFE Signal Box

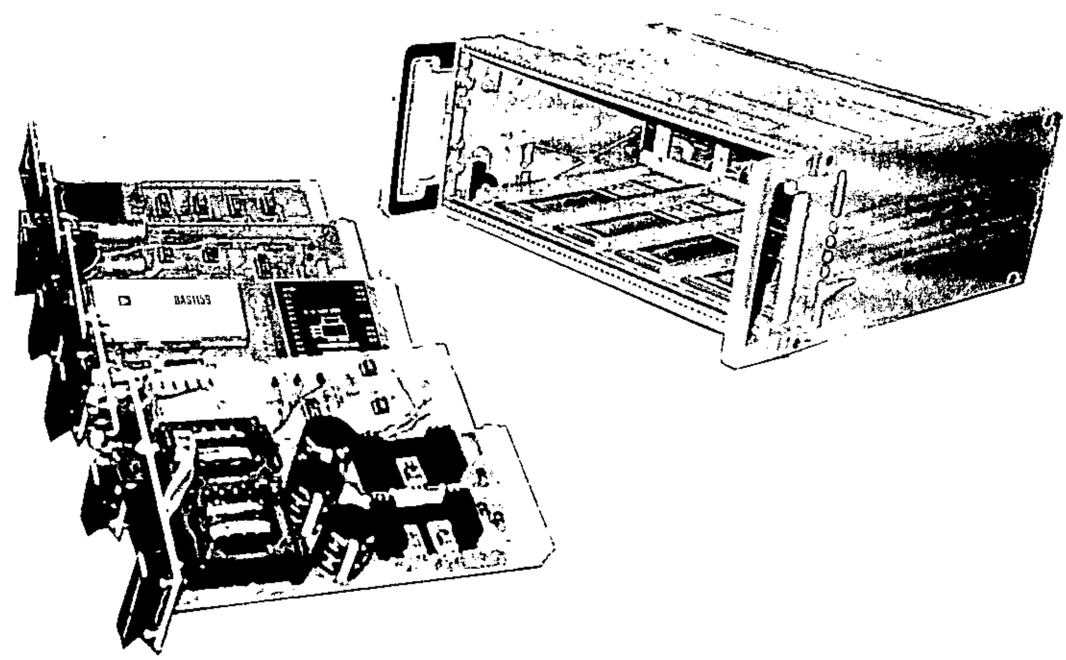
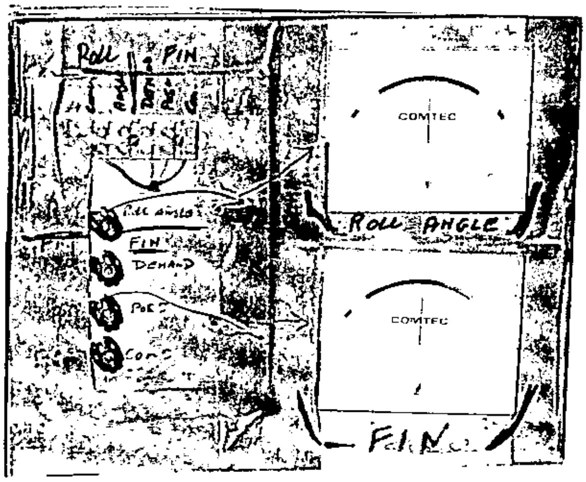
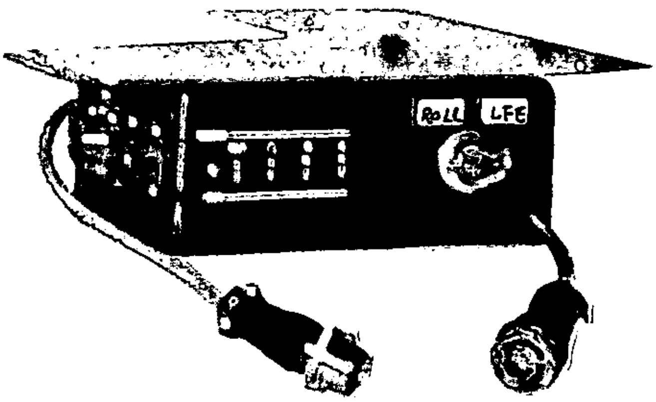


Plate 3



Fin and Roll / LFE Signal Box



Roll / LFE Turn-key Switch Box

Plate 4

APPENDIX A

Work example for converting the K values for the CCU

1. Gain term values

Figure A shows a block diagram of the LFE control law algorithm. The overall gain and the gains of the three terms are independently adjustible through the use of four sets of switches on the parameters card. These switches can be set in the range 00 to 99, allowing each gain term to be set from 0% to 99% of its maximum defined value. The values assigned to each gain term were confirmed by varying the parameter card settings while using the following service module routines to display their current values (to 2 decimal places).

	Service module switch setting	Maximum value
Proportional gain	1 0 1 4	4.0
1st derviative gain	1 0 1 5	2.0
2nd derivative gain	1 0 1 6	3.0
Overall gain	1 0 1 7	10.0

The frequency response of the existing Type 23 CCU uses an overall gain term set to 86, with the ships roll period (T) set to 8 seconds. These values give rise to the following gain terms within the control law algorithm.

Angle	:		=	1.0
Velocity	:	T/2PI	=	1.273
Acceleration	:	(T/2PI)**2	=	1.621
Overall	:	86/50 * 1.54	=	2.6488

To emulate the Type 23 CCU with the LFE CCU configuration, the switches were set as follows.

Angle (max=4.0)	:	(1.0/4.0)*100	=	25
Velocity (max=2.0)	:	(1.273/2.0)*100	=	64
Acceleration (max=3.0)	:	(1.621/3.0)*100	=	54
Overall (max=10.0)	:	(2.6488/10.0)*100	=	27

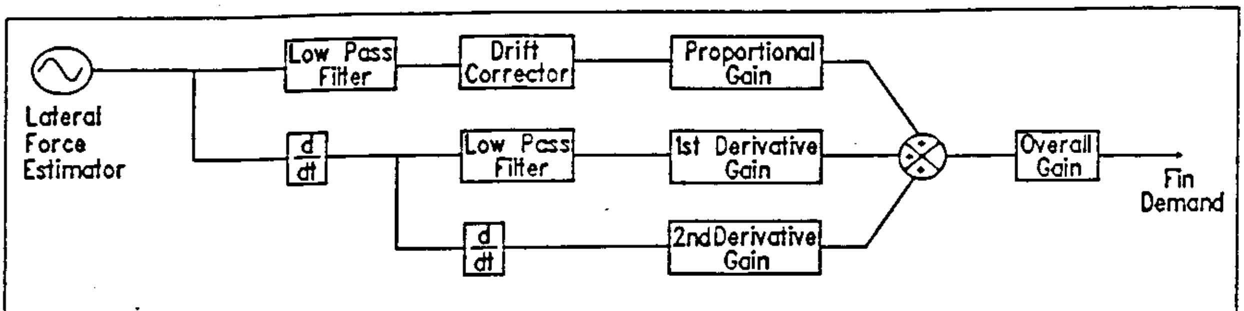


Figure A : LFE Control Law Algorithm

APPENDIX B

L.F.E. stabilisation Numerical Studies

1. Introduction

Ship motions can impair the human operator's ability to work onboard a vessel, degrading its operational effectiveness. Although it is possible for crew members to adapt to their working environments, it is unlikely that their adaptability would allow a high level of performance to be sustained as motions worsen. When motions become excessive, even the most experienced sailors would find it difficult to perform the simplest of manual tasks without overbalancing, stumbling, sliding on deck or having to hang on to something. This loss of balance of a person's stance is potentially harzardous and disruptive to work.

In a study by Baitis et al (1983), it was suggested that this loss of balance is related to the ship's lateral accelerations known as the Lateral Force Estimator (LFE). Based on this parameter, a performance index called the Motion-Induced Interruption Index (MII) can be derived, which can be used to assess the likelihood of disruption to tasks. Some applications of this index can be found in Baitis et al (1984) and Graham (1990). However, from the motion point of view, LFE is more relevant as it is the vector sum of the rigid body accelerations in the plane of the deck including a term directly proportional to the roll angle, which can be monitored readily onboard ship. In Lloyd (1986), LFE has been used to assess the limiting ship motions for helicopter operations with some promising results, and in Monk (1989), LFE was used as a criterion to assess roll motions. It is quite obvious from these studies that a low LFE level is desirable.

At present, most of the warships in the Royal Navy are stabilised for roll using the active fins system. But if LFE is a better measure for crew's performance, it is quite logical to seek ways to suppress LFE instead of roll motions alone. In fact this sentiment has been expressed in Lloyd (1986). This led to the initial investigation into LFE stabilization made by Theobald (1986), but unfortunately it stopped short in the early stages. In the past, this type of stabilization was known as the stabilization to the apparent vertical (Bell (1965)).

The main objective of this report is to investigate the feasibility of LFE stabilization using the existing active fins system. The numerical study was carried out on a modified version of the PAT-86 (MOD seakeeping computer program), making the fins sensitive to LFE signals instead of roll angles. Some related studies on MII and vertical motions are also included for future developments.

2. Numerical Studies

From Lloyd (1989), it can be shown that

$$LFE - \ddot{y} - z\,\ddot{\varphi} + x\,\ddot{\chi} - g\,\varphi \tag{1}$$

for small roll amplitudes, where x, y, z are the co-ordinates of the point in question, and φ and χ are the roll and yaw angles respectively.

Equation (1) was used to replace the roll angle term in the transfer function in the numerical model, giving

$$\frac{\beta}{LFE} - K' \left[\frac{K_1 + K_2\, s + K_3\, s^2}{b_1 + b_2\, s + b_3\, s^2} \right]$$

where K_1 ,K_2 and K_3, the roll angle, velocity and acceleration gain levels, which have to be specified, while b_1, b_2 and b_3, the fin angle, velocity and acceleration gain levels, are fairly standard for a chosen control system. A tuning procedure proposed by Marshfield (1980) has been followed for the selection of the K_i values. More details about tuning for LFE can be found in Theobald (1986). The overall gain K' was set by trial and error until the r.m.s. fin value coincides with a prescribed value operating in a specified sea state. In the present study, all simulations were carried out using the ITTC two parameter spectrum (long-crested) with a significant wave height of 5.5 metres and modal period of 12.4 seconds. The ship speed was set at 20 knots running in beam seas. Different conditions were used for the MII study.

2.1 Tuning

As a first attempt, the LFE system was tuned at the roll natural frequency. The output were compared with the roll stabilized and unstabilized case, using the r.m.s. LFE at c.g., r.m.s. roll angle and r.m.s fin angle as the basis of comparison. Table (1) shows the corresponding data.

Table (1) Comparisons for Tuning at the Roll Natural Frequency			
	r.m.s. fin (deg)	r.m.s. roll (deg)	r.m.s LFE (m/s^2)
unstab'	—	3.840	0.791
roll stab'	8.2	0.957	0.439
LFE stab'	8.2	3.752	0.694

It can be seen that the LFE system is not as effective as the roll stabilised system. A glance at the transfer function plots in fig. 1 reveals that only a small reduction in amplitude has been achieved near the natural roll frequency, which accounts for a substantial part of the lateral accelerations. At higher frequencies, the LFE system produces greater reductions. Although this set of results is not too encouraging, the phase angles in fig. 2 do show consistent trends to the asymptotic approximations given in Schmitke (1978) as the wave frequency tends to zero, which gives some confidence to the credibility of the results.

From equation (1), it can been seen that LFE is also a function of the sway and yaw acceleration, which do not have resonant frequency characteristics. Therefore, it is quite possible that tuning at the roll natural frequency would not achieve the best possible result. With this in mind, a range of tuning frequencies was tried following the same tuning procedure. Indeed a minimum r.m.s. LFE response was found, which also coincided , interestingly enough, with the minimum r.m.s. roll displacement. The transfer function for this *optimum* tuning is given in fig. 3 - a substantial reduction near the natural roll frequency by the LFE system is apparent. It has lowered the r.m.s. LFE level by 19% and increased the r.m.s. roll angle by 3% compared to the roll stabilized case.

2.2 Parametric studies and Discussions

Thus the preliminary study proved that controlling LFE is feasible in beam seas. The next step would be to examine the wave angle dependence .

In the wave angle case (fig.4-6), it is evident that running in beam seas imposes the highest fin demand. The LFE system compares favourably to the roll stabilized case in LFE levels, especially near beam seas. However, in terms of roll displacement, the LFE system does not perform as well in bow seas. If accelerations perceived by human operators are more important, the roll displacement magnitude may not be too significant as long as it falls within some specified operational limits.

Across the range of the modal frequencies used, the r.m.s. LFE values are lower in the LFE stabilized case, and as the wave modal period increases, the r.m.s. response decreases (fig. 7). This trend stems from the fact that as modal period decreases, i.e. frequency increases, the sway term in the LFE becomes more apparent. Again, the roll displacement is higher in the LFE system (fig. 8). However, the fin demands should be taken into account when making assessments (fig. 9).

As ship speed increases, the lift produced by the fins to counter-act roll motions increases. In order that noise and cavitation problems are alleviated, speed compensation is essential to active fin stabilization. Therefore, at moderate speed (16-24 knots), little variation in the r.m.s. roll displacement would be expected when operating in a particular sea state. However, as speed decreases, the fins are made to work harder in order to achieve desirable roll motions, and below 10 knots, the fins are in fact switched off due to mechanical wear and tear. Fig. 11 and 12 seem to have borne out these facts quite well. The LFE system has again showed improvement in terms of LFE magnitudes over the standard roll stabilization(fig. 10).

3. MII studies

In Graham (1990), MII has been extended to the frequency domain, including the effect of vertical accelerations. As the LFE is readily available in the present study, MII would be a natural extension, which would be of use for seakeeping performance studies. According to Graham,

$$MII = \frac{T_T}{T_L} \exp\left[-0.5\left(\frac{(l/h)g}{LFE_{r.m.s.}}\right)^2\right]$$

$$T_L = 2\pi\left(\frac{m_0}{m_2}\right)^{\frac{1}{2}} \qquad (2a, b)$$

where T_T task duration

T_L zero crossing period

m_n n^{th} moment of the LFE spectrum

$LFE_{r.m.s.}$ replaces by GLFE when vertical accelerations are included

l/h ratio of moment arms about a person's foot to his c.g. (0.25)

Using these equations, the MII levels in relation to the LFE magnitude can be derived. A comparison to Graham and Baitis's data is shown in fig. 13. The difference may be due to the estimation of LFE and the T_L used, but on the whole, it suggests that equation 2 gives reasonable MII estimates.

The LFE level is expected to decrease as the vertical location of the point of interest approaches the keel. This fact is apparent in fig. 14, which is also consistent with Graham's result. However, the MII level in this study is far lower than that of Graham's. This is probably due to the lower wave excitation experienced by the ship in this simulation. The inclusion of vertical accelerations has changed the LFE levels but in a less pronounced fashion compared to Graham's. The present study suggests that there is little variation on LFE along the ship longitudinal axis with a slight maximum at the stern or bow (fig. 15). It should be pointed out that Graham's result might not have been worked out using the spectrum method but with a method similar to Monk (1989), in which case the LFE *estimates* could be very different.

4. Conclusions

It has been shown that with proper tuning, LFE stabilization can be a feasible alternative, employing the existing fin systems with relatively minor modifications. The extension to MII also proves to be promising, providing a sound basis for comparative seakeeping performance study. However, at this stage, it would be expedient to perform forced-roll trials on LFE before undertaking any major numerical studies as the tuning procedure relies quite heavily on the forced-roll LFE data.

Symbols

β	fin angle
s	Laplace transform $j\omega$
K_1	displacement gain in controller
K_2	velocity gain in controller
K_3	acceleration gain in controller
b_1	fin displacement gain
b_2	fin velocity gain
b_3	fin acceleration gain

Axis system

x	+ve forward
y	+ve starboard
z	+ve downward
φ	roll +ve starboard
χ	yaw +ve starboard

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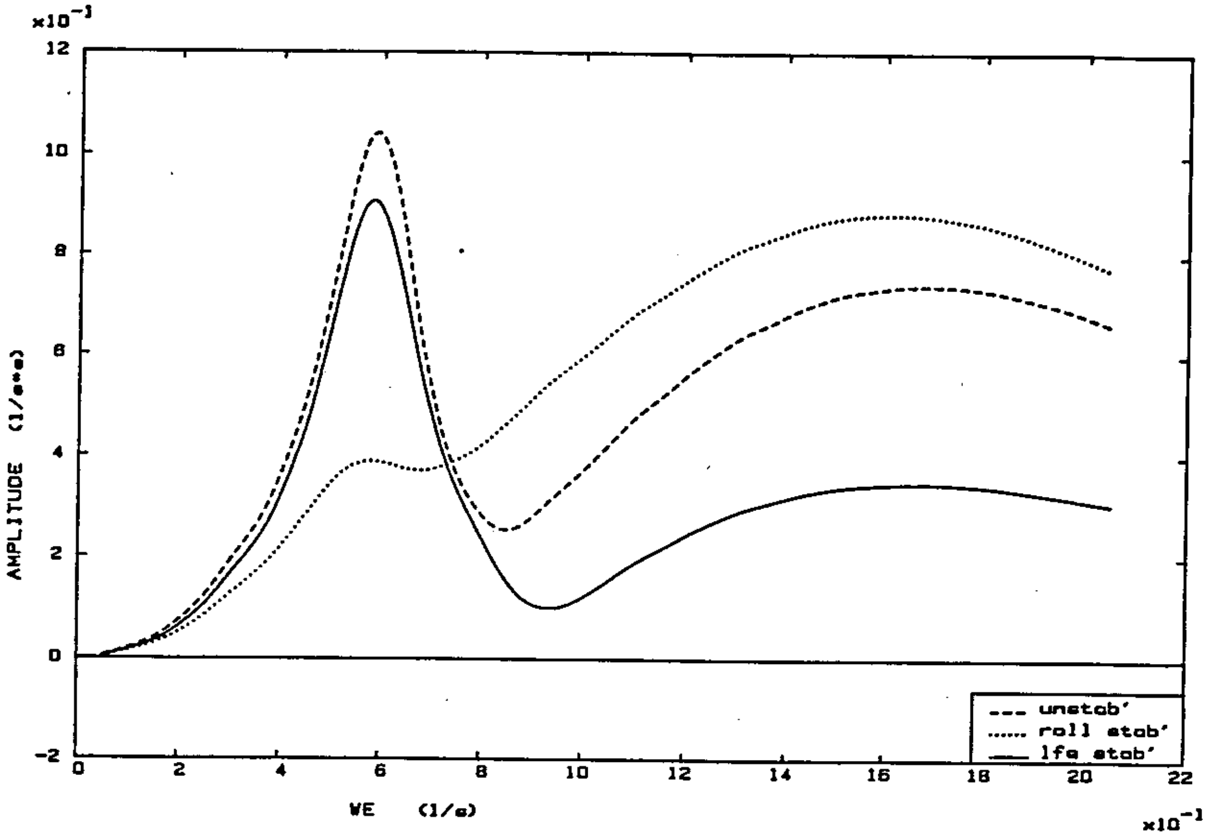


Fig. 1 LFE TRANSFER FUNCTIONS TUNING AT NATURAL ROLL FREQ

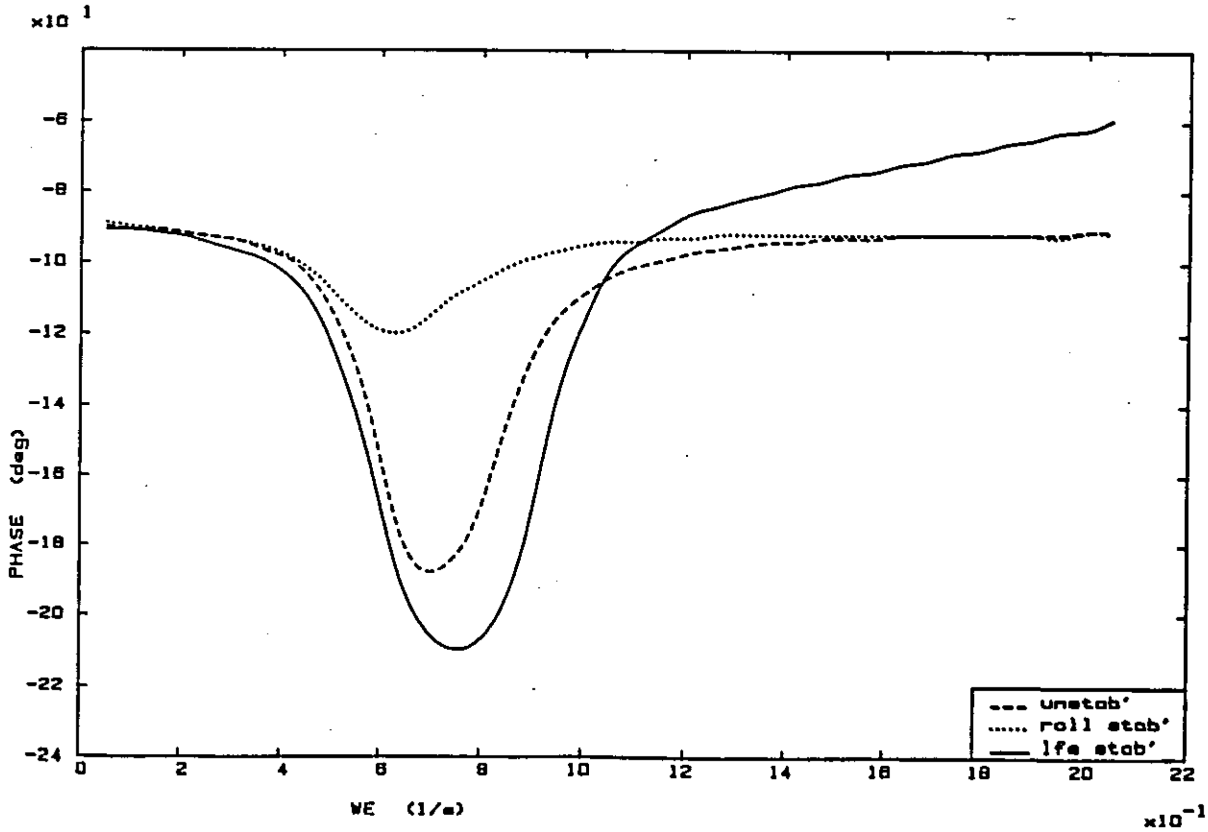


Fig. 2 LFE PHASE RESPONSE TUNING AT NATURAL ROLL FREQ

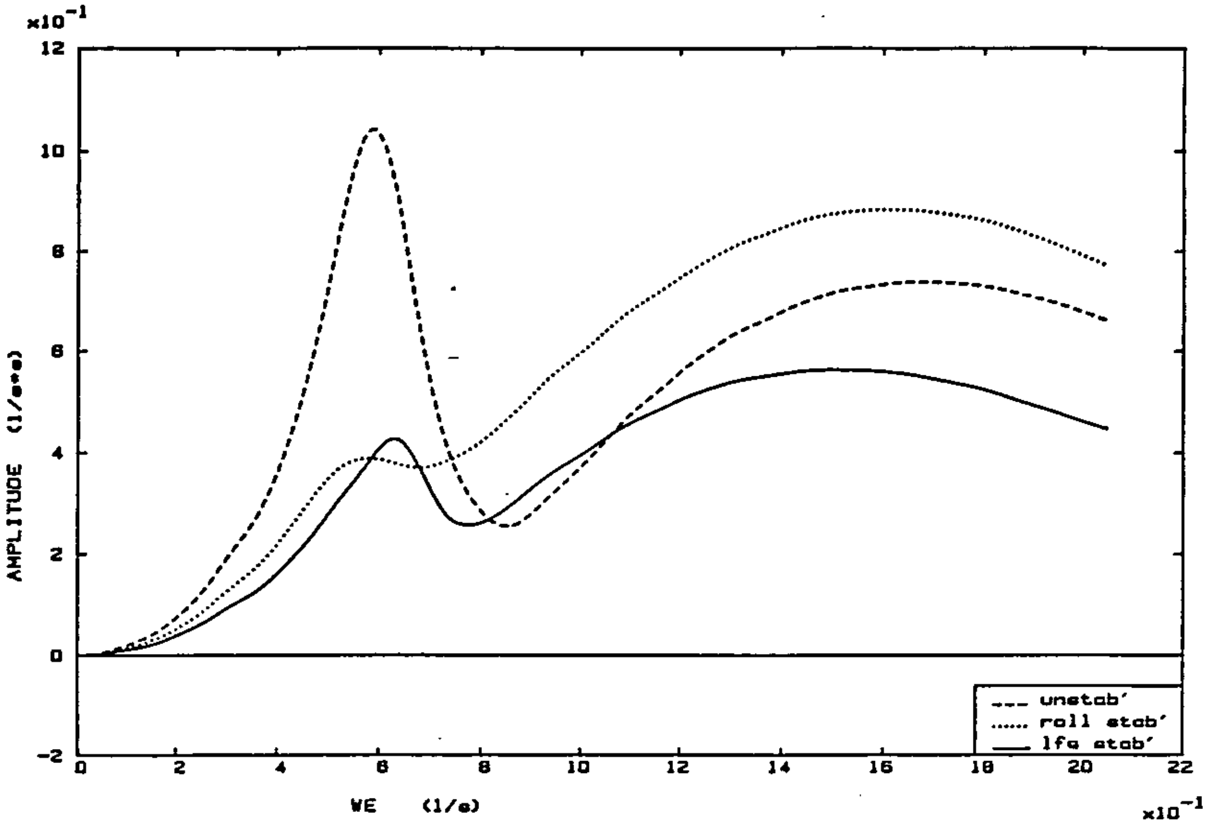


Fig. 3 LFE TRANSFER FUNCTIONS TUNING AT 'OPTIMUM' FREQ

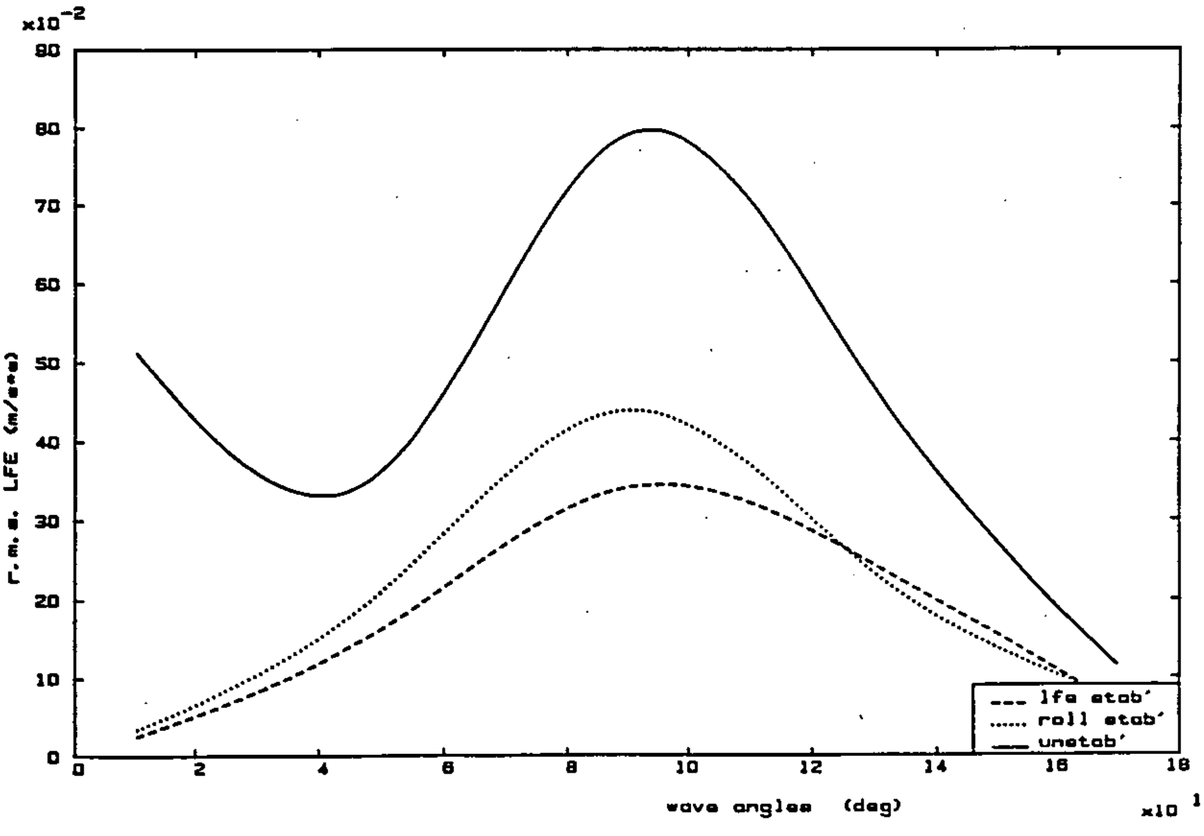


Fig. 4 LFE RESPONSES AT DIFFERENT HEADINGS

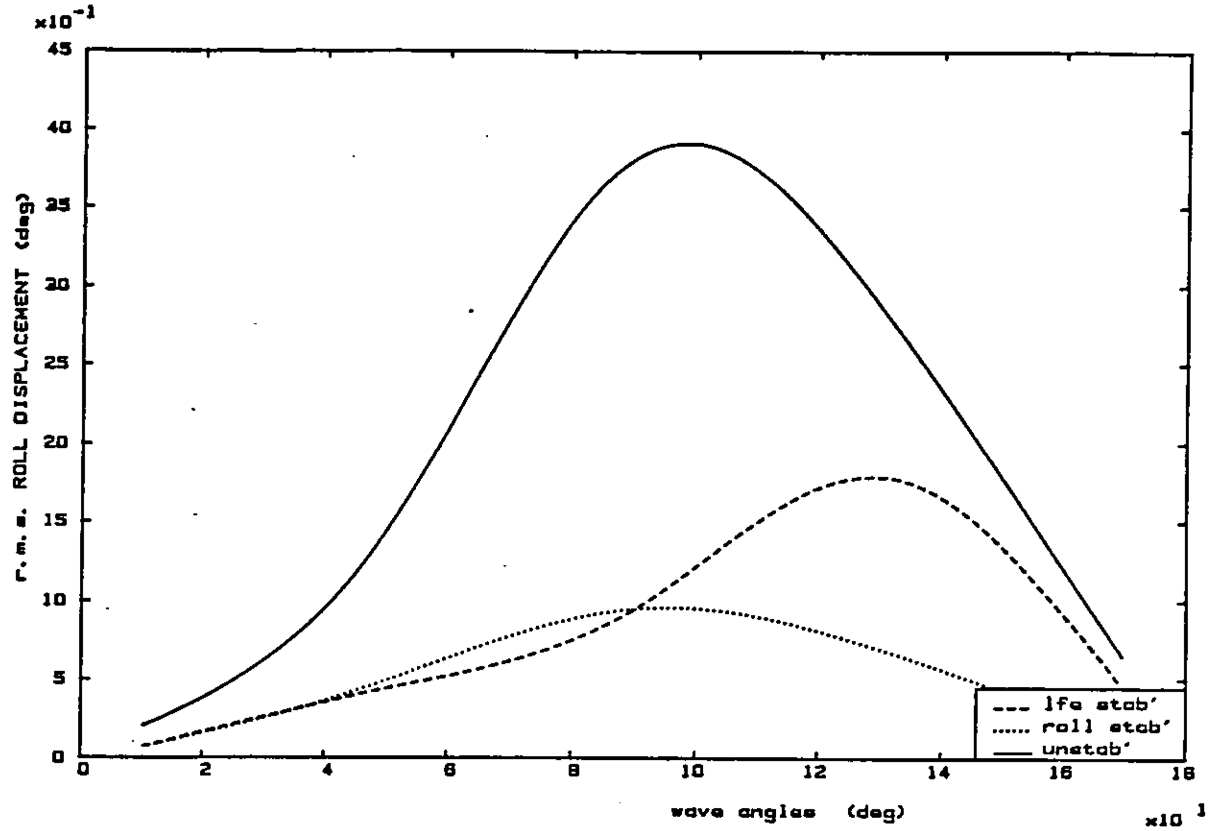


Fig. 5 ROLL RESPONSES AT DIFFERENT HEADINGS

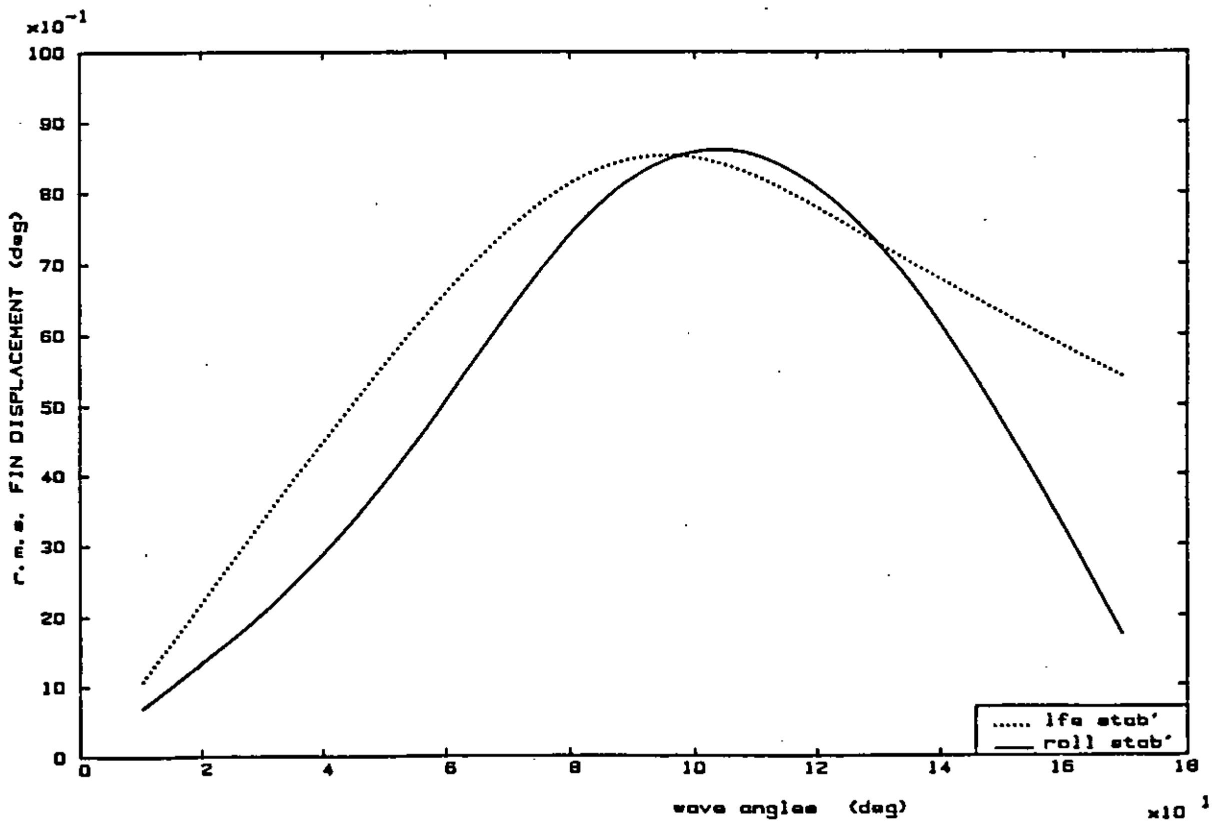


Fig. 6 FIN RESPONSE AT DIFFERENT HEADINGS

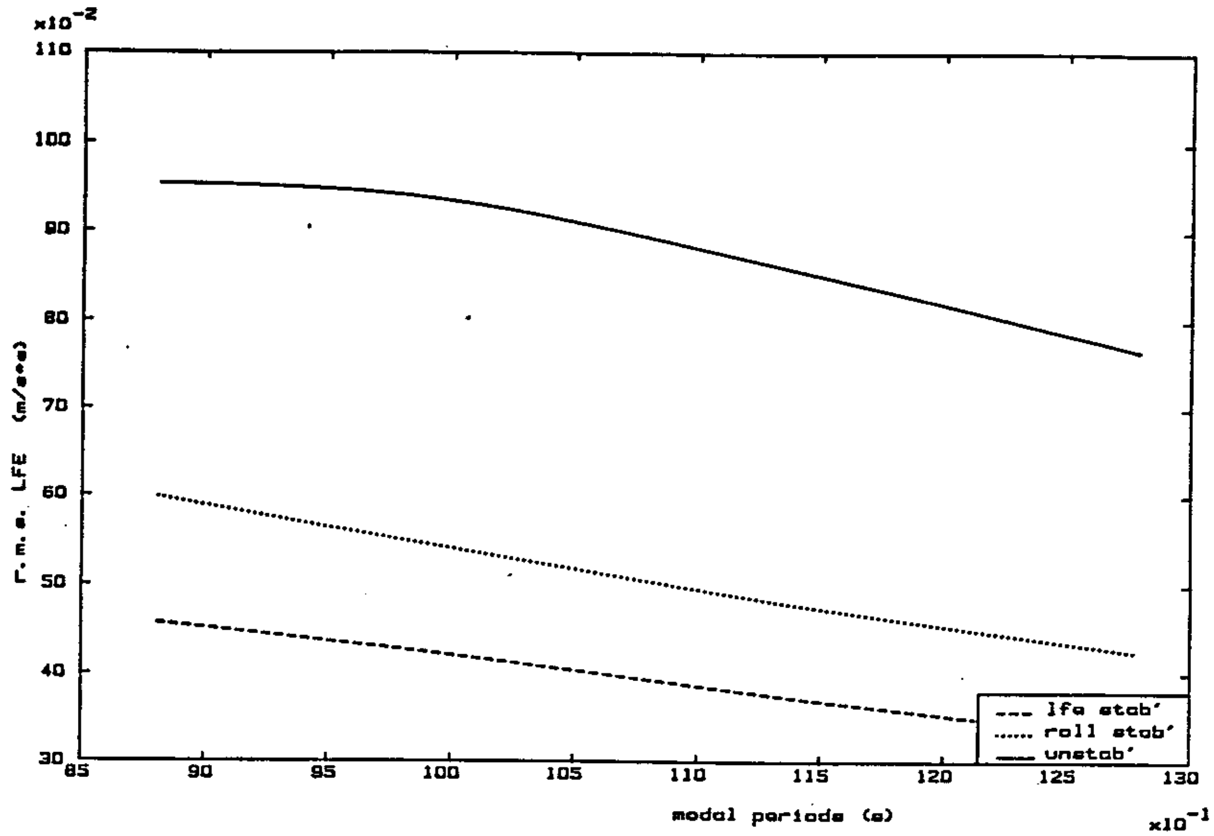


Fig. 7 LFE RESPONSE AT DIFFERENT WAVE MODAL PERIODS

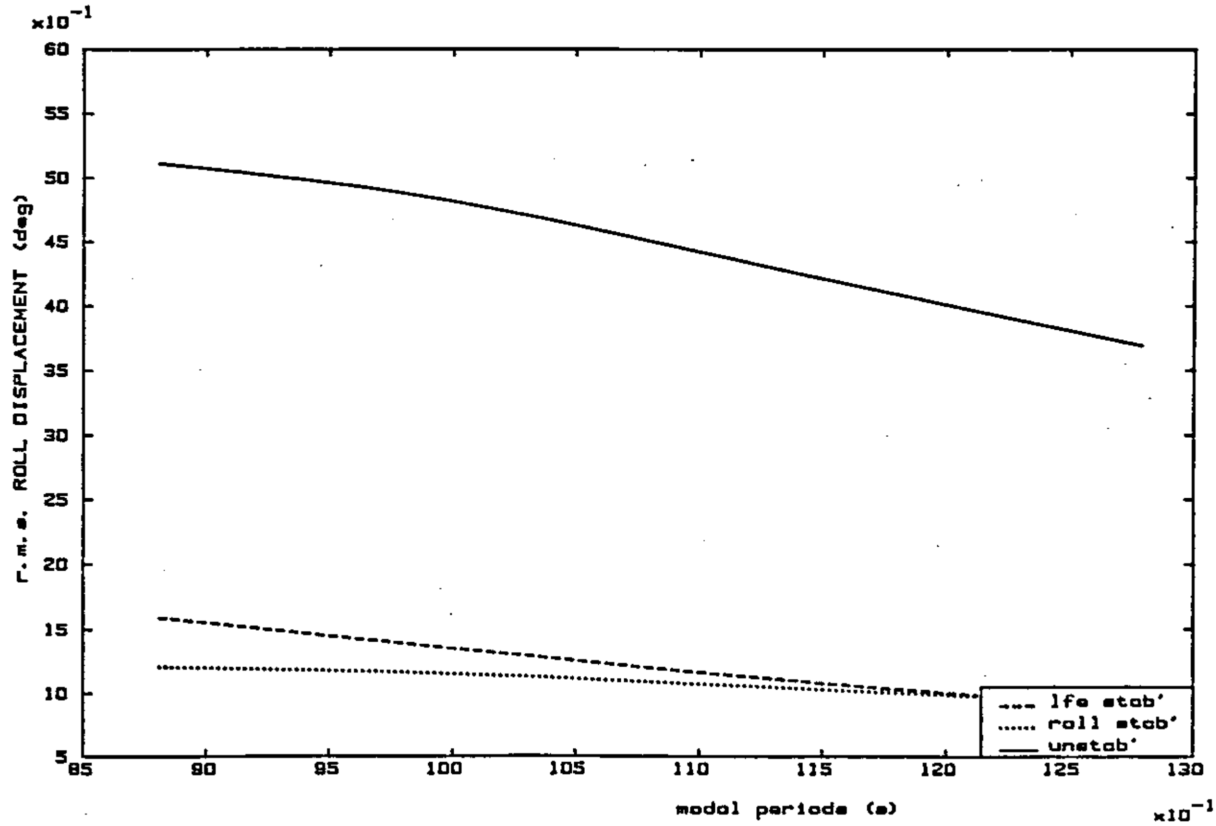


Fig. 8 ROLL RESPONSE AT DIFFERENT WAVE MODAL PERIODS

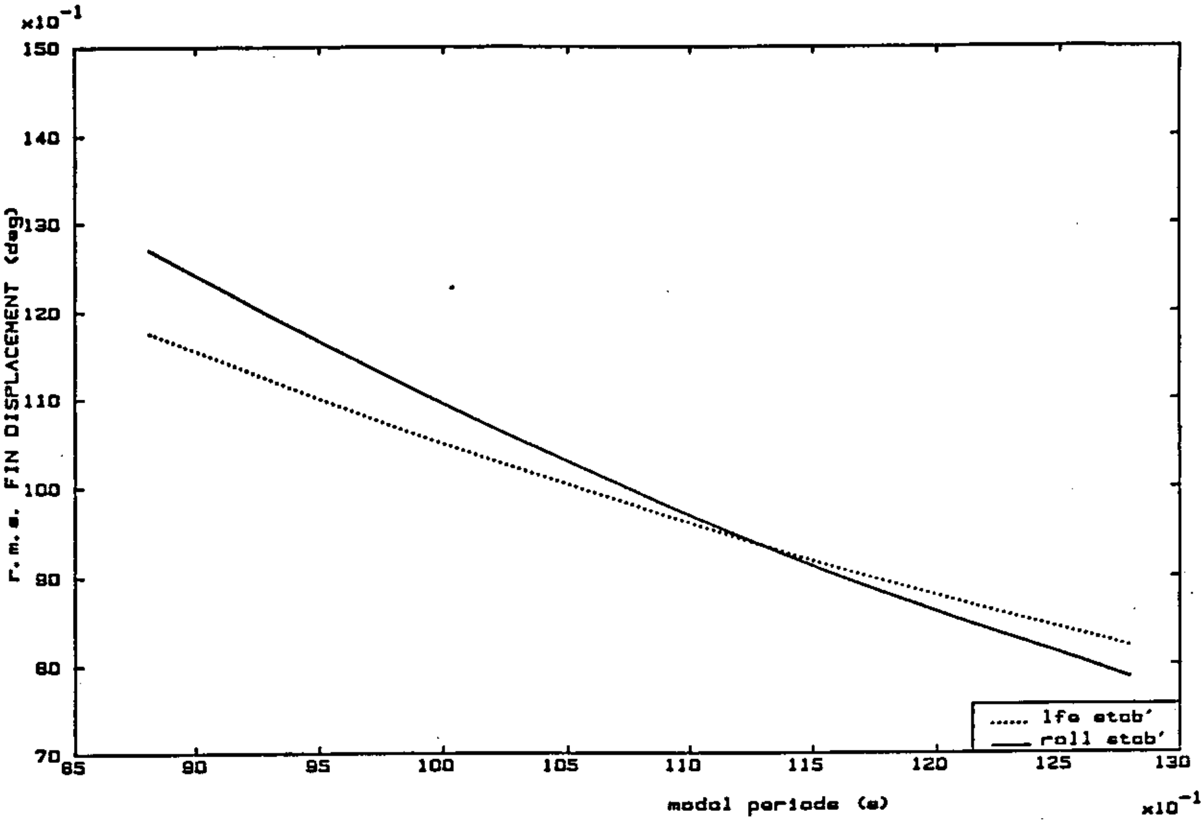


Fig. 9 FIN RESPONSE AT DIFFERENT WAVE MODAL PERIODS

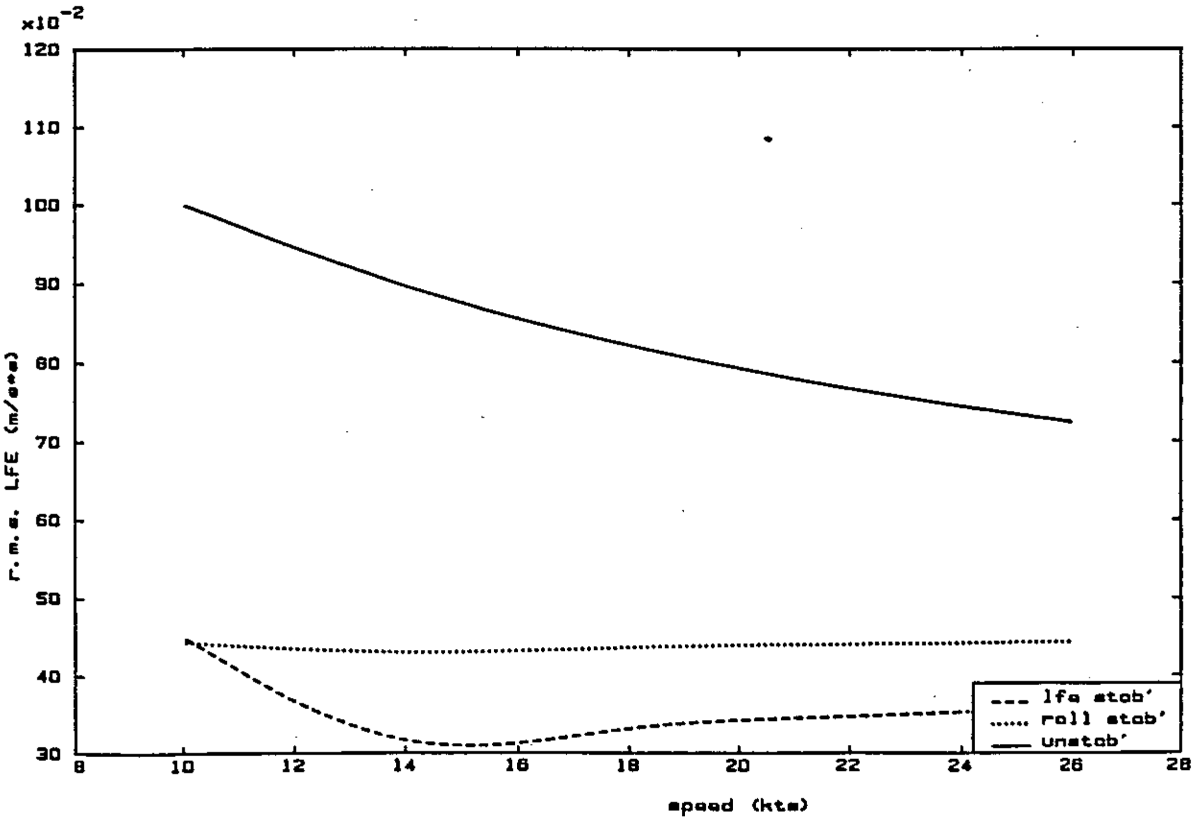


Fig. 10 LFE RESPONSE AT DIFFERENT SHIP SPEEDS

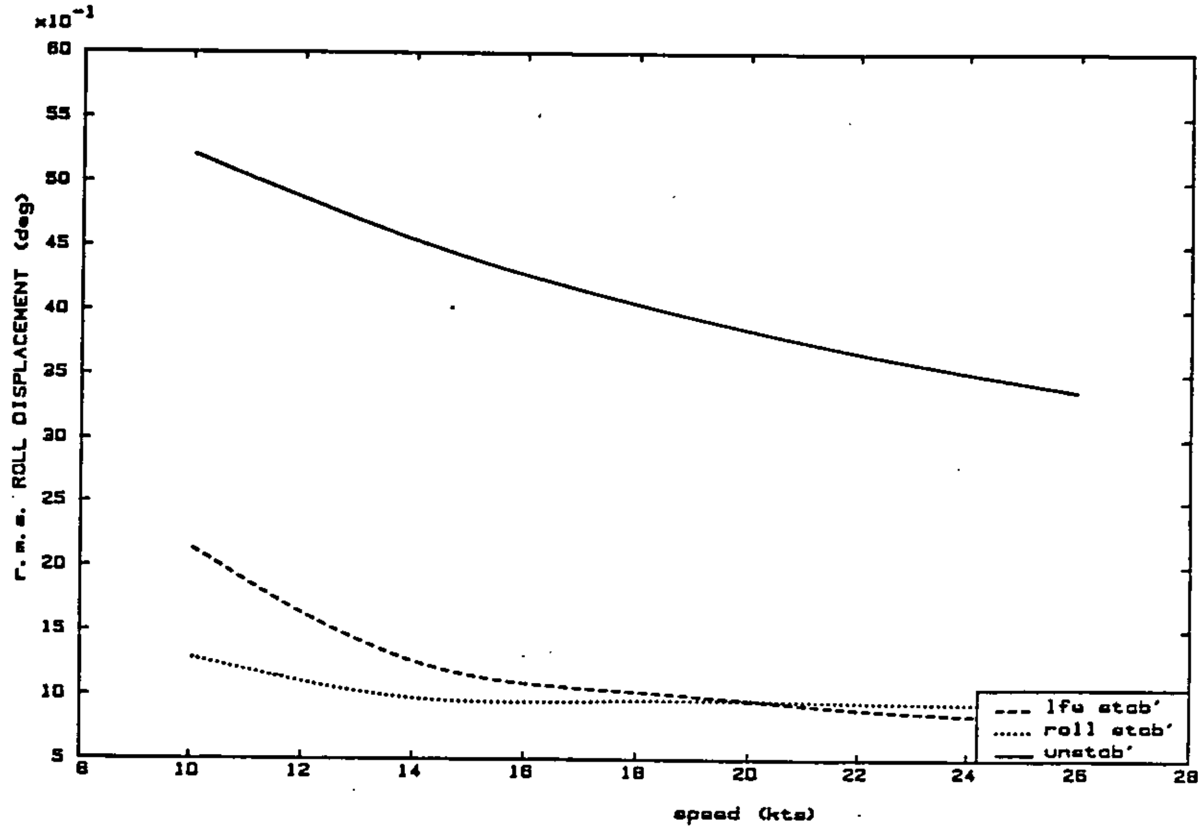


Fig. 11 ROLL RESPONSE AT DIFFERENT SHIP SPEEDS

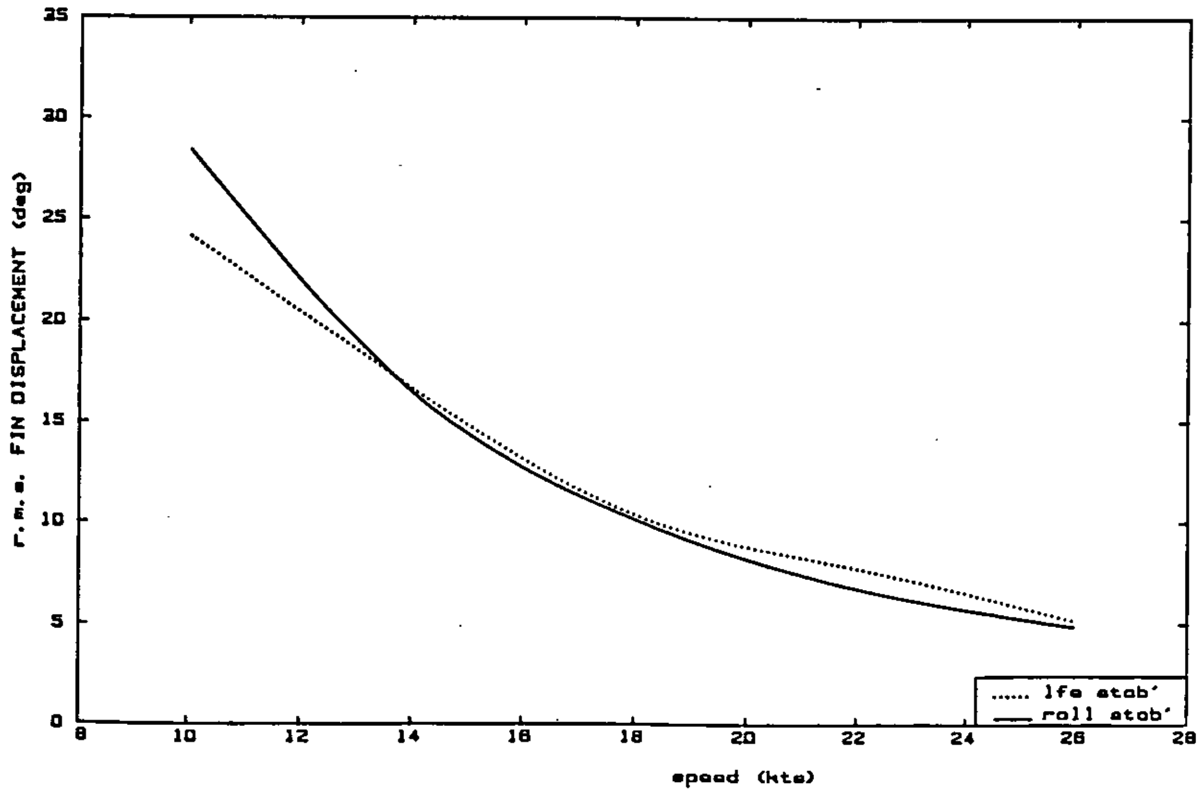


Fig. 12 FIN RESPONSE AT DIFFERENT SHIP SPEEDS

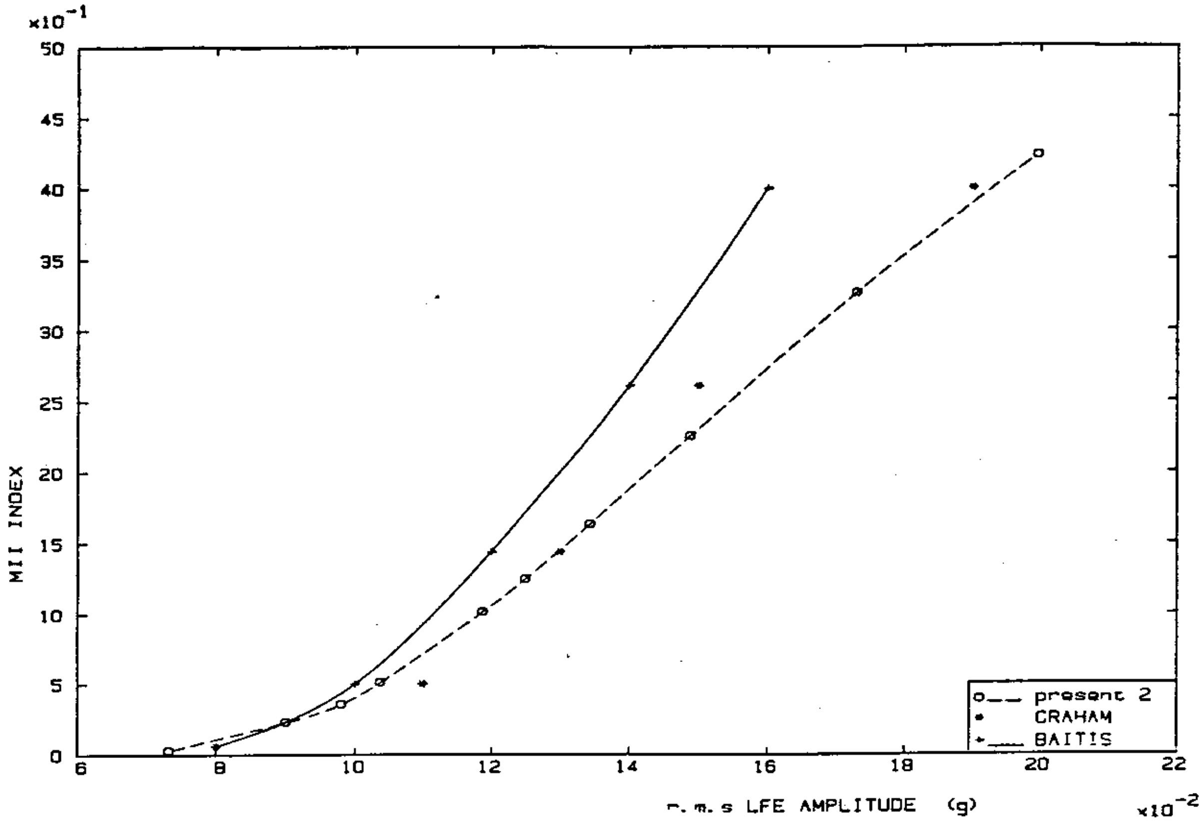


Fig. 13 MII LEVEL COMPARISONS

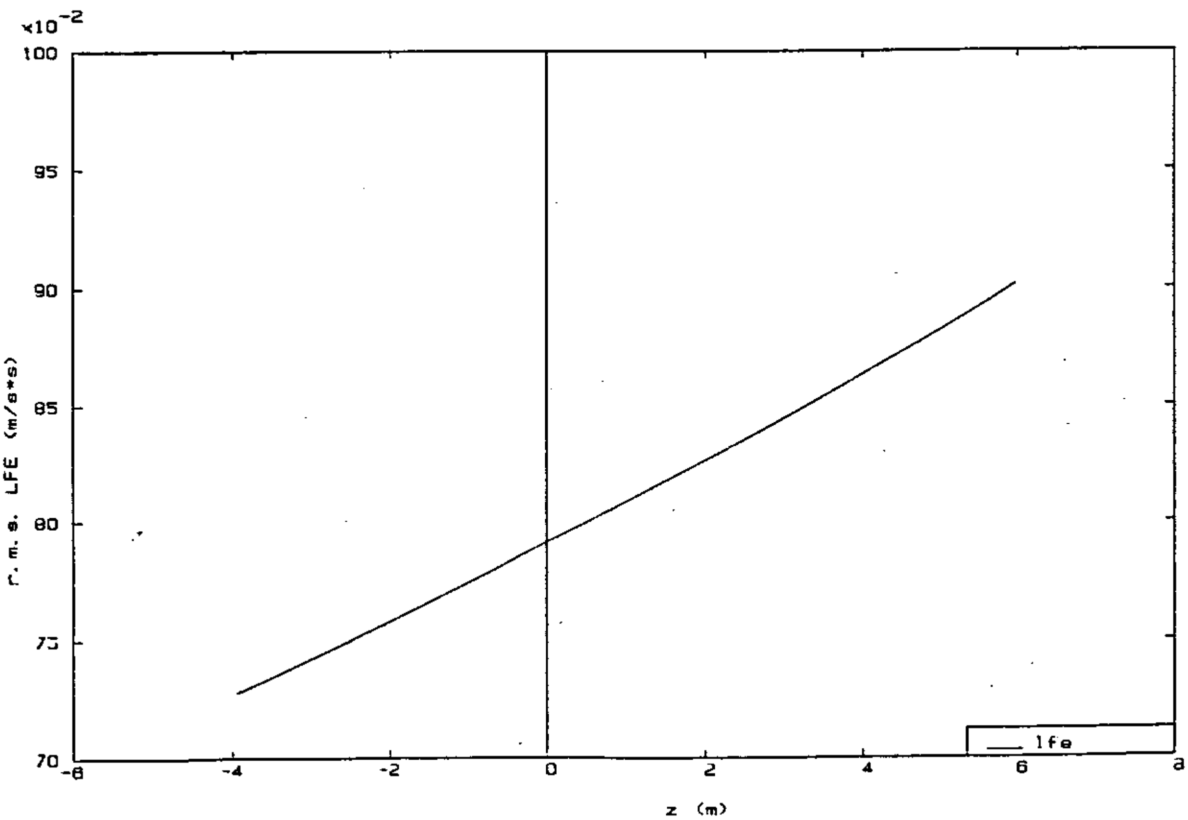


Fig. 14 Effect of vertical location on MII

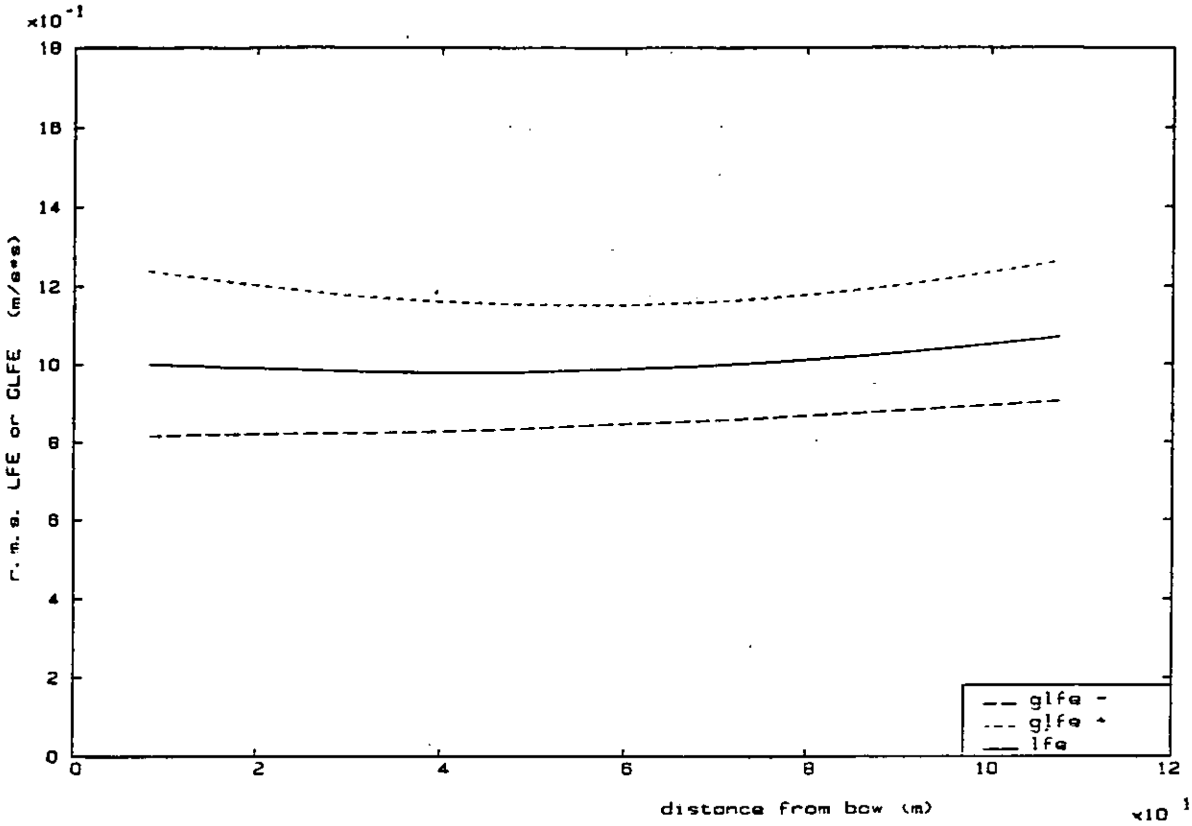


Fig. 15 GLFE RESPONSE AT DIFFERENT STATIONS 5.5m 10.8s 105deg

APPENDIX C

An investigation into L.F.E. stabilisation using the rudder

This report comprises of two parts. Part one is concerned with the numerical studies carried out to assess the possible use of the rudder for LFE stabilisation of a frigate. Part two is a summary of the hard-ware design for the acceleration signal conditioning to be used in LFE control for the sea-trials.

1.0 Introduction

It has been shown in the previous numerical study (Tang [23]) that LFE stabilisation using active fins can be a feasible alternative to the conventional roll stabilisation strategy. Coupled with the recent renewed interest in rudder roll stabilisation (RRS), this leads to the idea of making use of the rudder for LFE stabilisation (RLS). Intuitively, as the rudder is located near the flight deck, good motion control for the flight deck would be expected with the proper tuning of the rudder control system. In this study, a review on RRS was first carried out, from which lessons were drawn from the experience gained in the development of RRS. A similar procedure to the LFE stabilisation using the active fins outlined in Tang [23] was applied in exploring the feasibility of LFE stabilisation by the rudder. Comparisons were made with RRS to assess the merits of RLS and based on this numerical study, conclusions were drawn.

2.0 Review

Traditionally, course-keeping control and motion control (notably roll motion) have developed along independent lines. The rudder has been used solely for heading control, and active fins, for example have been traditionally been used for suppressing only roll motions. However, during the early days, some active fin systems were found to interfere with the steering control of a ship. This stemmed from the fact that the fins would produce a net yawing moment if they were not placed near the longitudinal centre of gravity (l.c.g.) and also if they had a relatively large angle of depression. This was really an indication of the inadequacy of the traditional single input/output approach, which ignored the cross-couplings effects in the motion characteristics of the ship. However, this undesirable interference effect can be overcome fairly easily with proper fin configurations. For example, a pair of fins can be placed fore and aft of the l.c.g., producing only small net yaw moments, or using one pair of fins near the l.c.g.. In the case of rudder induced roll motion, it can be said that the first major reference was published in 1970 by Taggart [22]. In this paper, excessive ship rolling was reported to have been caused by the rudder, which was under auto-matic steering control. This type of rudder induced roll motion was first noticed in a high speed cargo ship, which had a large rudder, powerful steering machinery and an automatic steering control system. Similar ship types are susceptible to this undesirable steering behaviour. However, the importance of this work lies in the fact that it registered the potential use of the rudder as an anti-rolling device, which initiated the research and development of RRS.

In the early to mid-seventies, the potential use of the rudder for roll stabilisation was mainly explored by the British researchers, for instance, Cowley [9,10,11], Carley [8] and Lloyd [17]. Two major factors were recognised for RRS to be a viable stabilisation scheme. Firstly, the side force generated at the rudder acts below the centre of gravity of the vessel which provides a moment arm aganist rolling; and secondly, the natural roll frequency is at a frequency much higher than the frequency at which yaw oscillations are dominant, which allows an effective decoupling of the two motions in designing the controller. In these early studies, model testings, numerical studies and sea-trials were conducted to assess the feasibility of RRS. From the work by Cowley and co-workers, initial studies carried out with model tests and numerical models suggested that RRS was quite effective but was not as good as tank stabilisers. Compensators were needed in the control circuit to improve system performance, and roll rate feedback was recommended. On the whole, together with the experience gained from the sea-trials, it was suggested that RRS was good for ships with small GM, low roll damping and relatively long roll period. Hence, RRS seemed to be of limited application and interest.

However, it was the prediction of the destabilising behaviour of RRS on ship motions that had temporarily dampened the development of RRS. In Carley [8], a detailed study on the behaviour of the control characteristics of RRS was carried out using classical control theories, in which the transfer functions for the roll and yaw dynamics of a frigate were derived from sea-trials. In this work, the interactions between the course-keeping and sea-keeping functions of RRS were analyzed. It was found that there were strong cross-coupling effects between roll and yaw control using the rudder. This was particularly pronounced when the excitation frequency was about 0.02 Hz, where roll motions was excited due to the yaw response of the vessel. Basically RRS was only effective in suppressing roll motions around the roll natural frequency. At frequencies either above or below this frequency, the RRS system tended to amplify roll motion! Worse still, manual steering would also experience difficulties. The destabilising behaviour of RRS was also predicted by Lloyd [17], in which the numerical study was based on manoeuvring equations. This low frequency of encounter corresponded to a ship travelling at relatively high speed in quartering seas, which could increase the chances of the broach-to instability.

Despite the unfavourable findings of these works for RRS, it has nevertheless laid the groundwork for subsequent research into RRS. So far, the main concern has been with cargo ships and naval vessels. In Van Gunsteren [31], based on some full-scale experiments, it was suggested that RRS for small craft could be a promising alternative for reducing roll.

In the mid- to late seventies, research on RRS has mainly been carried out as part of a global program that examined the integrated control system of a ship. This approach was first advocated by Carley [7], in which the cross-coupling effects between yaw and roll motions were discussed and an integrated control system strategy to ship motion control was recommended. Broome [5,6] carried on along similar lines in examining the yaw-roll interactions of a merchant ship. It was suggested that, for the particular ship studied, the rudder would have about 16% of the effect of the fin stabilisers in influencing ship roll and that considerable improvements can be obtained if ship control systems are designed to reduce the interaction effects. The numerical model studied by Eda [12] has also high-lighted the strong coupling interactions between yaw, roll and the rudder in high speed operations. Using modern control theories, the work by Whyte [32,33] had in some ways injected new ideas and interests in RRS by re-iterating the feasibility of RRS. In these studies, various feedback signals to the fins, rudder and the combination of fins and rudder were examined based on optimal and sub-optimal design.

An optimal design is one which all the state vectors in the state space model of the system are used in the feeback loop, whilst sub-optimal design refers to the case whereby only some of the state vectors are used for feedback. It was found that, in general, roll rate was the most important feedback parameter.

The renewed interest in RRS over the last decade and the subsequent successful installation of RRS system on-board ship could be said to have been sparked off by the work published in Baitis [1]. This work was the culmination of five year's research and development into RRS system for naval ships. The feasibility of RRS as an anti-roll device was successfully demonstrated in sea-trials with roll reduction up to 50% of r.m.s. motions. Roll rate feedback was the best compromise for simple control, when adaptive controllers were not available. It should be pointed out that the ship speed for the sea-trials was about 15 knots, which was far from the speed at which destabilising effects were suggested in Lloyd [17]. Following the recommendation in Whyte [33], Schmitke [21] performed some numerical studies on RRS using a ship motion computer program based on strip theories. Comparisons were made with an active fin system. Despite the better performance of the fin system in terms of roll motion reduction, it was suggested that this performance could be matched by up-grading the rudder actuator dynamics. The low cost in up-grading the rudder system would make RRS an attractive option. One important aspect in this work was the use of a band-pass filter, which effectively suppressed all the frequencies away from the roll natural frequency band. This may not be a easy task in practice as filters would incur phase-shifts in the system and also attenuation of the input signals.

At this stage of the RRS development, new impetus from the Swedish and Dutch has carried the concept of RRS back to the fore, and eventually brought to its practical realisation in the late 1980s. The Swedish effort can be summarised in the papers by Kallström and co-workers [13,14,15]. From their work, it was suggested that the minimum ship speed should be at least 10 knots and a rudder rate of 4 °/sec if RRS were to be effective. In general, 40 to 60% reduction in roll could be expected. The control system made use of a digital adaptive controller, which automatically optimises the demand signal for the specific sea condition or operation. This system seemed to have overcome the destabilising effect of RRS as demonstrated in sea-trials using a 35m fast-craft running at 35 knots in quartering seas. From eight ship years' operation of the RRS system, good performance has been reported across the range of ship types. The roll rate and roll acceleration signals have been used for feedback.

The Dutch work has been fairly well-documented in a series of papers by Van Amerogen et al [24-29] and Van der Klugt [30]. In their approach, model tests, numerical models and sea-trials were performed during the course of the development. The main problem discussed was rate saturation, which was due to the fairly sluggish rudder servo dynamics. It was suggested that a rudder rate of 15 °/sec would be required for RRS systems. During some sea-trials, the destabilising effects due to high speed operations in quartering seas were encountered, which high-lighted the limitation of the controller. This brought about the design of the adaptive gain control, which overcome the rate-saturation problem as well as providing optimal control gains.

The Danish installation was reported in Blanke [4], in which a new naval vessel was designed with three rudders for RRS purpose. The centre rudder was solely for course-keeping, whilst the port and starboard rudders served dual purposes - steering and roll stabilisation. From the sea-trials, it was shown that roll reduction of 35%-40% in quartering seas, 95% in beam seas and 50%-60% in bow seas were obtained. The roll-rate and roll angle were used for the feedback control. In this approach, it seems that the conventional fin system is merely replaced by two rudders with a more complicated control strategy.

Despite their early success with the prototype trials, the American team is still exploring the RRS system for different ship classes (see Baitis et al [2,3]). From their operational experience with the proto-type, it was found that rudder rate-saturation was a serious problem. A digital controller was used to up-grade the control system in order that optimal control could be achieved. A band-pass filter for roll-rate feedback was also experimented, but the phase-shift in the signal has made it an unworkable option. Instead, a high-pass filter was used to eliminate the low frequency interference to steering, and phase advance was also needed to copensate for the additional phase-lag incurred. For good roll reduction performance, it was suggested that a non-dimensional damping value between 0.35 to 0.5 would be desirable.

In a recent paper by Powell [18], it was concluded that the RRS approach is not advisable for the British Navy as the performance of RRS using existing steering equipment does not compare favourably with the active fins systems in service. This negative view seemed to have been derived mainly from the low rudder rate of 3 °/sec used for the study. However, the work by Roberts[20], in which the main interest was to study the yaw-roll interactions may bring forward a more favourable view of RRS or at least rudder assisted roll stabilisation to the RN.

Although successful RRS installation has been reported, research on RRS controllers is still quite active, for example Katebi [16] and Zhou [34]. In Katebi [16], different types of feedback control were explored using modern control theories. It was founded that roll-rate plus roll angle has limited advantages, roll angle and roll acceleration feedback interfere with steering, but that roll-rate alone was preferable. In Zhou [34], non-linear roll-damping could be taken into account, which made use of modern control theories and a technique called linear recursive prediction error. With continual research and refinements, the second generation RRS systems could be improved still further.

3.0 Numerical study and Discussion

It can be said that the main aim of LFE stabilisation is to reduce lateral accelerations on-board ship in order that crew members can perform their task more readily in supporting helicopter operations on the flight deck. This means that the demand for LFE stabilisation is basically of a short-duty cycle nature. Therefore, in the first instance, this study should concentrate on applying the existing rudder equipment before exploring other possibilities, such as requirements on rudder performance for effective RLS. As is evident from the review that, quite a combination of feedback signal have been suggested and used for RRS, namely roll-rate, roll-rate plus roll angle, roll rate plus roll acceleration and even sway velocity. However, by far the most often recommended feedback was roll-rate,.which is simple and easy to obtain. Hence simple roll-rate and LFE-rate feedback to the rudder will be studied in some detail.

The low rudder rate has posed some problems to the effective use of the rudder for stabilisation purpose, as phase-lag would set in at high frequency, altering the phase of the stabilising moment in such a way that motion amplitudes would actually increase. To avoid rate-saturation, which is a non-linear problem, the rudder-rate limit, which is quoted as 6°/sec in Roberts [20], should not be exceeded. This corresponds to a r.m.s. value of 2.8°/sec for 10% exceedance. Furthermore, according to Amerogen et al[26], the rudder-rate would impose a limit on the maximum rudder angle possible, which is related to the roll natural frequency and rudder rate of the vessel. In the present study, this rudder angle limit would be about 10°, which corresponds to a r.m.s. value of about 4.5°. From the simulations, in most cases when the rate limit was staisfied, this amplitude limit was also satisfied.

3.1 Forced rolling

Unlike the fin forced roll option within the sea-keeping program, which has been reported to give reasonable comparisons to sea-trial data, the rudder forced roll predictions have not been given much attention. Therefore, the first task would be to establish some confidence in the computer program predictions with the rudder. Fig.1 shows the rudder forced roll response of two frigates. The sea-trial data are for Newship, which was used in the last study (Tang [23]), whilst Nk denotes the vessel for the LFE trials. It can be seen that the computer model agrees quite well to the measured data. The good comparison is somewhat fortuitous as the wake and the effect of the propeller are not modelled. However, from a ship dynamic view-point, it would not be unreasonable to make a comparative study based on this model, especially when low frequency yaw-roll interaction is so well predicted. As for the ship Nk, the roll response spectrum has a wider band near the resonant frequency, which suggests that this vessel would be susceptible to a wider range of wave excitation frequency. The yaw responses are given in fig.2, which are typical of the sea-trial data from other classes of frigates. There are only small differences between the two ships and the steering characteristics of the two ships should be very similar. Fin forced roll was also performed in order to have a feel for the effectiveness of the rudder. It can be seen in fig.3 that for Newship, rudder is as effective as the fin in forced roll, whilst for Nk the rudder is in fact more powerful. Therefore, if additional roll stabilisation is needed in Nk, the rudder should be considered.

3.2 High speed destabilisation

It was pointed out in Carley [8] and LLoyd [17] that roll amplification would occur at low frequency of encounter with RRS. In order to check this effect, the computer model was run at 15 and 30 kts at various wave angles with a roll rate feedback of 5 (compare to traditional 2.5.2 control in fins). The results for Newship are given in fig.4-6. It is apparent form fig. 4. and 5 that while good reduction both in roll and LFE were obtained at 15 knots, motion amplification did occur in the quartering seas region when the ship is running at 30 kts. This corresponded to the 0.02 Hz predicted in Carley [8] and Lloyd[17]. From fig.6, the rudder-rate response did not show any abnormal behaviour in the quartering seas region and in fact the activities were relatively low. Generally speaking, roll and LFE reduction was about 30 to 40%. A similar simulation was performed with Nk (fig.7-9). The results shown similar trends to those with Newship, in which instability was also encountered at low encounter frequency. However, the rudder activities were higher in achieving a similar level of motion stabilisation to Newship, which indicated that the performance of the rudder system in Nk could be improved.

3.3 Tuning for RLS

Following the tuning procedure given in Tang [23], the rudder system was tuned for LFE stabilisation. It has been suggested in Lloyd [17] that the roll angle gain should be zero. However, it was found that in general, the roll angle gain derived form the tuning procedure is fairly small near the roll natural frequency. Also, if the k1 term was set to zero, negative gain term for k2/k3 would result. Furthermore, some preliminary comparisons with the two cases did not show any major differences when the motions near the natural roll frequency was examined. Because of these factors, the three term controller has been used. As the last LFE stabilisation was based on Newship, coupled with the better performance of its rudder, it was decided to use Newship again in this part of the investigation.

It can be seen in fig.10 that, unlike tuning for the active fins where a minimum response was found at certain tuning frequencies, the resulting LFE seemed to lessen with decreasing tuning frequency. This should not be realisable in practice as at low frequency, the yaw induced roll interference would put a lower limit to the tuning frequency. In order to select the *best* tuning frequency, it would be easier to make a comparison of the different sets of tuning in a sea-way. In this comparison, the ship speed was set at 20 kts. The gain levels were selected by trial and error till the rudder rate at beam seas would be just below the rate limit of 2.8°/sec. This was a rather time-consuming process and therefore in some cases, as can be seen in fig.13, the maximum demanded rate is slightly above the limit. The resulting LFE and roll responses with different tuning are given in fig.11 and fig.12, where case B denotes the lowest tuning frequecy whilst case G denotes the highest tuning frequency. As evident from the responses variation among different tuning, it has followed a similar trend to the tuning curve in fig.10 i.e. more reduction can be achieved at low tuning frequency. Slight destabilising effects are shown in the high frequency tuning cases. Therefore one may conclude that the lowest tuning frequency should be used. However, as suggested earlier that the lower tuning frequency would interfere with the yaw motion, it should be possible to show this from the simulations. One way to assess this would be to look at the response spectrum value for yaw response at the lowest frequency component. If yaw motion is affected, the spectral value would increased. A plot of this yaw response value for different tuning are given in fig.14. It can be seen that, indeed, the lowest tuning frequency has the highest yaw response, especially in the quartering sea region which has a value almost four times the unstabilised case. Therefore, in deciding which tuning frequency is best, the yaw response in RLS should also be taken into account. Judging from fig.11,12 and 13, case C would be a better compromise.

To check for destabilisation at high speed, the ship speed chosen was 30 kts, and the same procedure to the 20 kts case was followed. In fig.15, the destabilising effect of RLS in roll is strongest with low tuning frequency at quartering seas, but the margin is not excessive when compared to the RRS case in fig.5. Also in fig.16, the LFE response shows quite favourable reduction for the low tuning frequency cases. Furthermore, from the rudder rates in fig.18, the demand shows fairly acceptable variation for the low frequency tuning. The yaw response at low frequency has increased compared to the 20 kts, but the trends are very similar. From the rudder rate response in both speeds, it is quite clear that highest rudder demands are in beam seas. However the worst roll and LFE response are at quartering seas. So if the gain can be increased in this region, the RLS can be made more effective. By the same token, if destabilisation is sensed, a reduction in gain should alleviate the situation. Hence an adaptive controller is desirable if RLS were to perform more effectively. This is not dissimilar to the automatic gain control of the Dutch RRS system.

3.4 RRS and RLS comparisons

Having obtained reasonable performance from RLS, the next step would be to explore the rate feedback option to see how these RLS alternatives compared with the conventional RRS approach.

A ship speed of 20 kts was used in these simulations and the rudder rate was adjusted as in the last section. Fig.19 and 20 show that roll rate feedback in RRS performs far better than RLS, and that the three term feedback is better than rate feedback in RLS, both in terms of LFE and roll reduction. The rudder rate (fig.22) in RRS shows a fairly broad response, counter-acting roll in most headings as compared to a very high demand at beam seas for the RLS system, which has little influence at other headings. The yaw response in fig.21 also suggests better performance by RRS. A comparison of the response spectrum from RLS and RRS may reveal the reason for the difference in performance. In fig.23, the roll and rudder spectrum for the two strategies in beam seas are shown, where the l denotes RLS and the r denotes RRS. It is obvious that the rudder in the RRS case counteract the roll motion far better especially near the roll natural frequency. Near this frequency, the rudder activities in RRS is about five times the corresponding RLS system value, whilst the rudder in RLS shows a relatively high response at high frequency, which does little to suppress the dominant rolling near the natural period. This high rudder response at high frequency is almost certainly due to the sway term in the LFE feedback.

The ship speed was increased to 30 kts in order to compare the severity of motion destabilisation in the different cases. From fig.24 and fig.25, destabilisation in RRS is more severe in terms of magnitude, while in the RLS systems, it tends to affect a larger range of headings, i.e. frequency of encounter, but with lower destabilisation level. The yaw response in fig.27 shows similar features to the 20 kts case in fig.21. The response levels are 30%-70% higher at this speed. The rudder rate in RRS has decreased with increasing forward speed, whilst in the RLS system, rudder rates show a marked increase in activities near the bow seas region. Therefore, different feedback controls exhibit different speed dependencies.

In a conventional active fin system, there is an automatic speed dependent gain to optimise fin operation. This gain generally reduces with increasing speed. In the case of the rudder, a similar speed dependent gain should apply. Different feedback gain levels were then applied to the RRS system to ascertain the effect of gain level on the motion. The results are given in fig.28-30, where the '+' sign denotes twice the gain level i.e. 10 and the '-' sign denotes half the gain. Increasing the gain, causes motion destabilisation to become more severe, whilst a reduction in the gain alleviates the situation. However, the destabilisation is still present. The rudder rate in fig.30 shows that the response does not vary linearly with the gain.

Conclusions

Some experience on motion stabilisation using the rudder has been gained using the sea-keeping program. The two stabilisation strategies, namely rudder roll stabilisation RRS and rudder LFE stabilisation RLS, have been investigated and comparisons have been made to assess the two approaches. Based on the simulations, the following conclusions are made:

(i) The rudder forced roll option in the sea-keeping program gives reasonable roll motion prediction. The destabilisation effects of RRS at low encounter frequency has also been predicted, which is consistent with results from earlier works. This has further increased the credibility of the sea-keeping program for rudder application.

(ii) On the whole the RRS approach performs better than RLS, in terms of motion reduction, rudder response and the likelihood of steering interference. However, motion destabilisation at high speed in the RRS system is more severe compared to RLS.

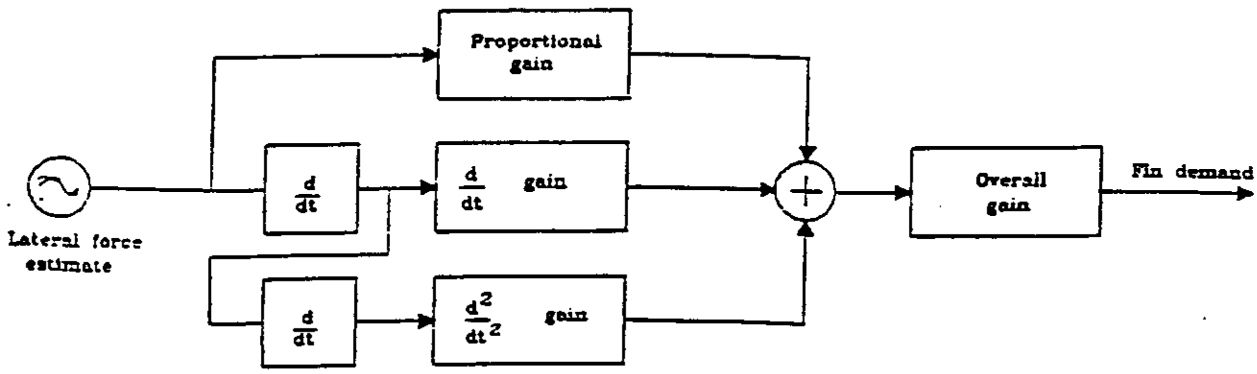
(iii) The tuning procedure that has been adapted from the active fin system gives reasonable feedback gains for motion reduction in the RLS system. The level of motion reduction is better than those derived from simple rate feedback control for the RLS strategy, but the likelihood of steering interference increases.

(iii) Despite the relatively low rudder rate imposed, the RRS system with roll rate feedback gives reasonable motion reduction (30%-40%) at 20 kts ship speed. The destabilisation effect at low frequency encounter could be lessened with the *correct* control of the gain level.

This part of the report briefly describes the design and components of the electronic hard ware that modifies the LFE signals into a suitable form for the fin stabiliser controller supplied by Brown Brothers Ltd. The design criteria and some of the limitations governing the design are discussed.

The fin stabiliser controller supplied by Brown Brother Ltd. is a relatively modern system, which makes use of digital control as compared to the analogue circuits employed in the conventional controllers in the RN. This new design has imposed greater demand on the accurate design of the LFE circuitry as well as the complexity.

According to Brown Brothers Ltd., the LFE signal input will be processed as illustrated in the diagram below:



The LFE signal input to this system has to be in synchro form, which is basically a digital interface signal required for the computer based controller. The analogue signals from the accelerometer therefore not only have to be conditioned in additional circuitries for the interface, but also needed to be converted into digital form. Once the signal is in digital form, another converter is needed to convert this digital signal into the required synchro form. This process is quite complicated compared to conventional analogue designs. An illustration of this signal process is given in fig. 2.1. This signal process will be described in two parts, which deal with the analogue and digital circuits separately.

Analogue Signal

This part deals with the accelerometer, the signal conditioning circuitry and the power supply.

(i) Accelerometer

For the sea-trials, the accelerometer should be robust, of high accuracy and resolution with low noise properties. Above all, it must be of high enough quality for control purposes. The linear accelerometers produced by Schaevitz have been suggested to have the required standard as some of the products have been used in control systems. The accelerometer A223 $\pm 1g$, with a natural frequency of 95Hz from the LSB series has been selected. Apart from the required properties, it can also withstand 100g shock loading, which would be a desirable feature in the sea-trial environment. However, the accelerometer should be protected from sea water.

(ii) signal conditioning

Under the sea-trials conditions, the accelerometer would register signals from a wide spectrum of frequencies. A band pass filter should be used to limit the frequency to the region relevant to the motion control otherwise unnecessary demand on the controller would degrade the stabilizer performance. A high pass filter ($>$ 0.002 Hz) is used to remove the very low frequency components, which are effectively d.c. signals. This is compatible with the controller function which does not compensate for list. A low pass filter ($<$ 10 Hz) is used to remove high frequency signals, which could arise from ship vibrations. From computer simulations, the response from these filters have a linear range between 0.01 Hz to 7 Hz (see fig.2.7). The phase response should also be good so that phase-lag would be minimal. Fig. 2.8 shows good time response. As yaw response is about 0.02 Hz and high frequency motion would be much less than 1Hz, the frequency range of the filters are adequate. A gain control is needed for the final tuning of the signal level for maximum sensitivity but at the same time within the overloading limits. The gain control range is designed between $\pm 0.2g$ and $\pm 2g$ for ± 5 V output, the voltage limit for the analogue to digital converter. This range was selected on the basis that the maximum acceleration level would be 0.5g corresponding to a roll angle of 30°. To prevent overloading from excessive motions, power failure or accidental stray signals, a clipping circuit for ±5V was installed before digital conversion. An overall impression of this analogue signal process can be seen in fig. 2.2.

(iii) power supply

A stabilised and regulated power supply is essential for the safe operation of the digital and analogue circuits of the LFE signal hardware. The power supply unit should prevent spikes and high voltage surges in the main power supply on-board from damaging the electronic hardware. The design of this power supply unit is shown in fig. 2.6.

Digital Signal

This part deals with the analogue to digital converter (ADC), the digital to synchro converter (DSC) and the functions of the necessary hardware required to operate this signal process.

(i) ADC converter and auxiliary circuit

The selection of this piece of electronic hardware is normally quite straight forward. However, due to the high accuracy required by the controller, a minimum of a 14 bit signal is necessary. A 16 bit converter would complicate the auxiliary circuit requirements further, whilst a 12 bit converter would degrade the resolution. The main concern in designing the auxiliary circuit is to remove stray glitches such as narrow spikes within the digital system. To achieve this, the sample and hold is triggered to hold the level of the analogue signal. After a short time delay the A to D is triggered to convert analogue level stored in the sample and hold. As the A to D takes a set time to convert the input to digital outputs, a time of at least twice this value is delayed before triggering the temporary store. This pulse is narrow, so that the possibility of a change in the sample and hold level or false bits from the A to D or any other spikes in the analogue signal, that occurs at the same time as this pulse is very remote. The sample and hold circuit is then released to follow the input signal until triggered again. The cycle time taken is one millisecond. This time should not introduce any significant phase delay into the signal. The temporary store is required before the D to S because the digital outputs from the ADC cannot be guaranteed to remain stable during conversion. A clock pulse is needed to synchronise the sample and hold and the trigger and store operations, described above.

(ii) DSC converter

The DSC converter is a specialist chip which converts the digital signal into an analogue three phase signal for the controller. For synchronisation purpose, a reference source from the ship is required. For detail information of this chip, the manufacturer's manual should be consulted. Fig. 2.5 shows some of the main signal input\output ports. An overall view of the signal process is shown in fig.2.2.

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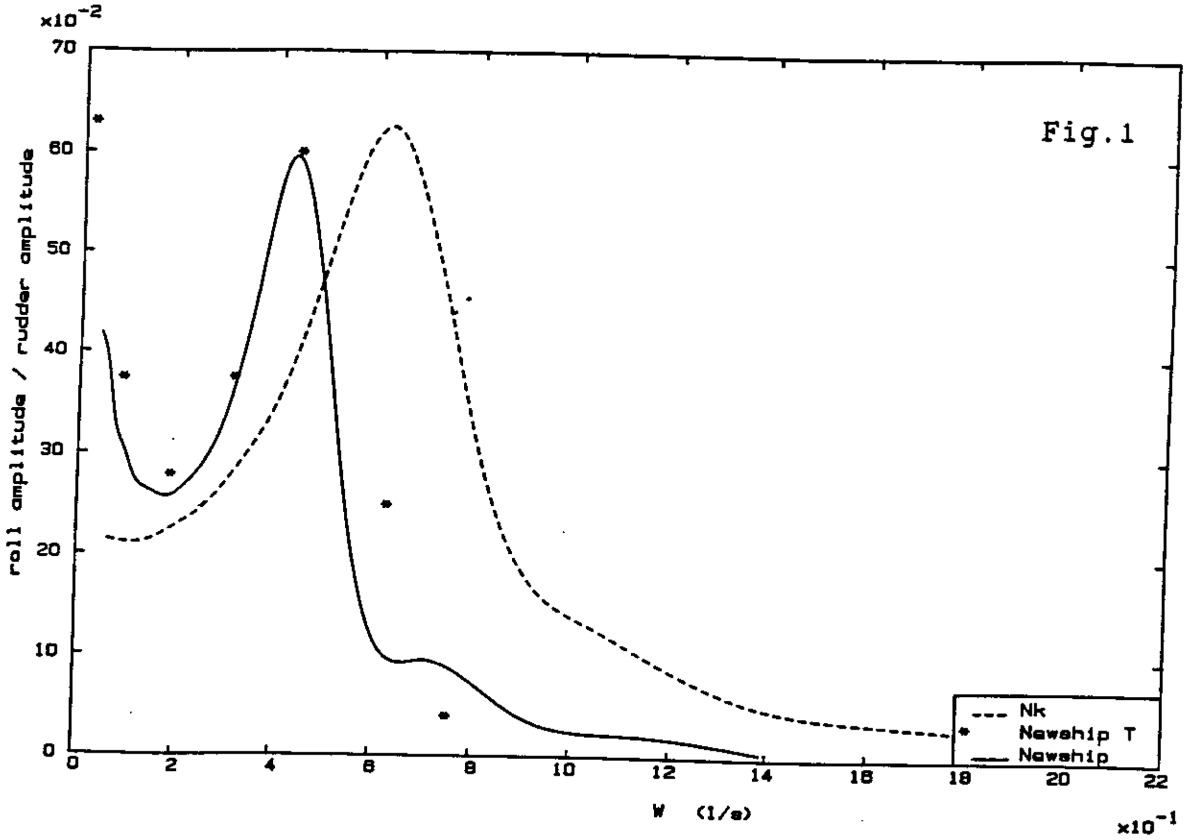
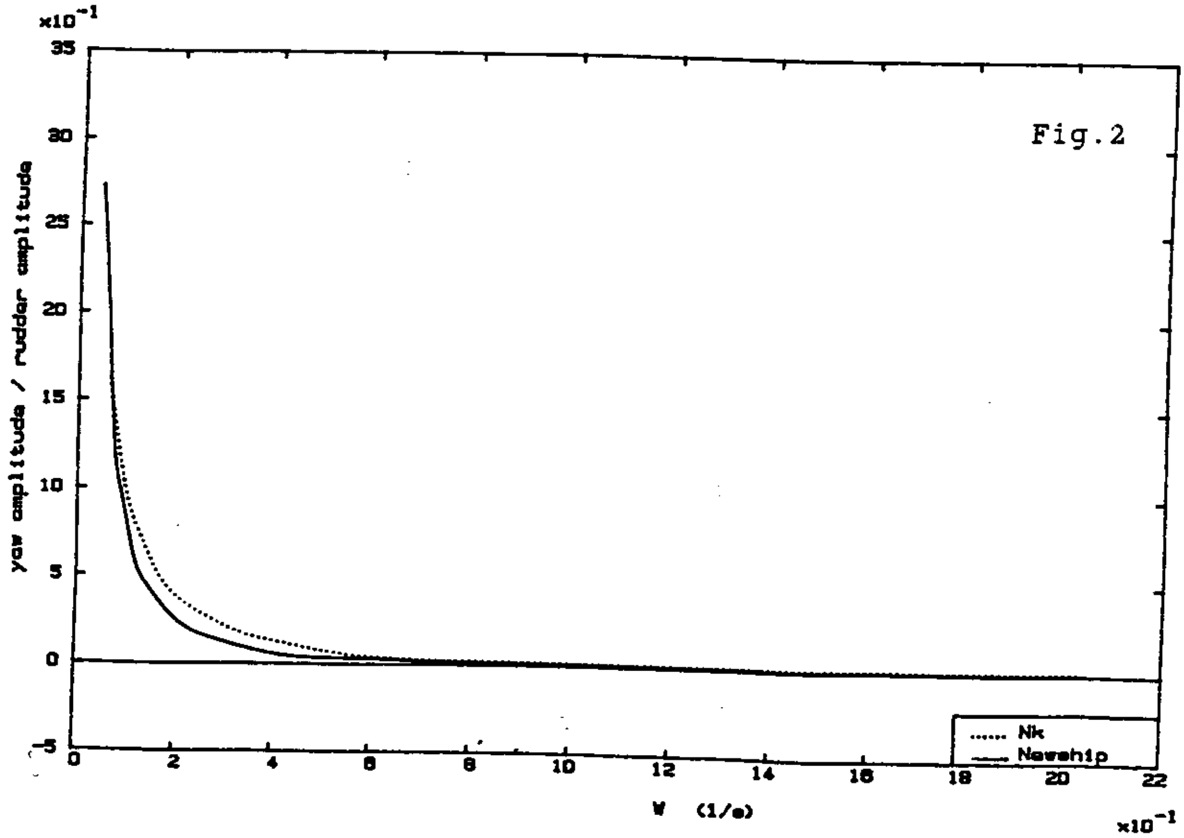
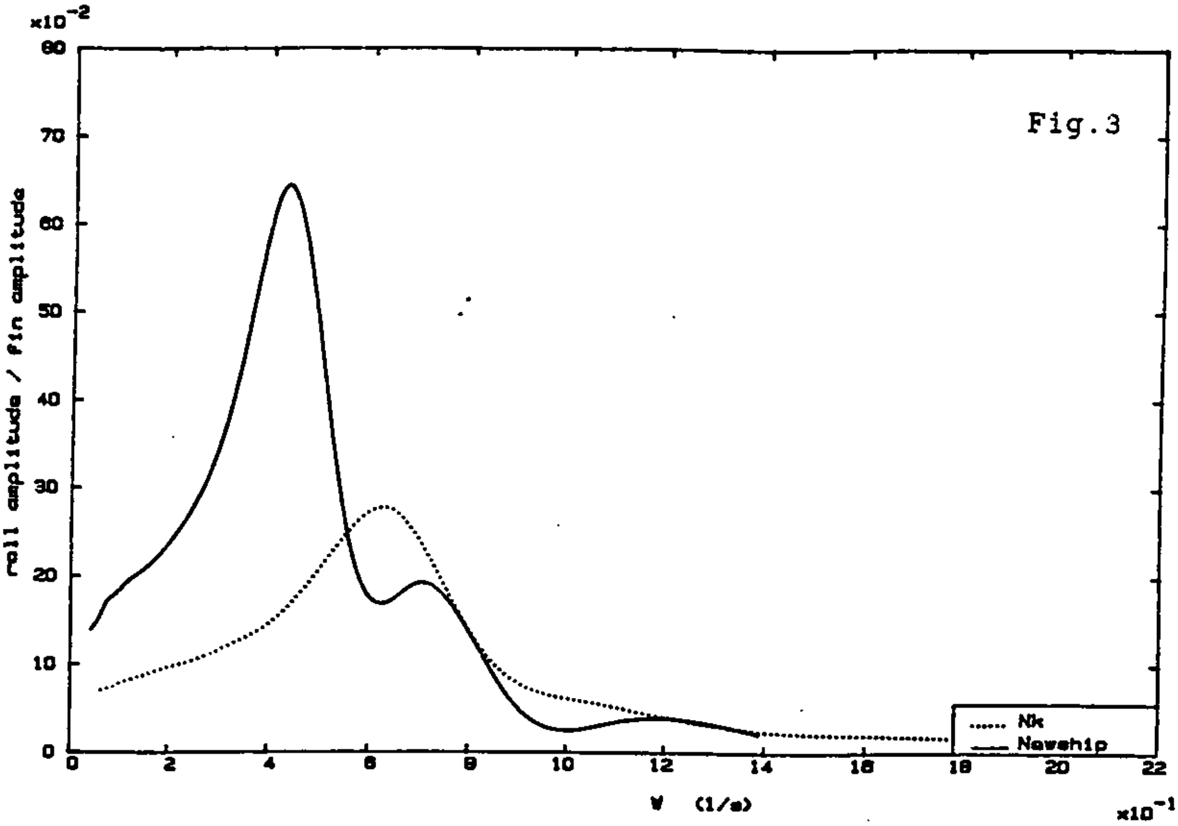


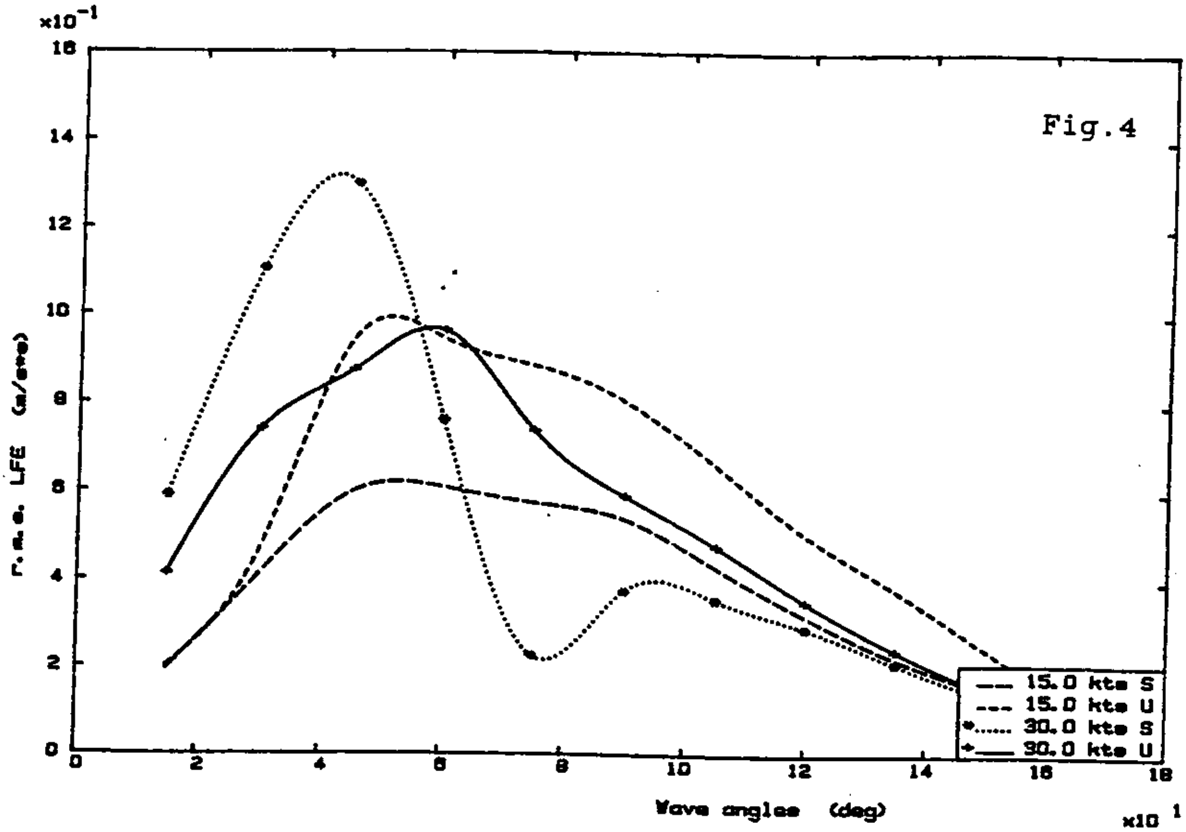
Rudder forced roll response



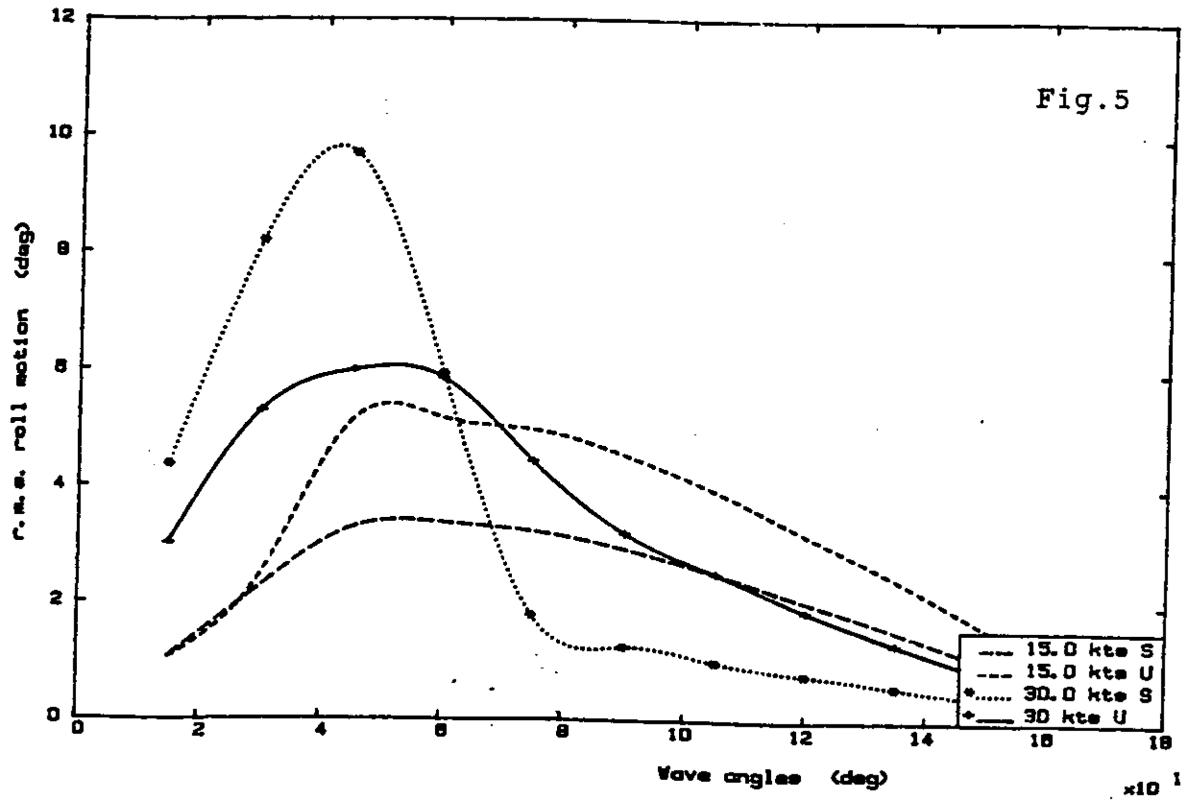
Rudder forced roll yaw response



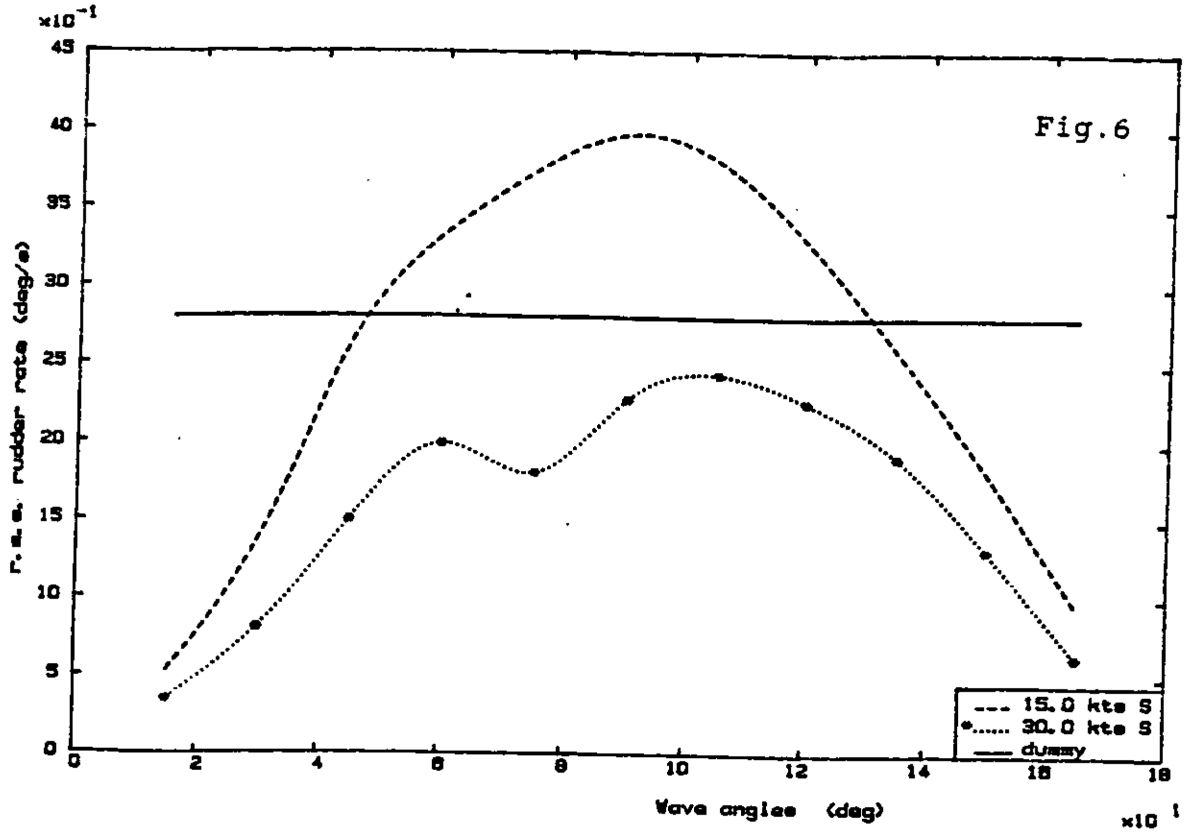
Fin forced roll response 20 kts



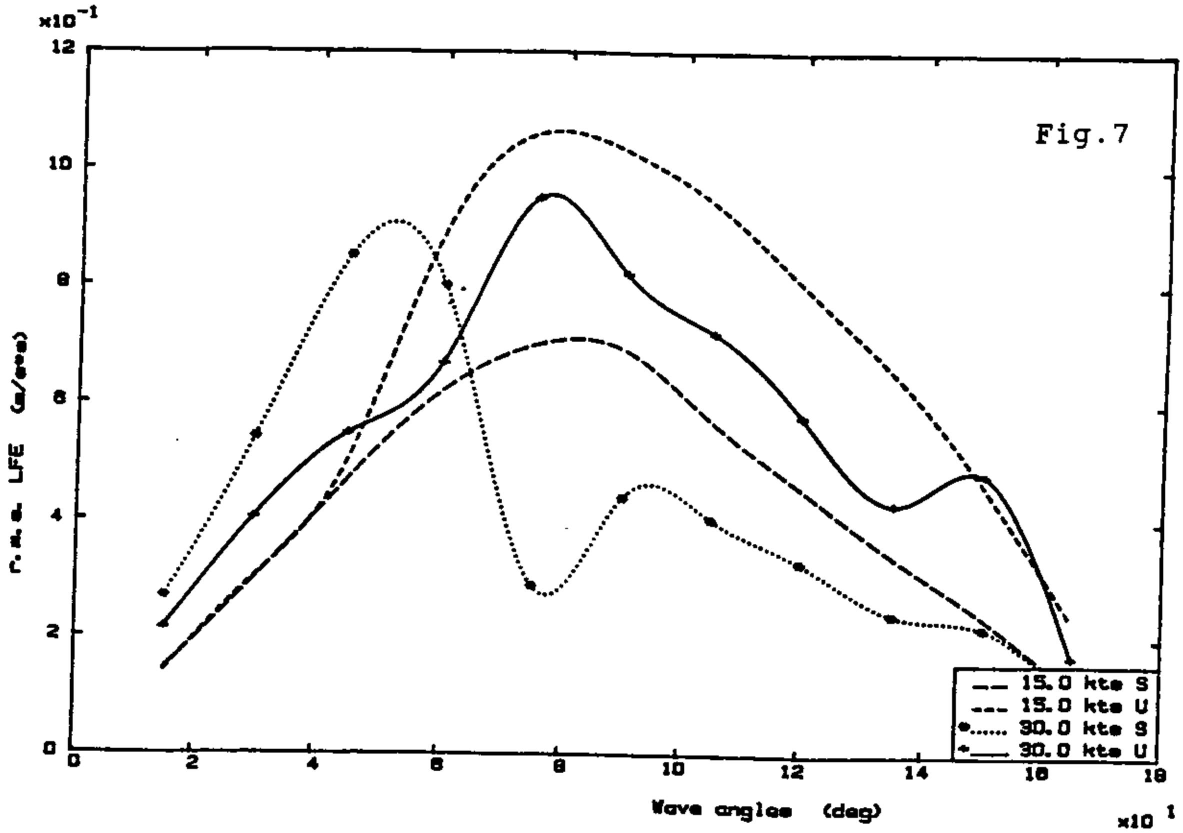
LFE response with rudder stabilisation 5.5m 12.4sec Newship



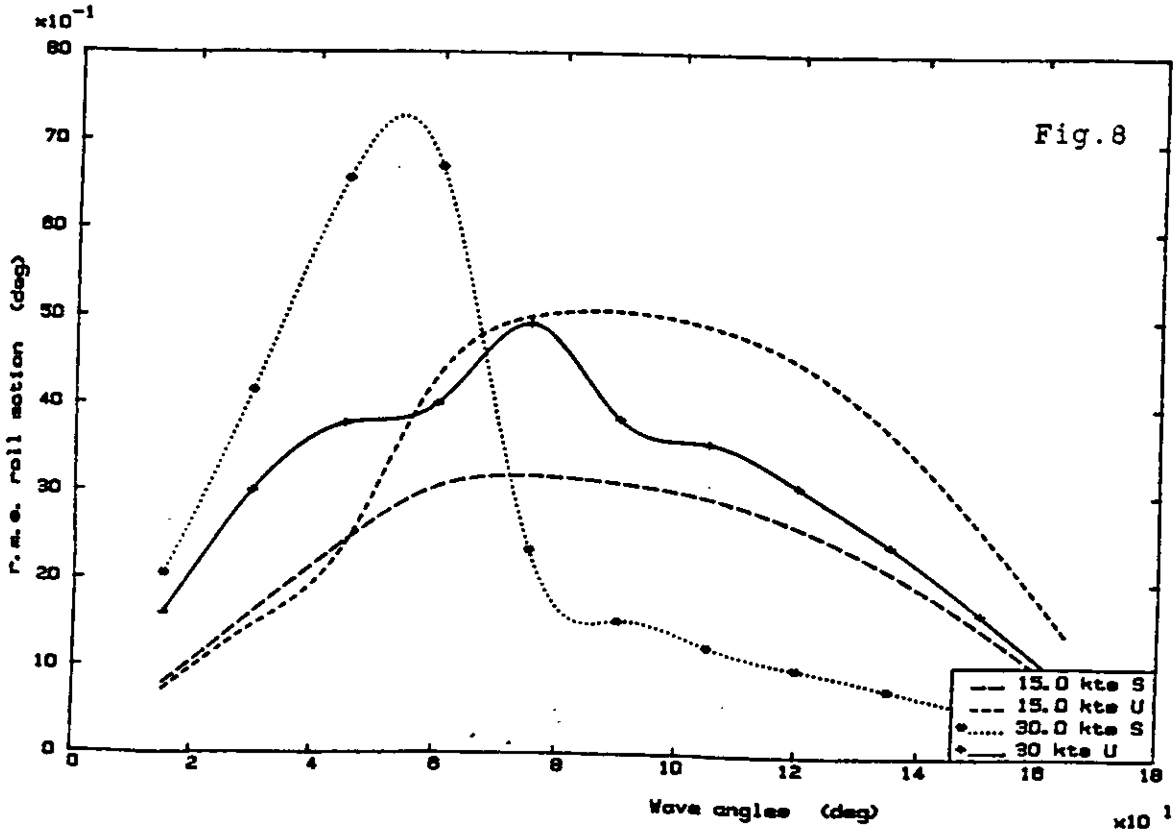
Roll response with rudder stabilisation 5.5m 12.4sec Newship



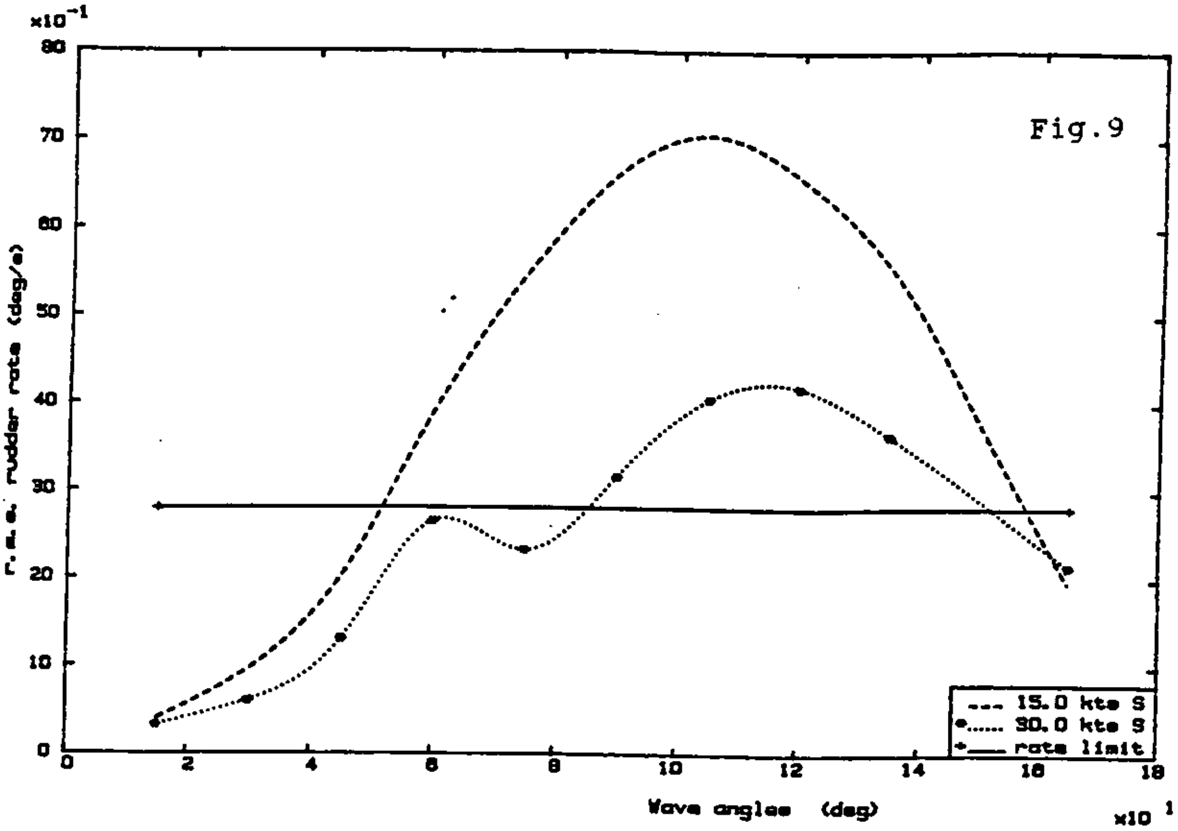
Rudder rate response with rudder stabilisation 5.5m 12.4sec Newship



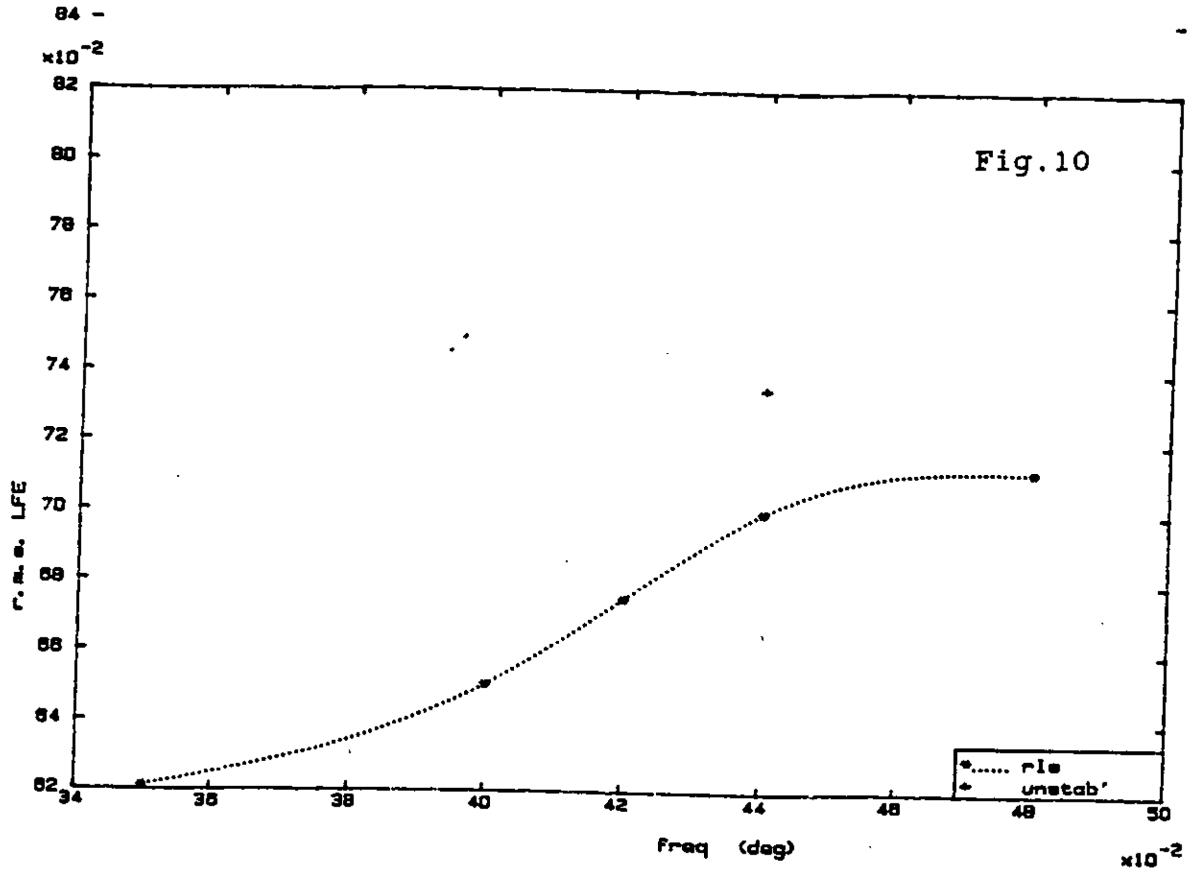
LFE response with rudder stabilisation 5.5m 12.4sec Nk



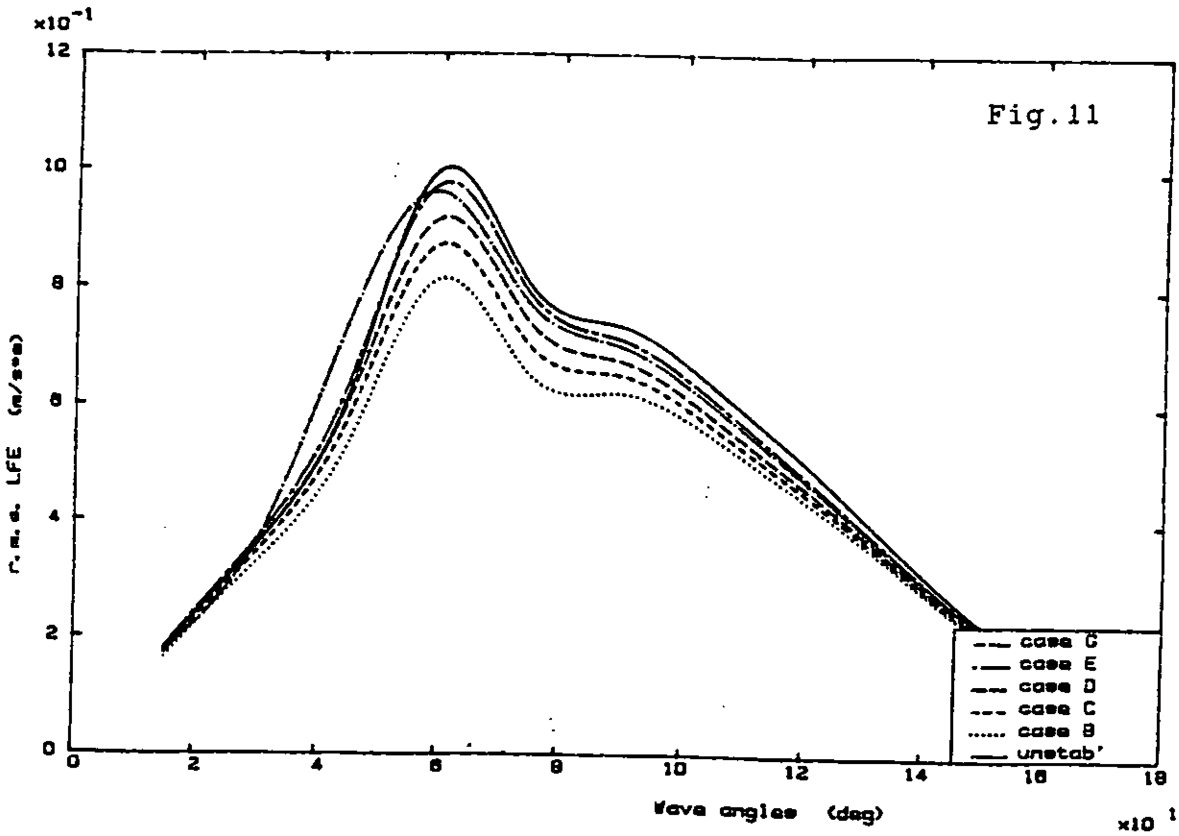
Roll response with rudder stabilisation 5.5m 12.4sec Nk



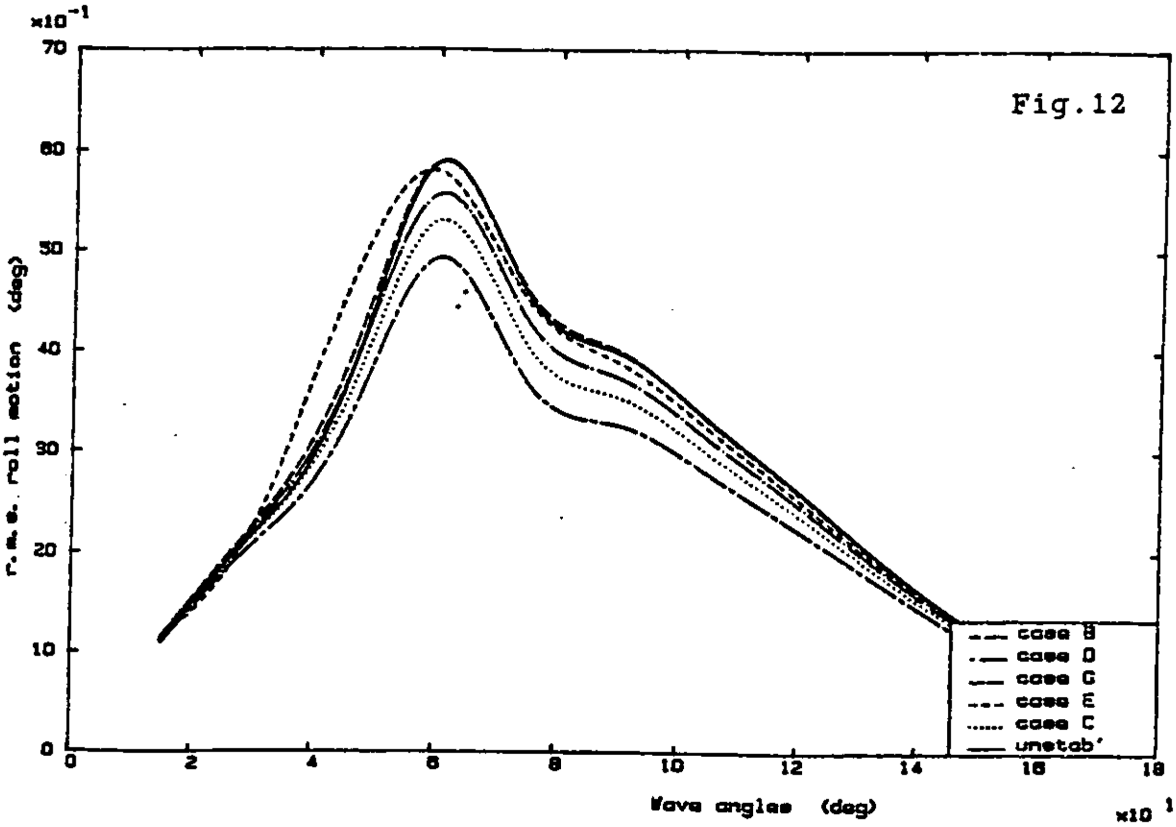
Rudder rate response with rudder stabilisation 5.5m 12.4sec Nk



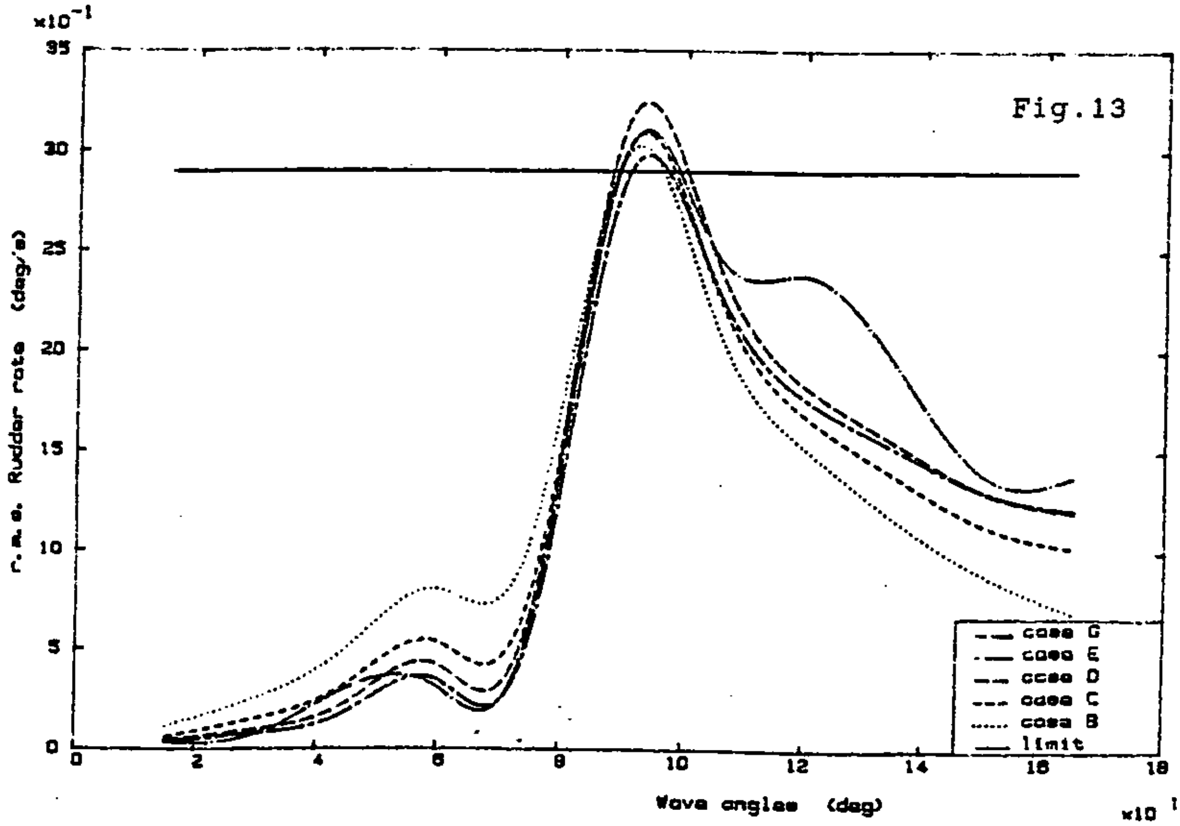
Tuning for RLS stabilisation 5.5m 12.4sec 20 kts Newship



LFE response with RLS stabilisation 5.5m 12.4sec 20 kts Newship



Roll response with RLS stabilisation 5.5m 12.4sec 20 kts Newship



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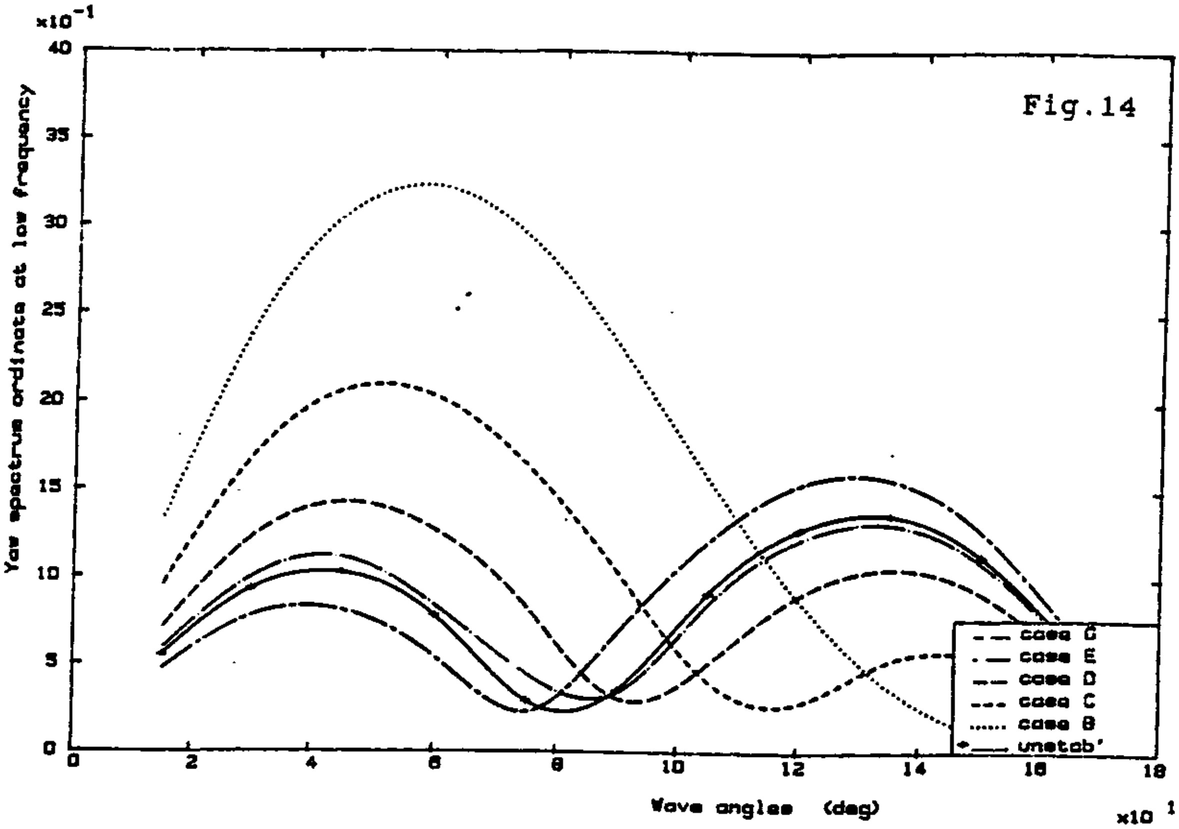

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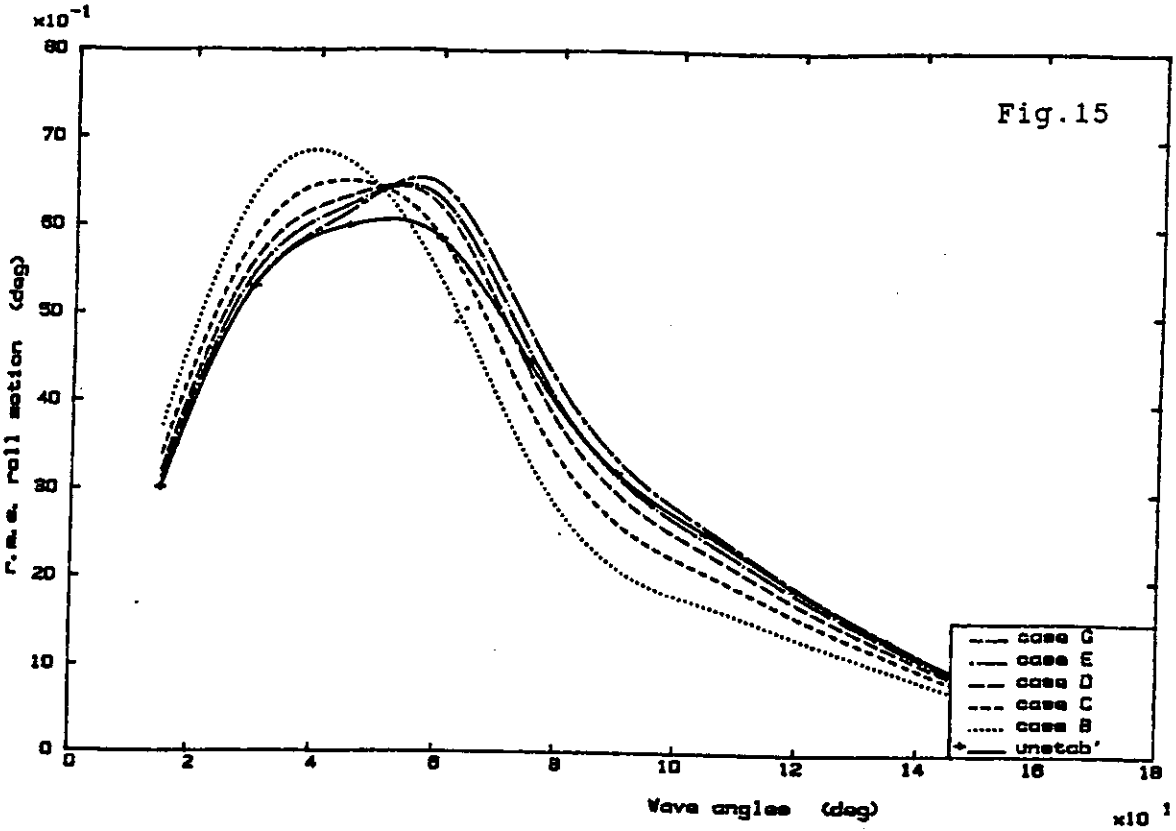

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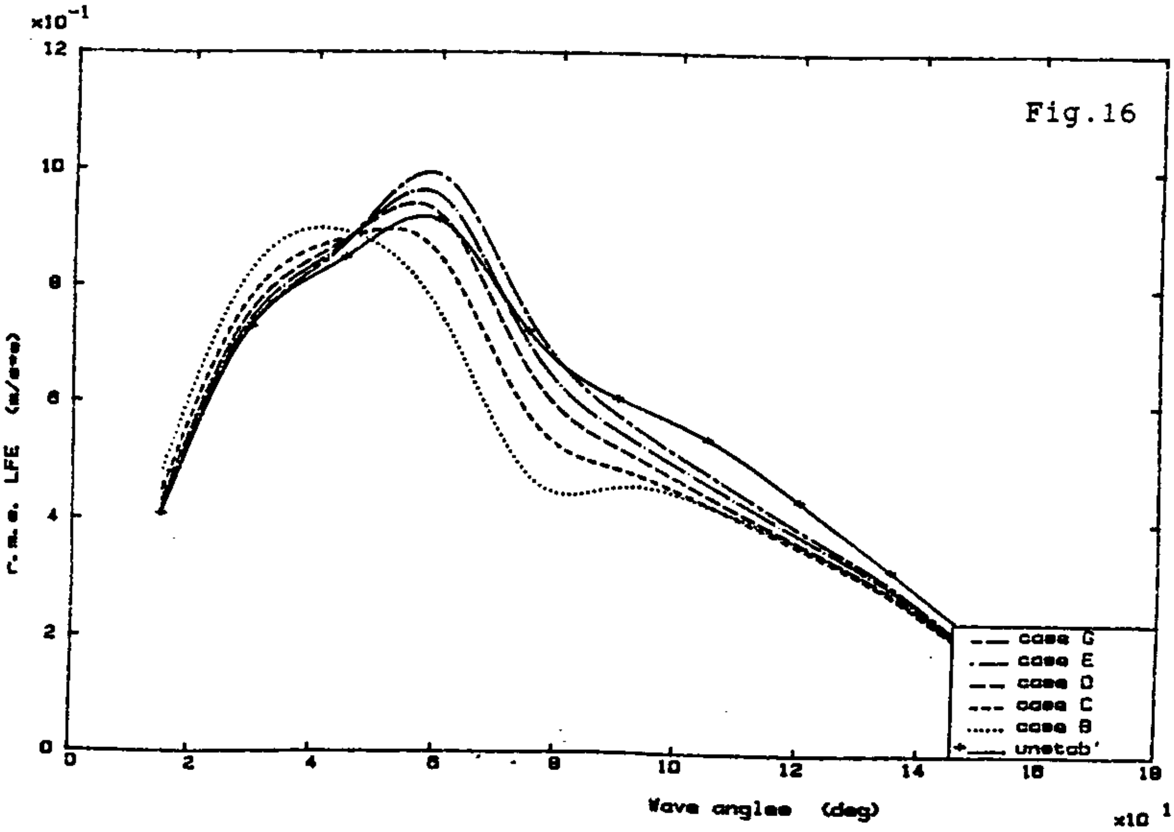
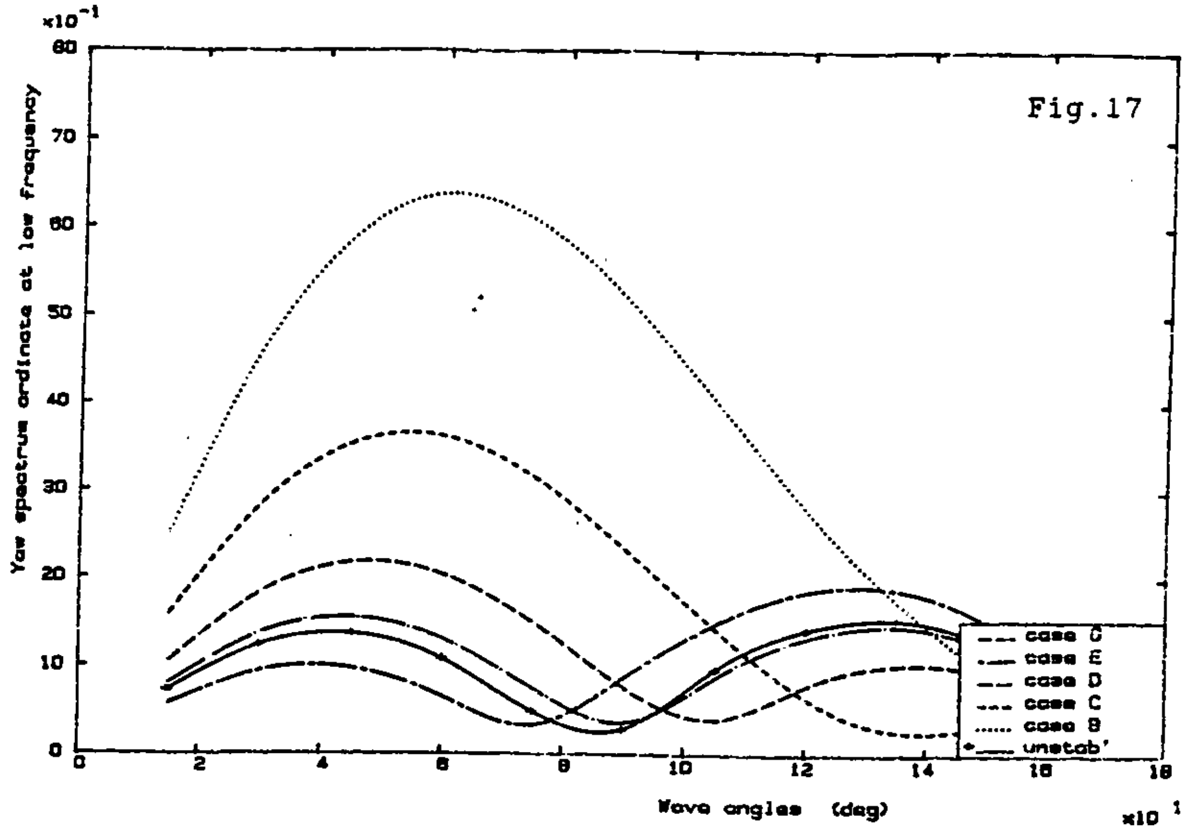


LFE response with RLS stabilisation 5.5m 12.4sec 30 kts Newship



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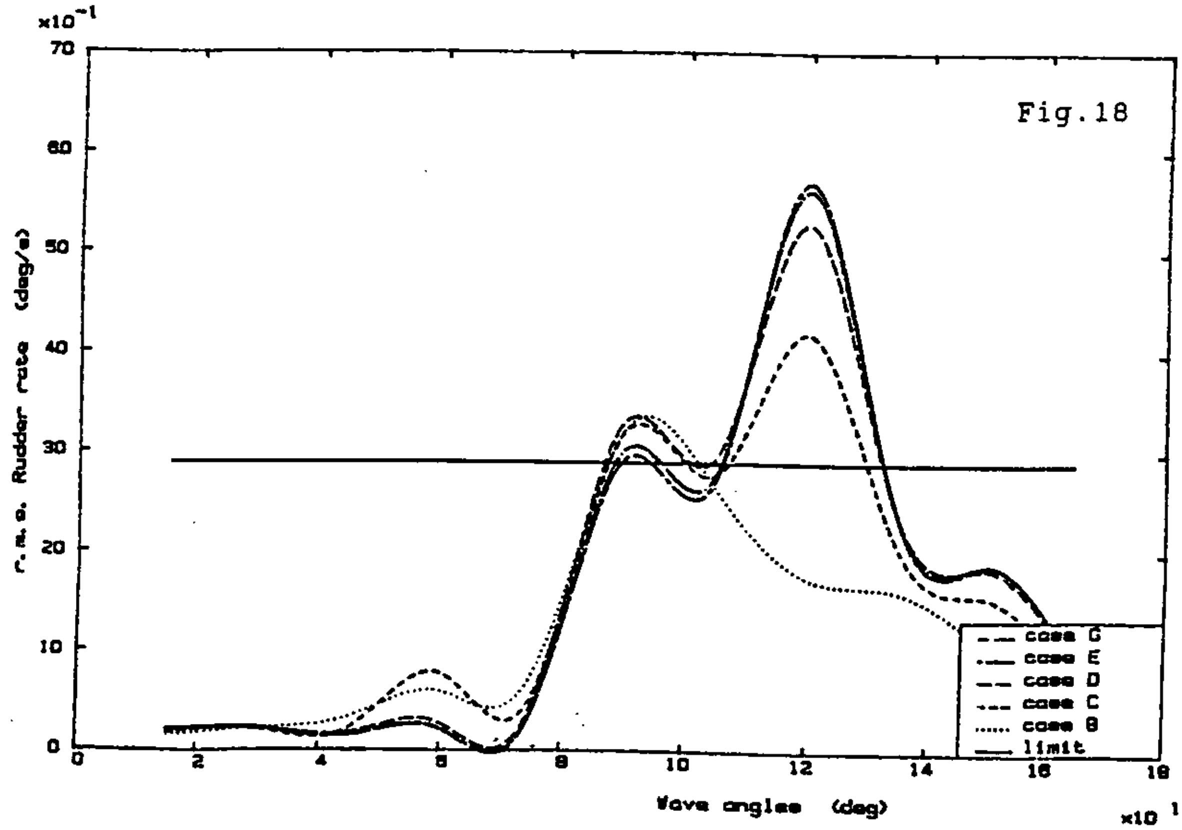

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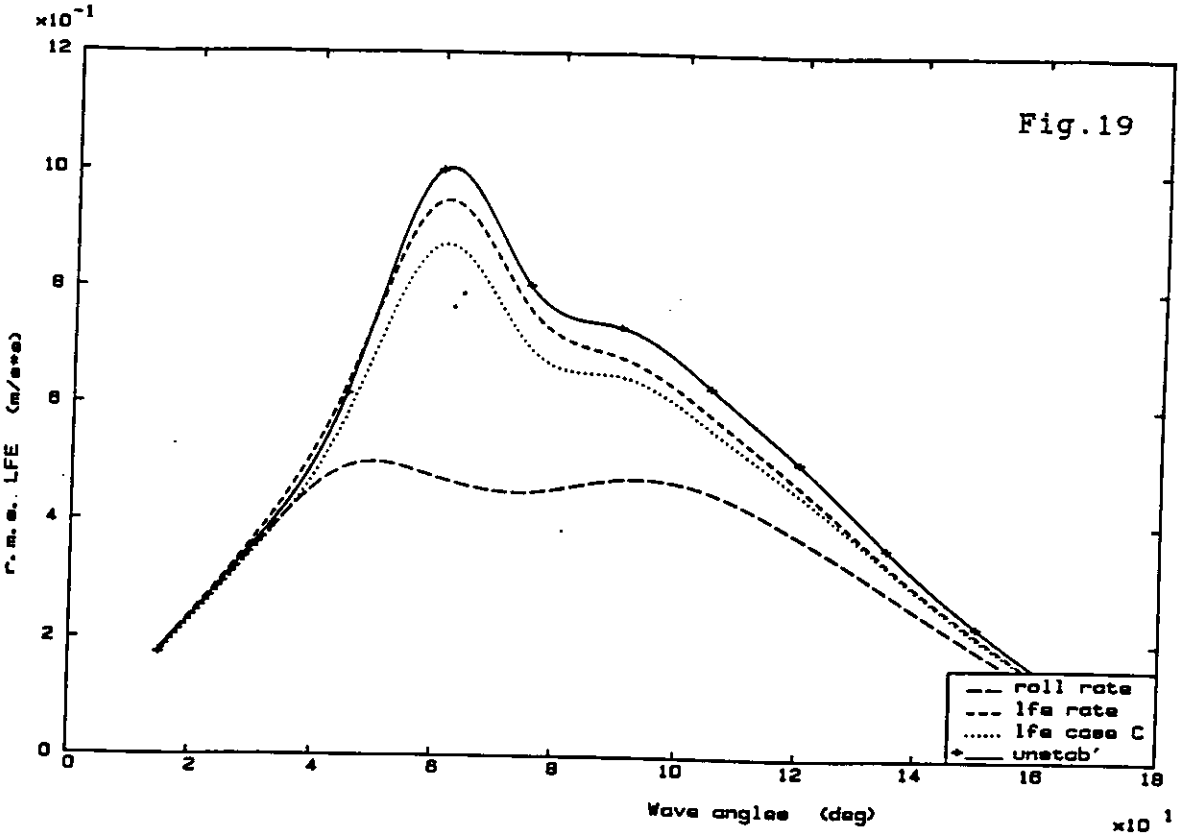

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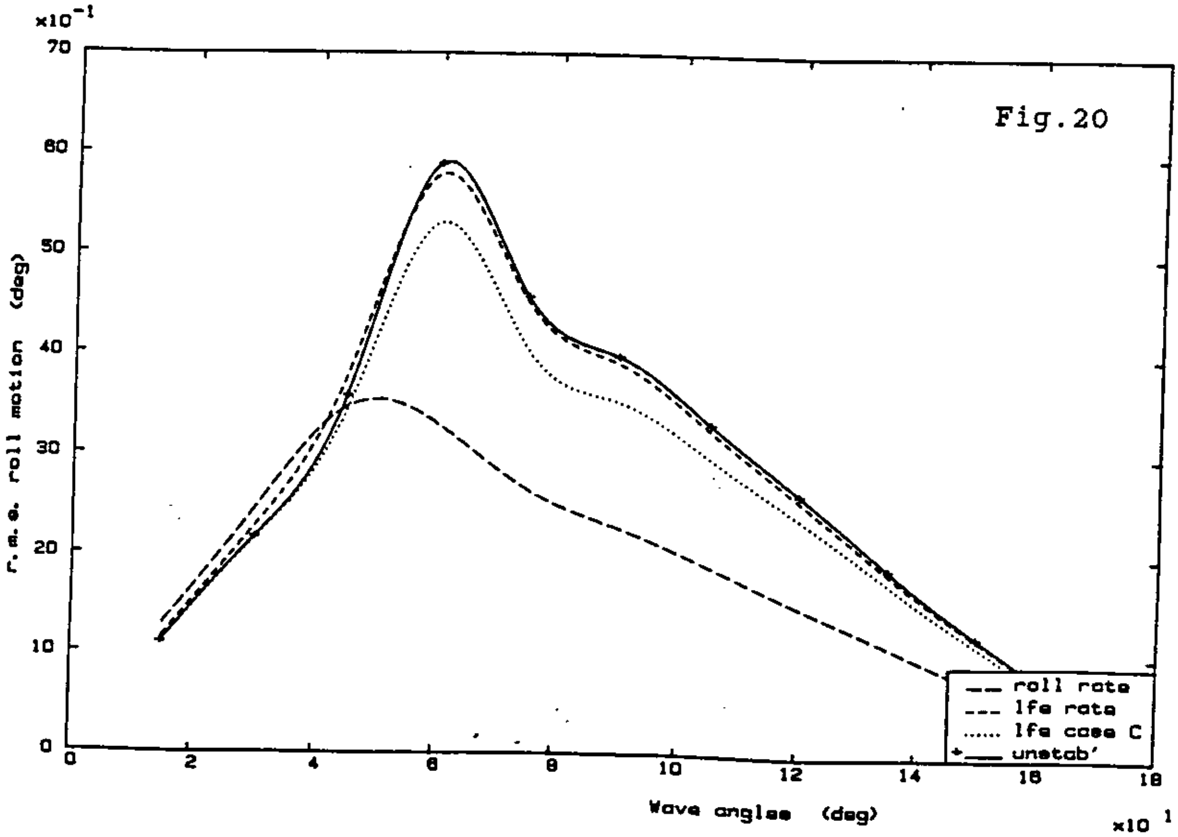

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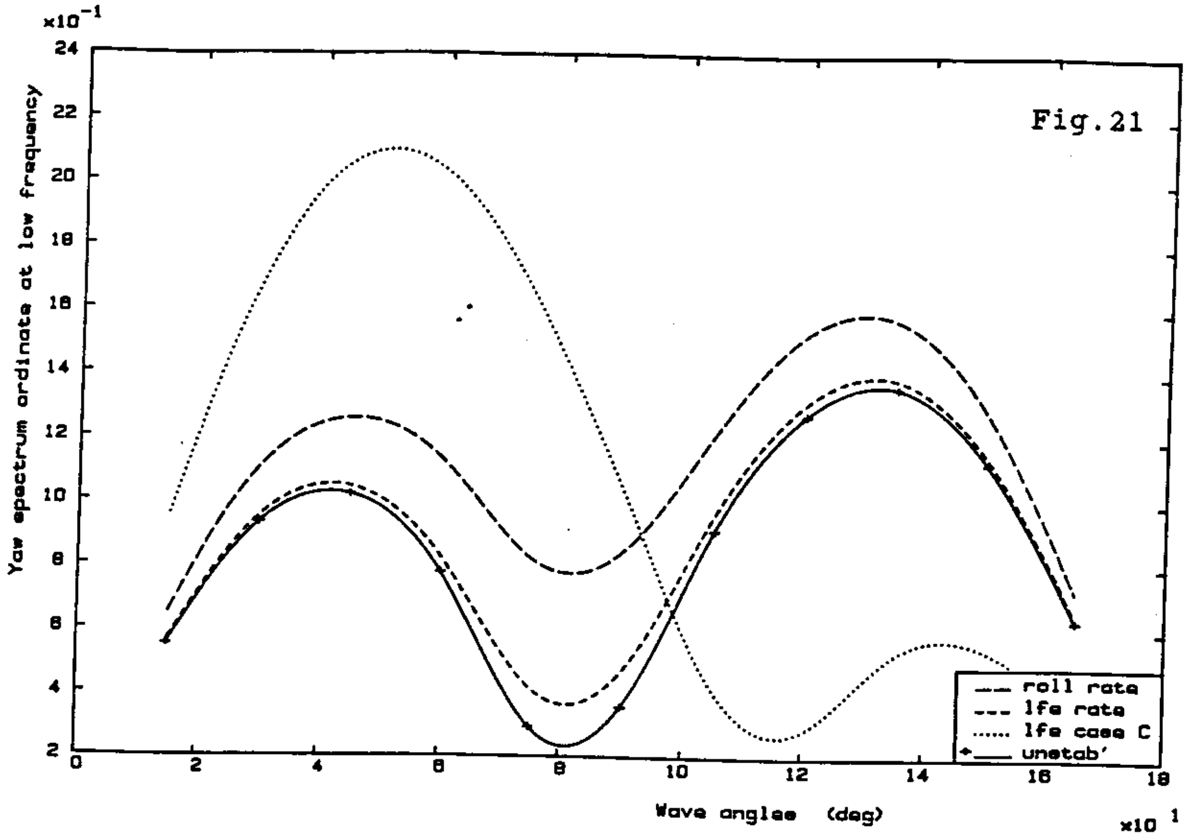

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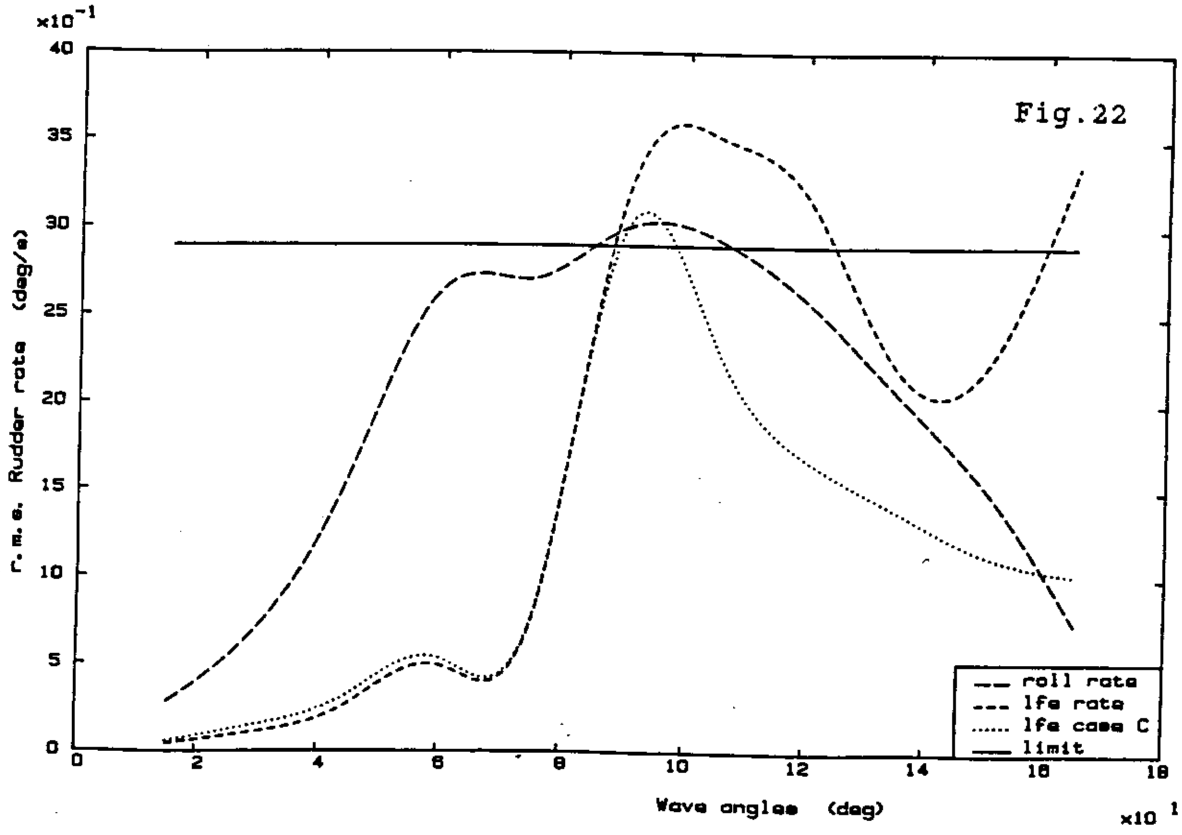

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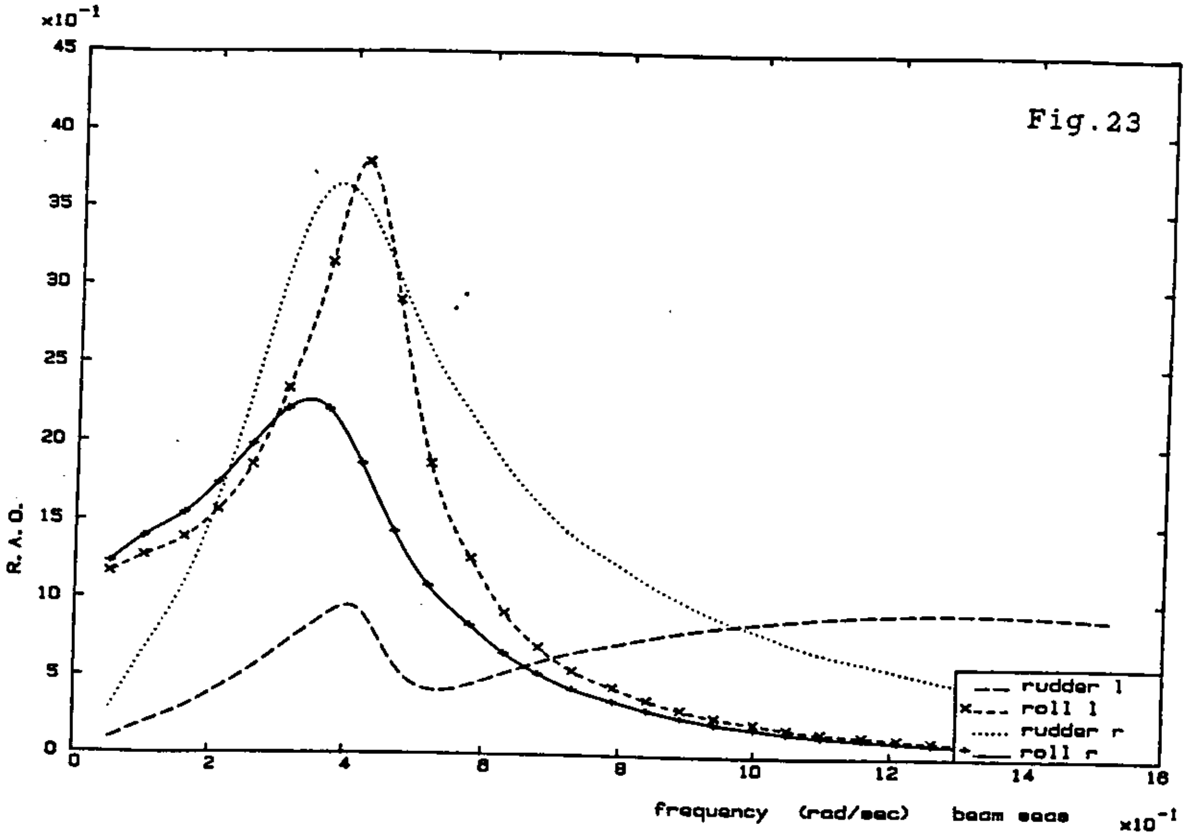
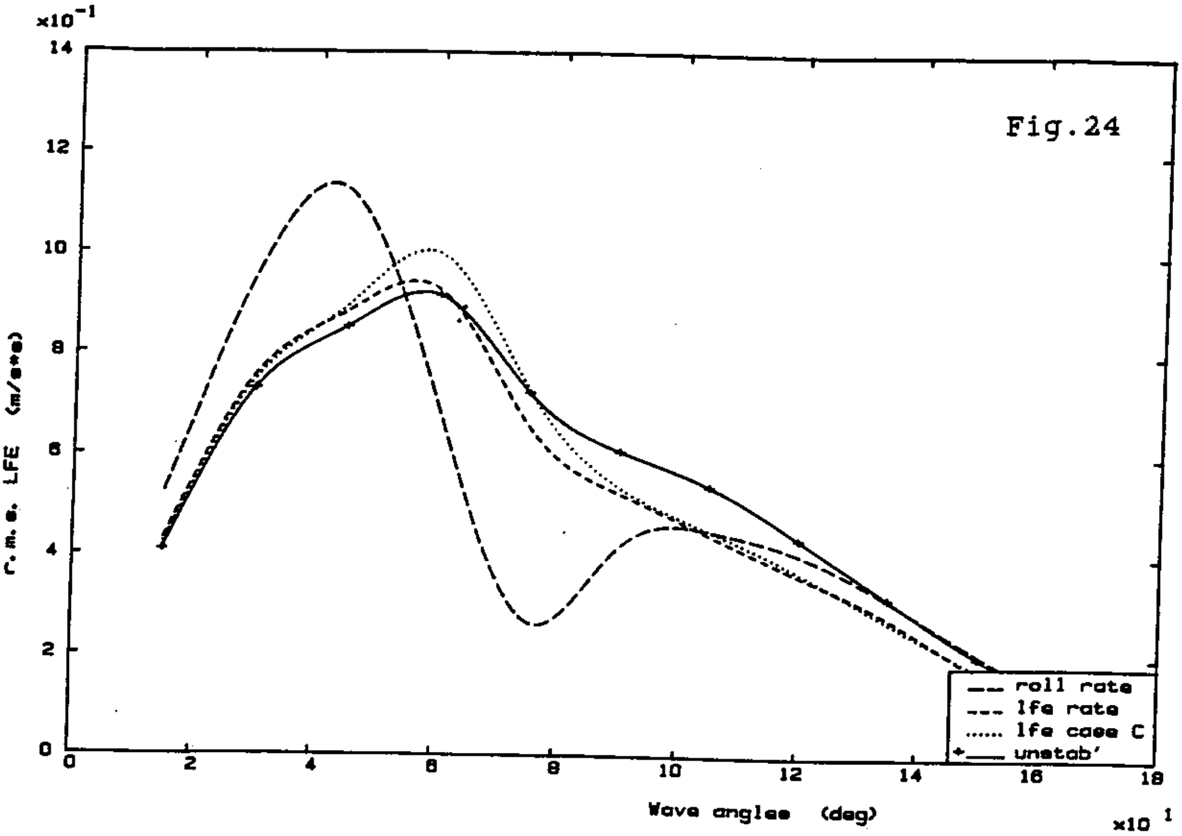
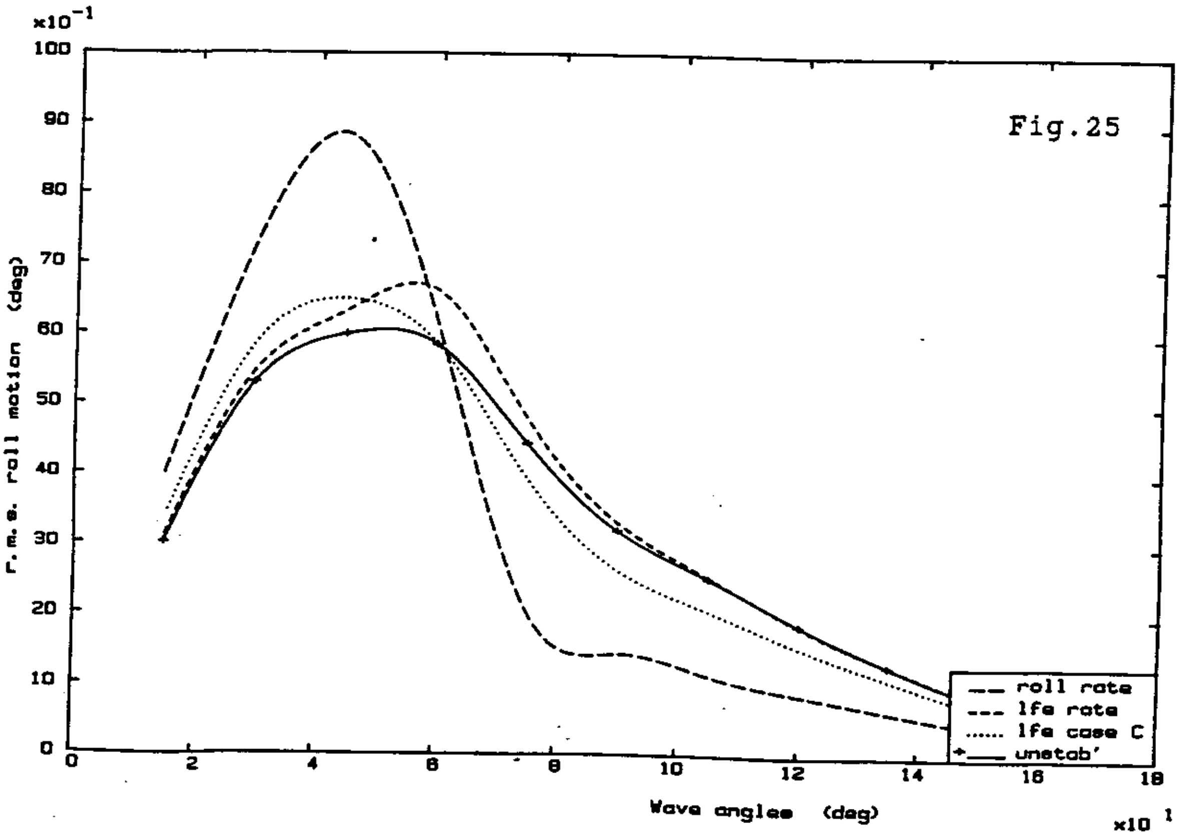


Response spectrum with RLS stabilisation 5.5m 12.4sec 20 kts Newship



LFE response with RLS stabilisation 5.5m 12.4sec 30 kts Newship



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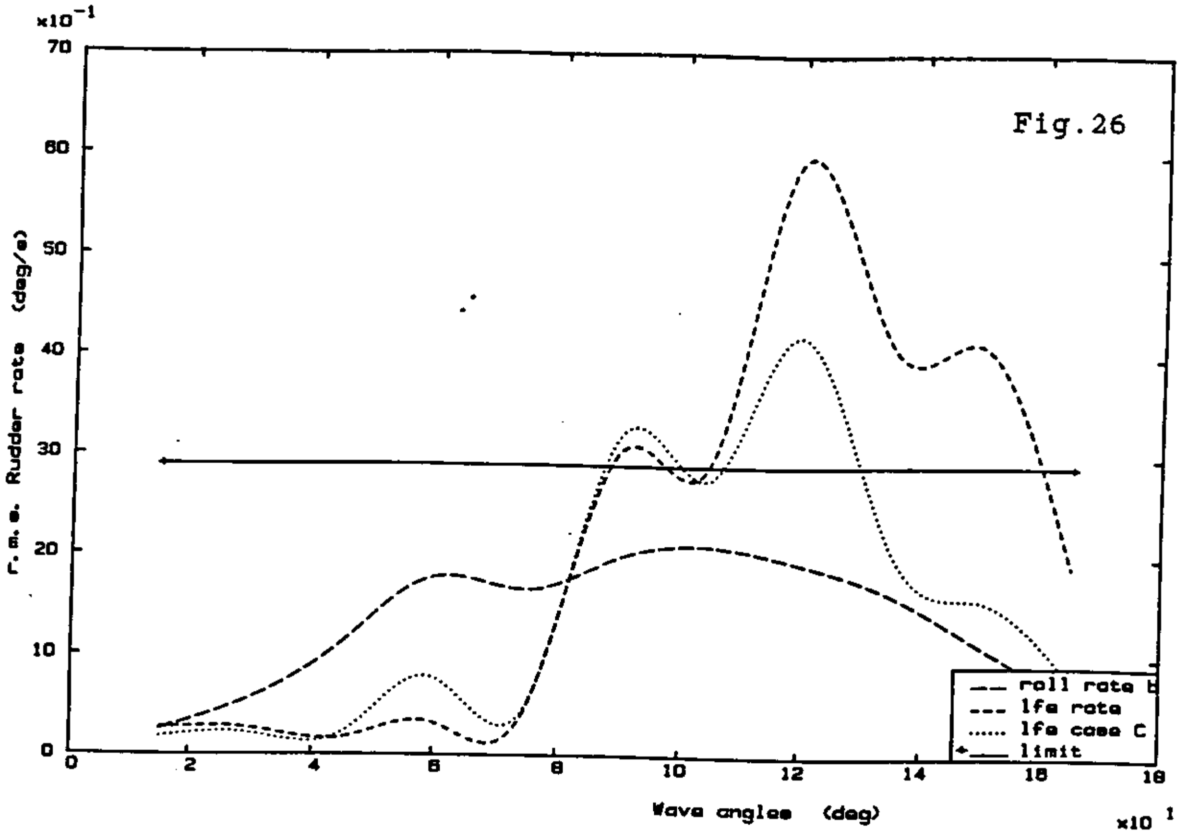

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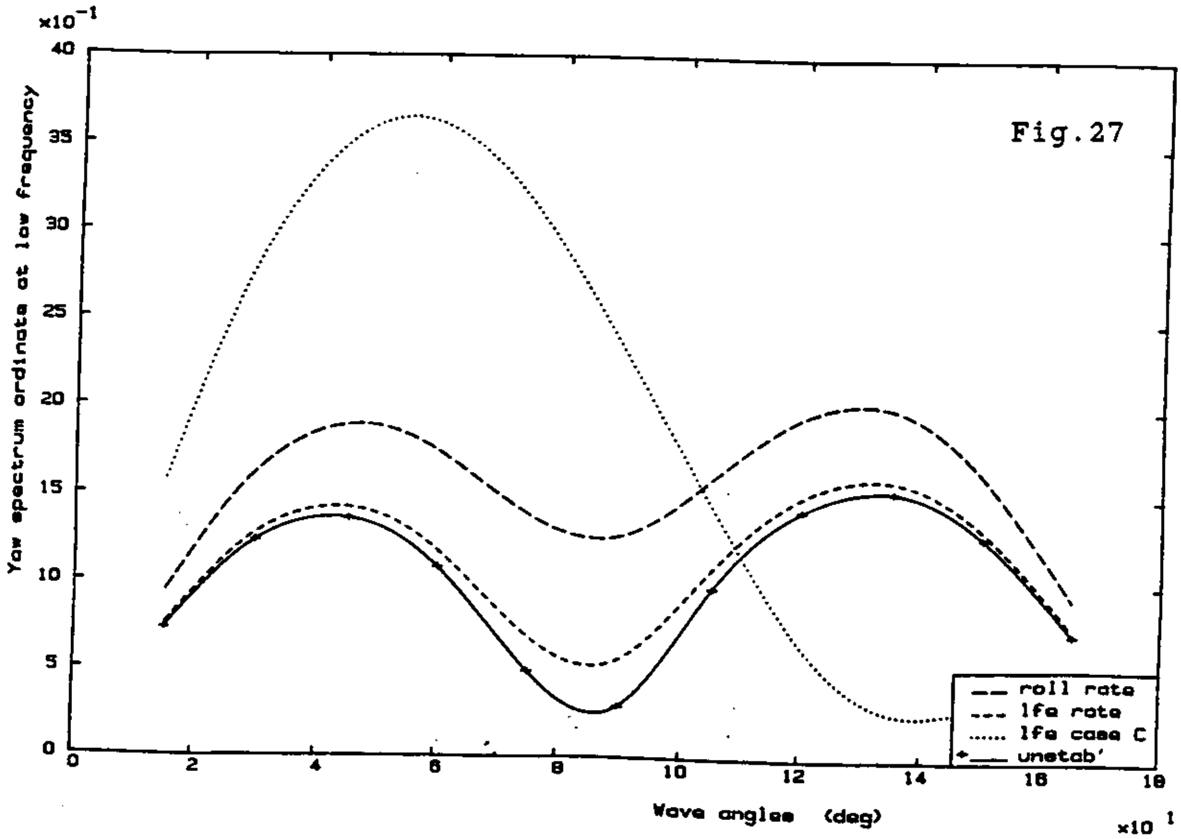

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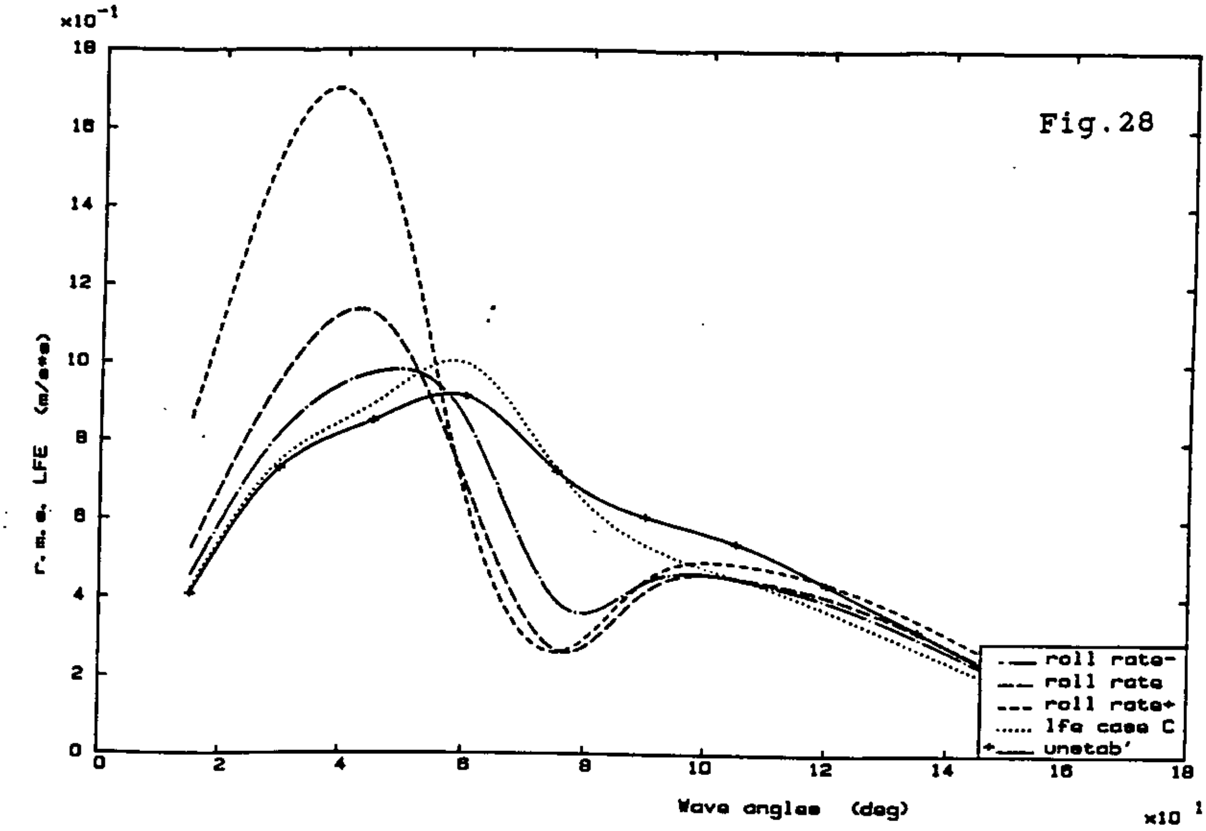

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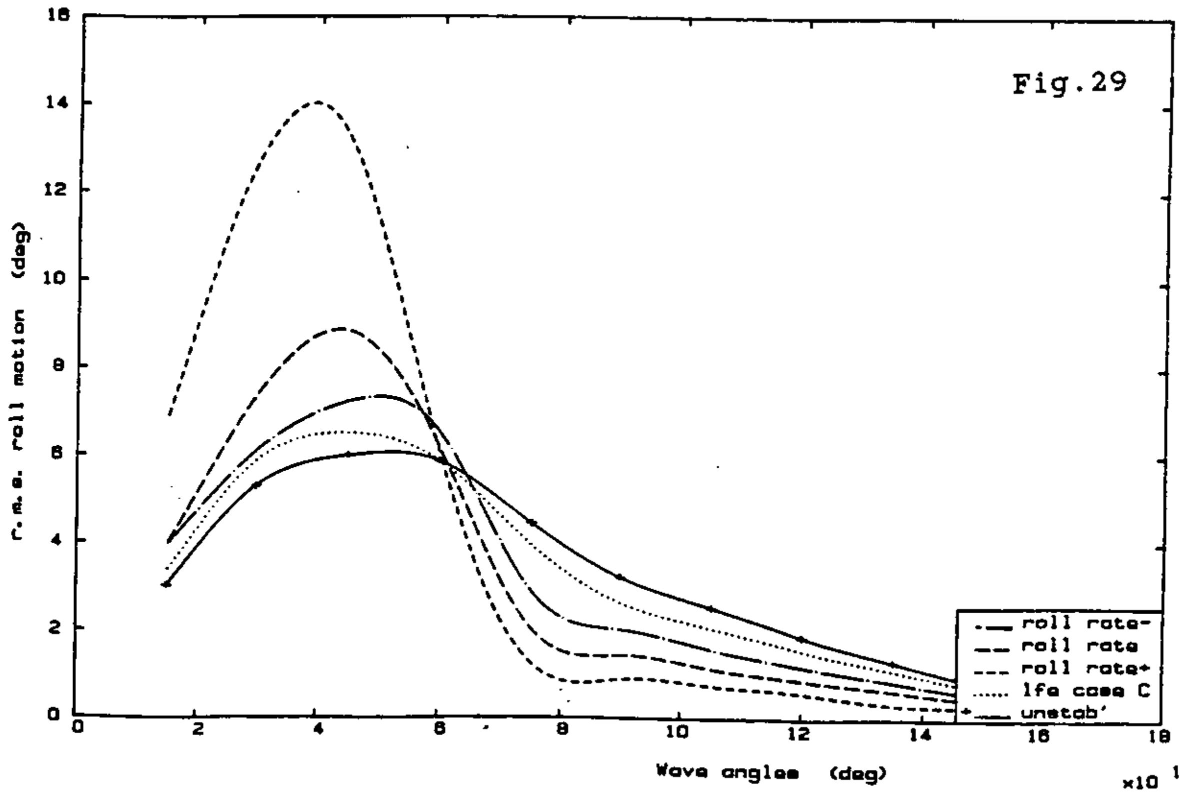

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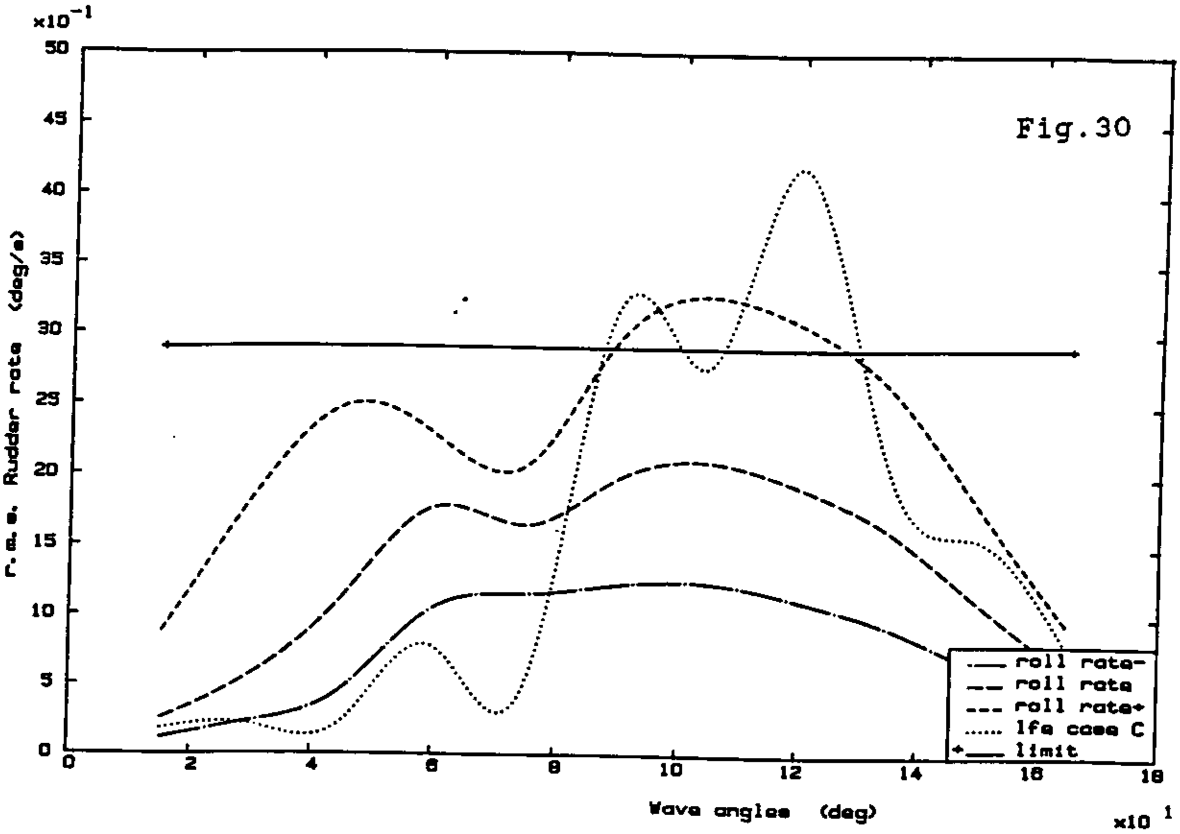
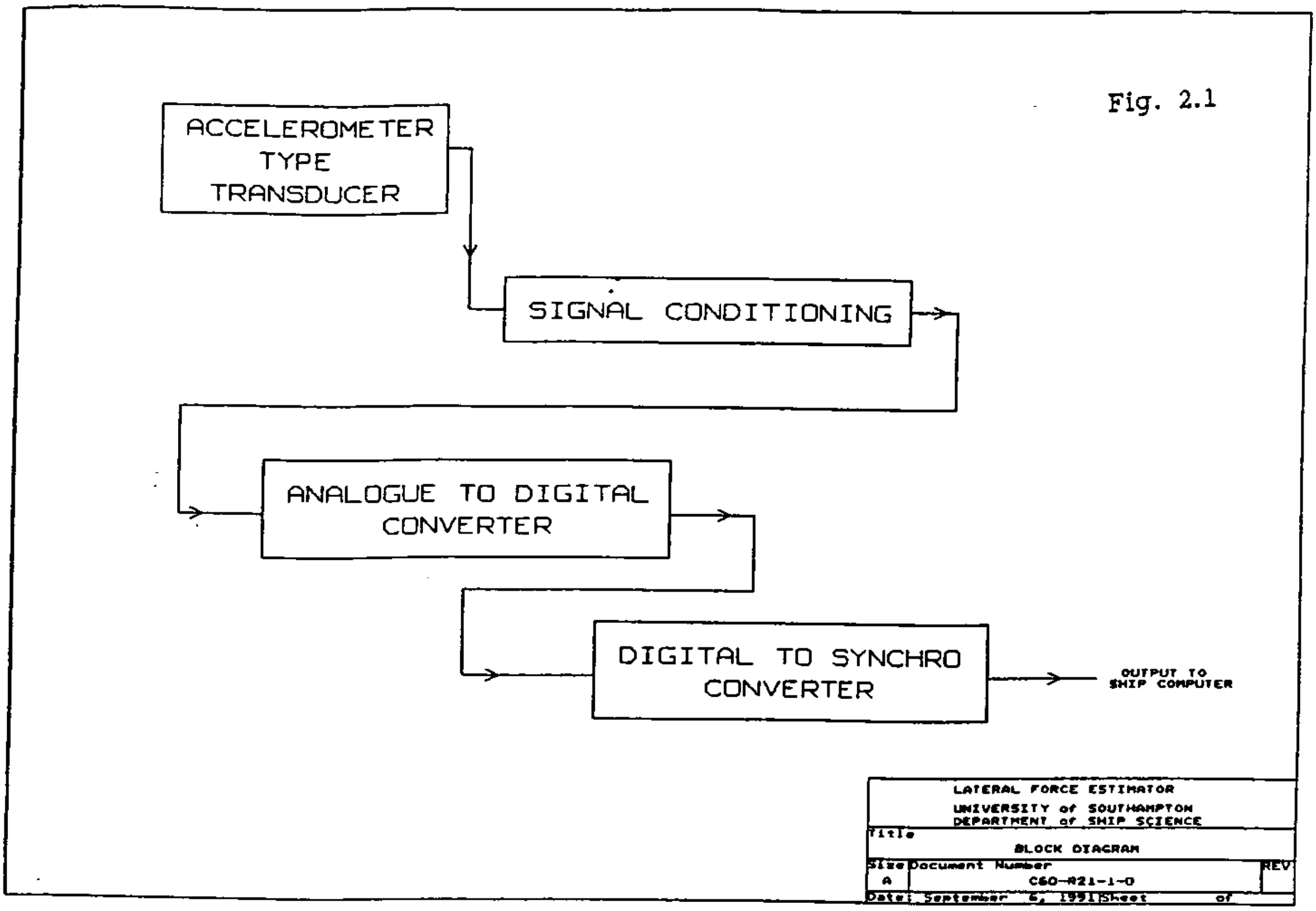


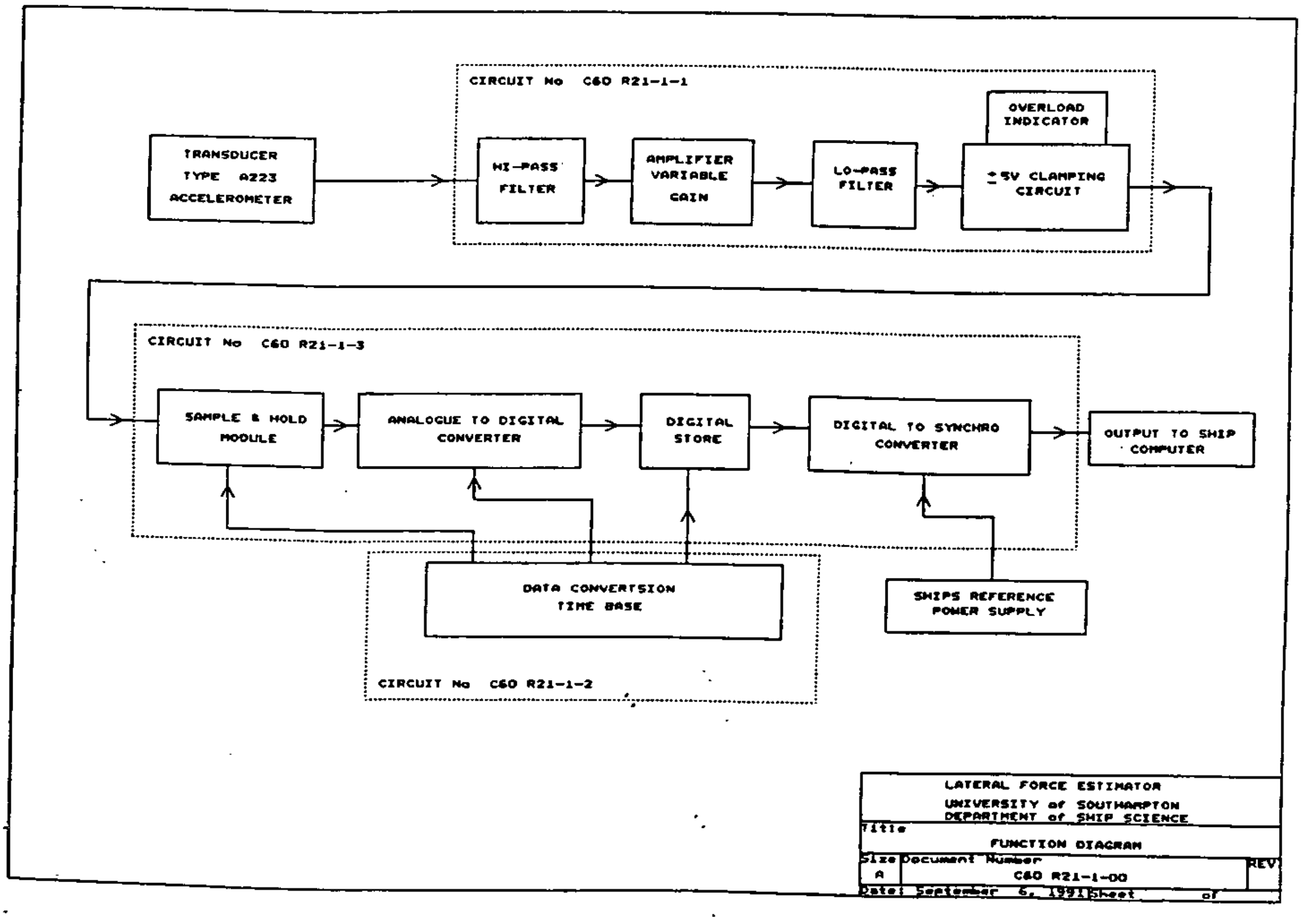
Rudder rate with RLS stabilisation 5.5m 12.4sec 30 kts Newship

Fig. 2.1



ACCELEROMETER TYPE TRANSDUCER
SIGNAL CONDITIONING
ANALOGUE TO DIGITAL CONVERTER
DIGITAL TO SYNCHRO CONVERTER
OUTPUT TO SHIP COMPUTER
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
BLOCK DIAGRAM
Size A
Document Number C60-R21-1-0
REV
Date: September 6, 1991 Sheet of

Fig. 2.2



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TRANSDUCER TYPE A223 ACCELEROMETER
HI-PASS FILTER
AMPLIFIER VARIABLE GAIN
LO-PASS FILTER
OVERLOAD INDICATOR
±5V CLAMPING CIRCUIT
CIRCUIT No C60 R21-1-3
SAMPLE & HOLD MODULE
ANALOGUE TO DIGITAL CONVERTER
DIGITAL STORE
DIGITAL TO SYNCHRO CONVERTER
OUTPUT TO SHIP COMPUTER
DATA CONVERTSION TIME BASE
SHIPS REFERENCE POWER SUPPLY
CIRCUIT No C60 R21-1-2
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
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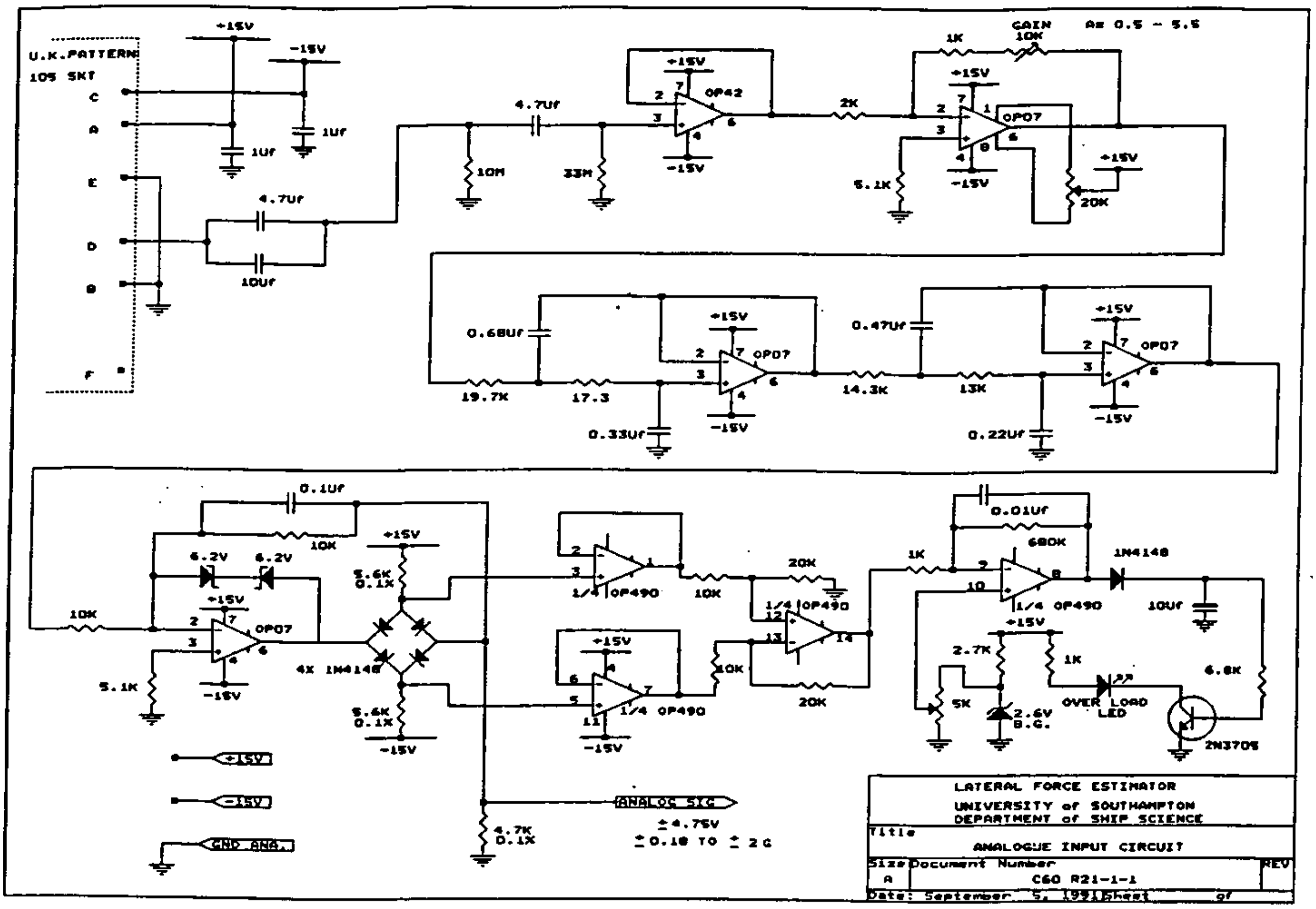
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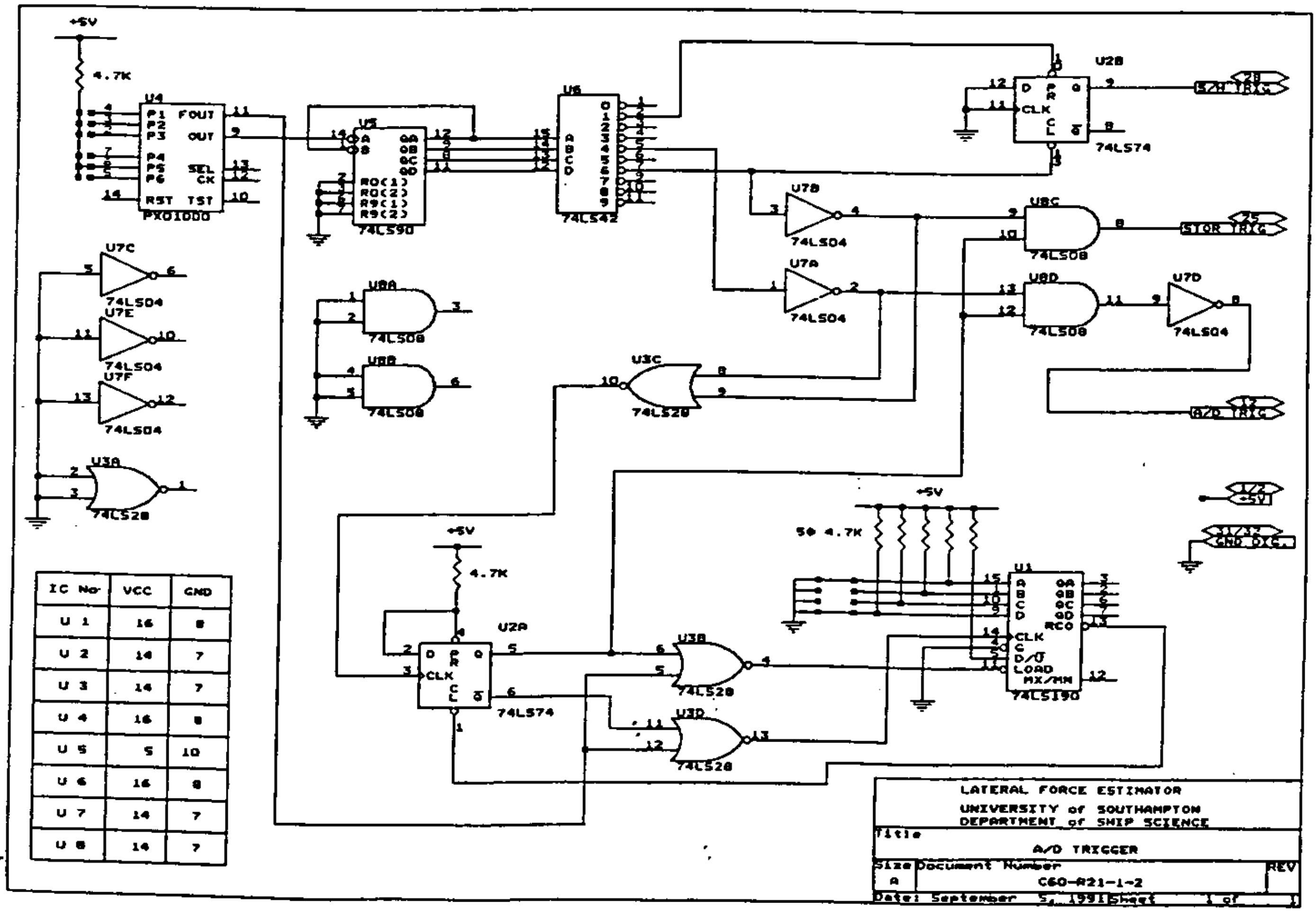
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
ANALOGUE INPUT CIRCUIT
Size Document Number REV
A C60 R21-1-1
Date: September 5, 1991 Sheet of

Fig. 2.3

Fig. 2.4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UNIVERSITY of SOUTHAMPTON
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A/D TRIGGER
Size Document Number REV
A C60-R21-1-2
Date: September 5, 1991 Sheet 1 of 1

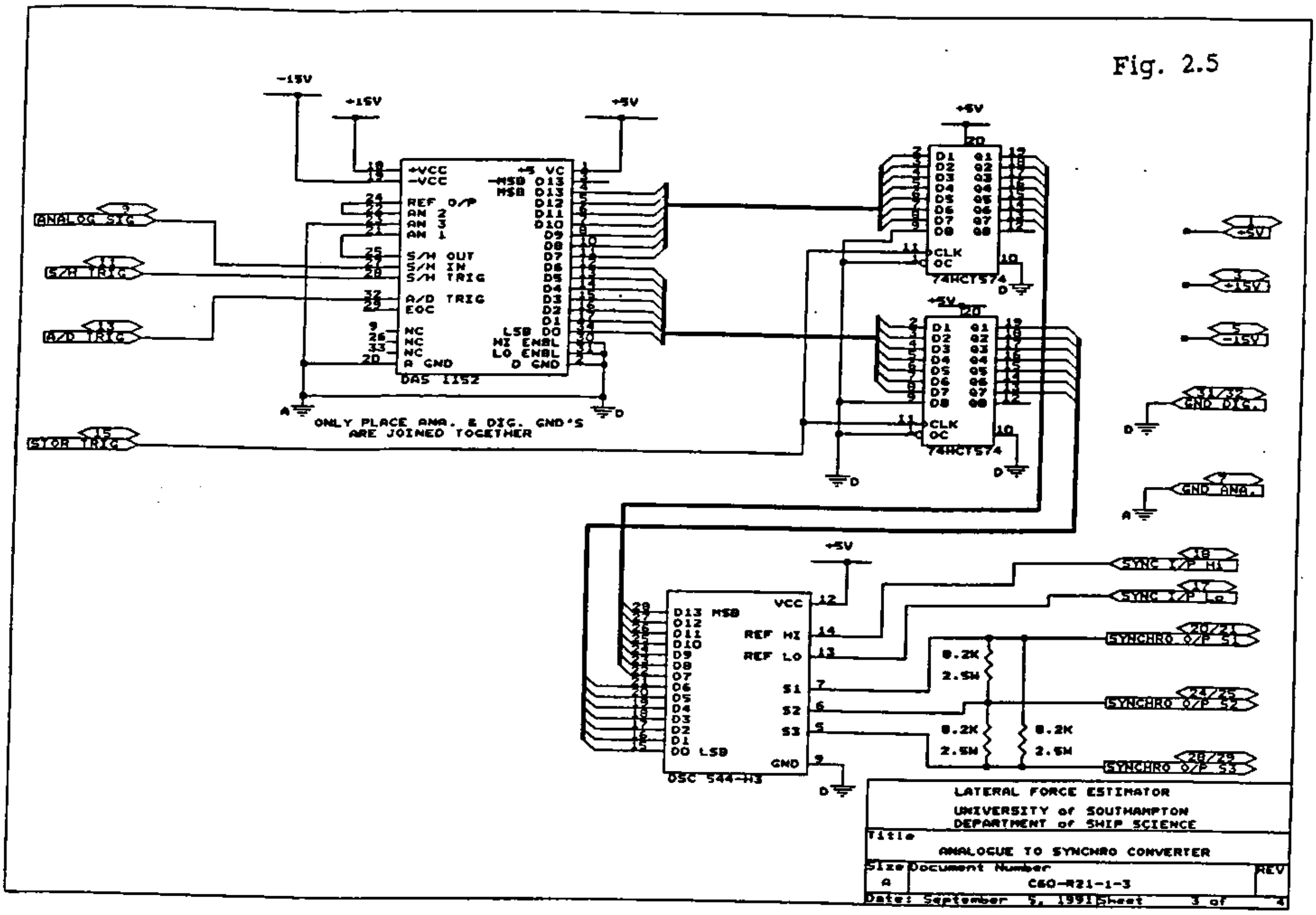
Fig. 2.5
-15V
+15V
+5V
DAS 1152
ONLY PLACE ANA. & DIG. GND'S ARE JOINED TOGETHER
ANALOG SIG
S/H TRIG
A/D TRIG
STOR TRIG
74HCT574
74HCT574
DSC 544-H3
SYNC I/P Hi
SYNC I/P Lo
SYNCHRO O/P S1
SYNCHRO O/P S2
SYNCHRO O/P S3
8.2K
2.5W
GND DIG.
GND ANA.
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
ANALOGUE TO SYNCHRO CONVERTER
Size Document Number
A
C60-R21-1-3
REV
Date: September 5, 1991 Sheet 3 of 4

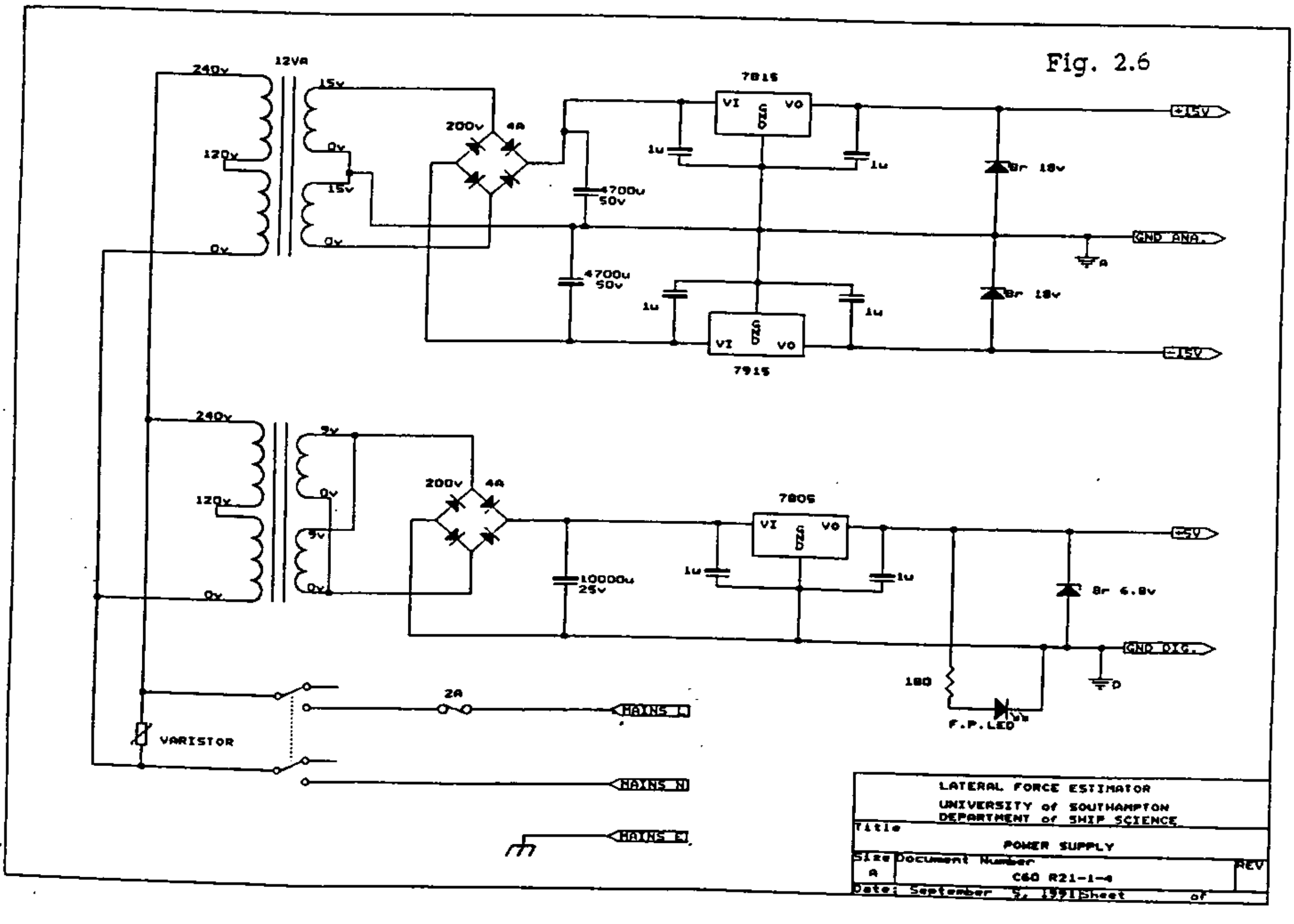
Fig. 2.6
12VA
240v
120v
0v
15v
200v
4A
7815
7915
4700u
50v
1u
8r 18v
+15V
GND ANA.
-15V
9v
7805
10000u
25v
8r 6.8v
+5V
GND DIG.
180
F.P.LED
2A
VARISTOR
MAINS L
MAINS N
MAINS E
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
POWER SUPPLY
Size Document Number
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C60 R21-1-4
REV
Date: September 5, 1991 Sheet of

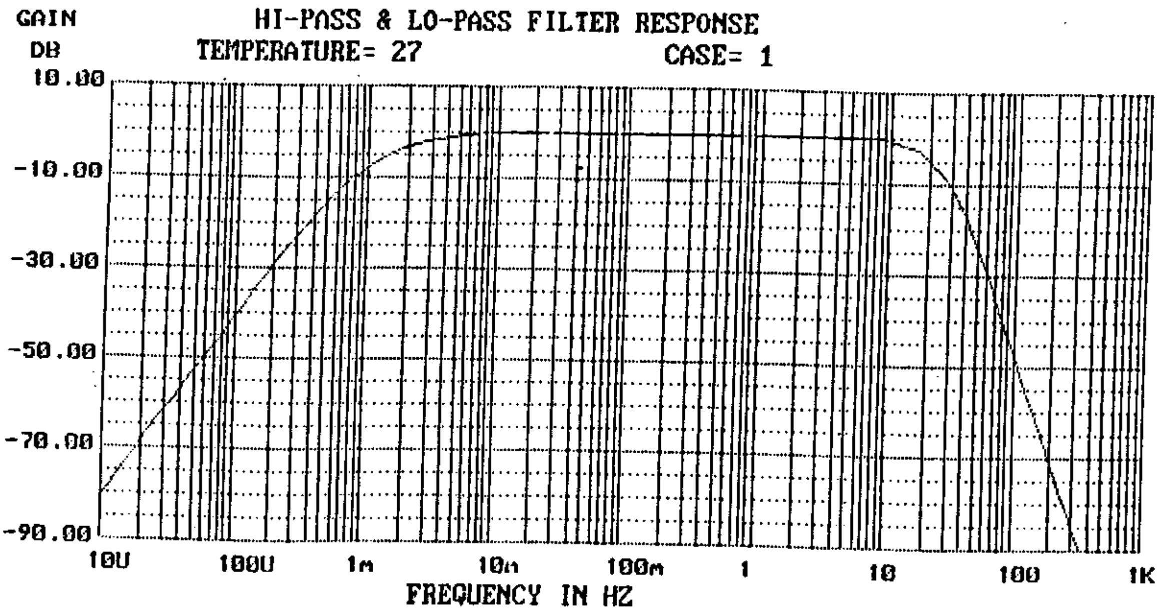

GAIN
DB
HI-PASS & LO-PASS FILTER RESPONSE
TEMPERATURE= 27
CASE= 1
10.00
-10.00
-30.00
-50.00
-70.00
-90.00
10U
100U
1m
10m
100m
1
10
100
1K
FREQUENCY IN HZ

fig. 2.7

Fig. 2.8

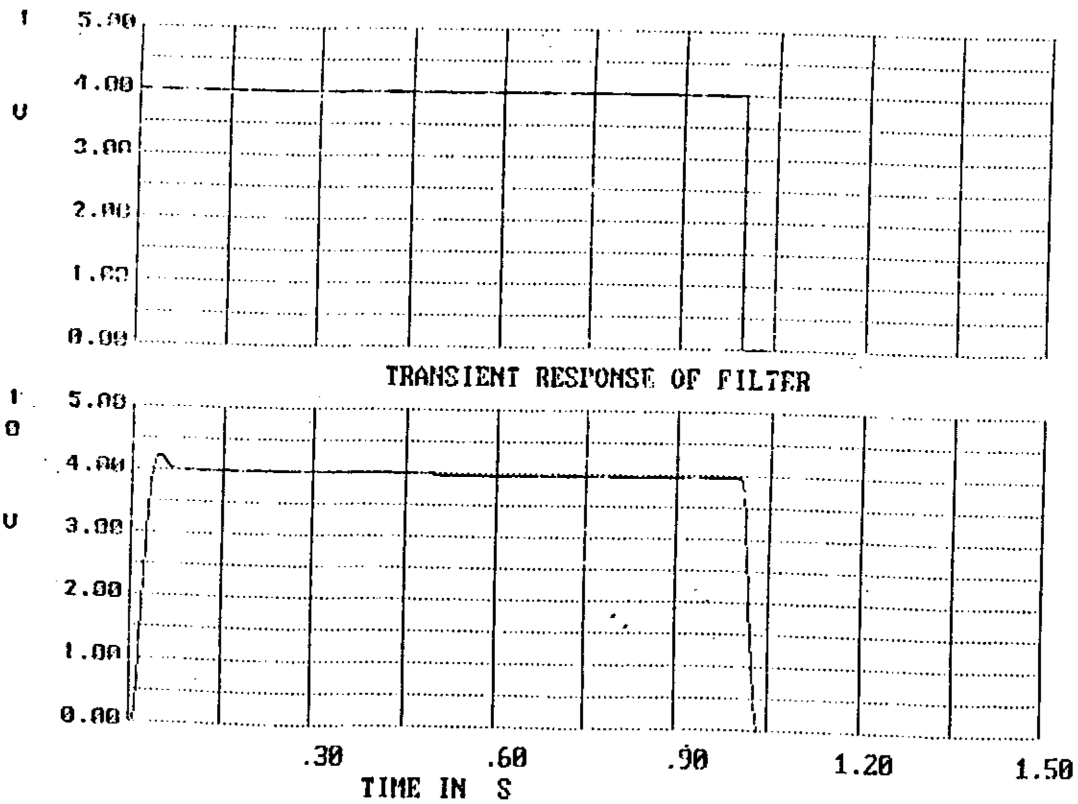
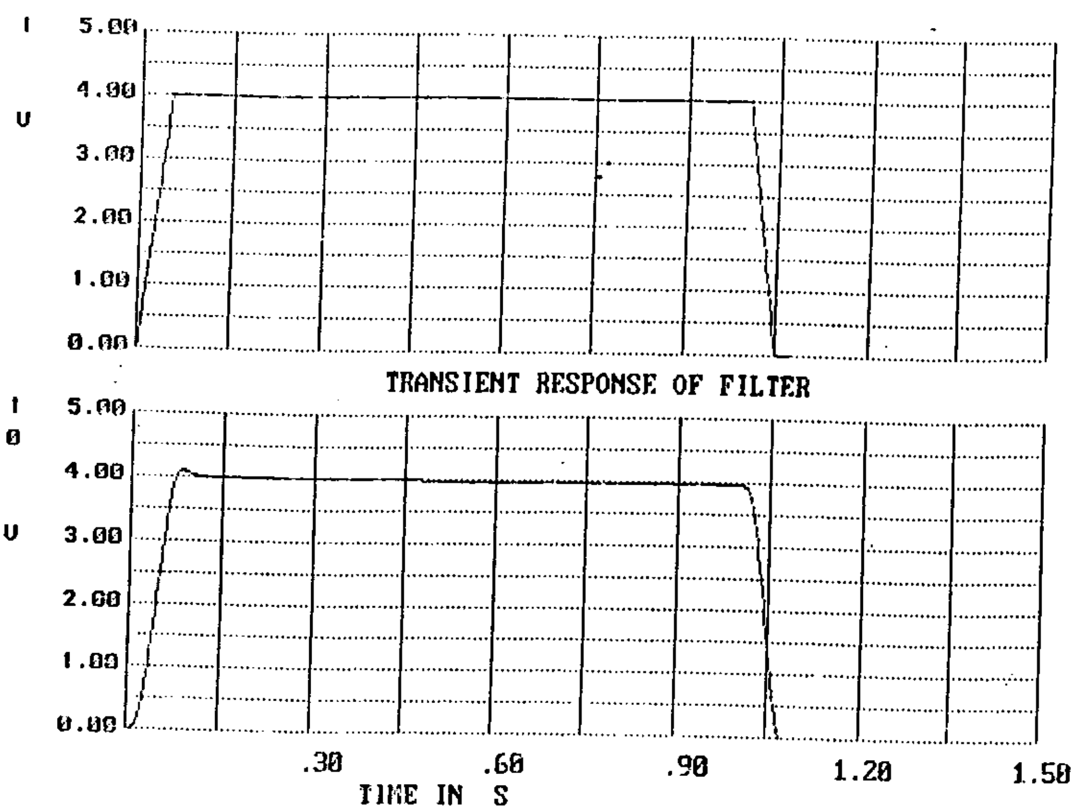

I
V
5.00
4.00
3.00
2.00
1.00
0.00
TRANSIENT RESPONSE OF FILTER
I
O
V
5.00
4.00
3.00
2.00
1.00
0.00
.30
.60
.90
1.20
1.50
TIME IN S

fig. 2.9


I
V
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1.00
0.00
TRANSIENT RESPONSE OF FILTER
I
0
V
5.00
4.00
3.00
2.00
1.00
0.00
.30
.60
.90
1.20
1.50
TIME IN S

APPENDIX D

L.F.E. stabilisation (contract report)

This report comprises of two parts. Part I is a study on the use of filters to improve rudder LFE stabilisation, which is an extension of the last report. Part II is divided into three main sections. Section A is concerned with the forced roll/LFE trials conducted in June onboard HMS Lancaster. Section B gives an up-date of the electronics hardware for the LFE signal conditioning unit. In section C, the test plan for the equipment with the CCU at Sultan is described.

PART I

Introduction

In Tang[4], it was found that the relative sharp rise in the high frequency response of the rudder in the rudder L.F.E. stabilisation (RLS) system has made it rather ineffective. An obvious way to improve the system performance would be to eliminate the high frequency content in the L.F.E. feedback signal, which does little to counteract the dominant motion response near the roll natural frequency. In theory, a low-pass filter can be used to eliminate the undesirable high frequency content. However, owing to the phase-lag characteristics of this type of filter, there are practical limitations. This has been pointed out by Baitis [1] and Van Amerogen [5] when low-pass filters were used for the RRS system. This phase-lag effect has been overlooked in Schmitke[3] when filtering was used in their numerical model based on strip theory. The resulting performance study of RRS, which has shown very favourable trends, should be taken with caution.

The present study is to incorporate some simple but realistic filters into the numerical model in order to examine their effects on the motion responses. It is anticipated that the phase-lag characteristics of low-pass filters would cause undesirable motion responses. Therefore, some phase-lead compensation circuits have been explored and applied in a cascade fashion in the system with the expectation that this will counter-balance the undesirable phase-lag incurred by the low-pass filters selected in this study.

This study is divided into three sections. In section one, a brief introduction of the filters used in this study and their transfer function characteristics is given. The effects of low-pass filter on motion responses are examined in some details in section two. Section three examines the effects of phase-lead compensation on the overall motion responses with RLS.

Basic Background

Filters are often used to emphasize or eliminate the frequency components of an electrical signal, such that a desirable signal is obtained. They are normally classified as low-pass, high-pass, band-pass and stop-pass, depending on the ranges of frequencies that are being suppressed. In this section, attention will be focused on some low-pass filters, looking at their basic characteristics and examining the transfer functions that govern the output to the input. In terms of hard-ware, these filters can be constructed using resistors, capacitors and operational amplifiers as the basic components.

A low-pass filter is one in which the pass band extends from zero frequency ($\omega = 0$) to a cut-off frequency ($\omega = \omega_c$). Ideally, a low-pass filter would pass signals from zero frequency up to the cut-off frequency with no amplitude attenuation and rejects all signals above ω_c, as illustrated in fig.1a, where ω_c is at 1.0 rad/sec. However, in practice, this ideal filter cannot be realised. The basic features of a low-pass filter can be demonstrated using a linear low-pass filter having a transfer function defined by:

$$T(j\omega) = \frac{1}{s\tau_f + 1}$$

where $s = j\omega$ and τ_f is the time constant, which has a single pole on the left hand real-axis of the complex-plane. Varying the excitation frequency from zero to a high value, the frequency response of the filter can be constructed. The resulting amplitude and phase response of this filter are given in fig. 2a and 2b. As can be seen from these figures, the amplitude response is far from the ideal and that the phase-lag is quite noticeable. Also, the cut-off frequency is not so readily recognized as in the ideal case. In practice, the cut-off frequency is defined as the frequency at which the amplitude response is 0.707, the half power point. In this particular case, it corresponds to 1 rad/s. This shows that the transition from the pass-band to the stop-band is rather slow and that undesirable frequency contexts are not suppressed totally. The attenuation of this filter in the transition band is in fact -6dB/octave. This transition band can be reduced to bring about a higher attenuation. For example, a second order filter with two poles on the left hand side of the complex plane can be used to achieve a -12db/octave attenuation. However, a Butterworth filter is of more practical interest.

A Butterworth filter is defined by the transfer function below:

$$| T_n (j\omega) | = \frac{1}{(1 + \omega^{2n})^{1/2}}$$

By increasing the index n, the number of pole terms in the complex plane increases, which can bring the resulting amplitude response close to the ideal filter. For example, when n=5, the amplitude and phase responses are given in fig.3a and 3b. From these figures, it is quite clear that in order to suppress the unwanted frequency totally, the cut-off frequency should be at most about a third of that particular frequency. Within this pass-band, the phase-lag can be up to 100 degrees. Although the amplitude response can be 'improved' by increasing n, for example, for n=5 attenuation is about -30db/octave comparing with -60db/octave for n=10, the phase-lag incurred will also increase quite markedly (about twice the n=5).

Other types of filter such as the Chebyshev and Elliptic filter can also be used to reduce the transition band further. However, ripples are present in the amplitude response of these filters and the phase-lags are also higher than the Butterworth filters of the same order. Therefore, for the present study, only the linear filter and the Butterworth filter with n=5 will be used to explore their effects on the motion response.

Effects of Low-pass filters

For this part of the study, the RRS strategy with simple rate feedback has been used to examine the effects of the two low-pass filters in the last section. These filters, which are defined by their transfer functions, have been incorporated into the numerical ship motion model, suppressing the high frequency content of the feedback demand signal. A comparison of the r.m.s. roll motion response due to these filters are given in fig.4.a with the corresponding r.m.s. rudder rate demand in fig.4.b. The simulations were based on an ITTC two-parameter spectrum (long-crested) with a modal period of

12.4 seconds and a significant wave height of 5.5m. The cut-off frequency of the filters was set at 1 rad/s whilst the natural frequency of the vessel is about 0.42 rad/s.

From fig.4.a, it can be seen that the linear filter has improved the roll motion at quartering seas slightly whilst there is a marginal increase in the motion at other wave angles compared to the case of simple rate feedback. The corresponding rudder rate demand in fig.4.b suggests that if the feedback gain was increased, the overall motion response could be improved a bit further. The reduction in the rudder rate is due to the relatively wide transition band in the filter amplitude response with a gradual increase in attenuation, which effectively reduces the pre-set feedback gain. The relatively low phase-lag characteristic in the frequency range of interest and the fairly wide transition range have contributed little towards a more favourable motion response nor brought about any detrimental effects.

The resulting motion response with the application of the Butterworth filter is of more interest. In fig.4.a, it is quite apparent that between quartering to head seas regions, the resulting roll motions have been reduced more substantially, but on the other hand, for wave angles greater than 60 degrees, roll motions have in fact increased. The rudder rate in fig.4.b shows a similar trend. It is quite evident that some form of detuning process was brought about by the Butterworth filter. In order to find out what is causing this detuning effect, the process should be looked at in terms of the encounter frequency and not the wave-angles. Also, the phase relationships between the ship motions, wave moments and the stabiliser moments should be examined.

Diagrams 1a to 1c illustrate the phase relationships of a stabilised ship with simple rate feedback, which were drawn based on the response spectrum data. At low frequency, the roll motion leads the waves by about 90 degrees; near resonance, the phase difference is near zero, and at high frequency, the motion lags the wave by about 90 degrees. It can be said that at low and at high frequency, the wave moments and roll motions are about 90 degrees out of phase, and the induced motions are small, whilst near resonance, the phase angle is small and the induced motion is high. Therefore, the motion can be related to the cosine of the phase angle between the roll motion and the waves, i.e. if the stabiliser moment (S) brings the phase angle between the resultant wave moment (W + S) and motion (M) towards 90 degrees, the net effect is stabilisation, whilst if the phase angle is moving towards zero degrees, then it is destabilisation.

It is quite clear that, at low frequency, the rate feedback is destabilising as illustrated in diagram 1.a. Near resonance, the rate feedback is very effective as it brings the phase difference between the wave moments and motion towards 90 degrees, whilst at high frequency, the stabilising effect is marginal due to the small stabilising moment available. Therefore, simple rate feedback is only effective when applied near resonance as was pointed out by Conolly [2].

Now, by introducing a Butterworth filter, the phase relationship between the stabiliser moments and the motions is altered. These phasor diagrams are illustrated in diagrams 2.a-2.c. At low frequency and at high frequency, the filter has brought about hardly any changes in the motion response. However, near the resonance region, the phase-lag affected by the filter has caused unfavourable phase relationships between the resultant wave moments and the motions, i.e. reducing the phase angle between the motion and the wave moments towards zero.

Having found what is causing the motion amplification, the trend in fig.4.a for the Butterworth filter can be explained more fully. In the quartering to head sea region (less than about 45 degrees), the encounter frequencies are about half the cut-off frequency. Although the near resonance response is destabilising, the sum total of the stabilising effects at the low frequency end has improved the overall motion response. However at 60 degrees, quite substantial part of the encounter frequency is near and slightly above the resonance frequency, which makes the phase-lag effect more prominent, with the resulting detuning effect augmenting the motion. At larger wave angles, i.e. higher frequency of encounter, the phase-lag effect at these frequency is marginal, but the response near resonance is still dominant. Hence there is an overall motion amplification. The same is true of the linear filter, except that the phase-lag effect is less obvious due the more gradual transition band.

The detuning effect of this Butterworth (n=5) low pass filter is even more noticeable when the cut-off frequency is placed near the natural roll frequency. In fig. 5a, it can be seen that the phase-lag introduced by the filter with a low-cut off frequency would amplify the motion by almost a factor of two compared to the high cut-off frequency case. The rudder is in fact working harder in this case as shown by the higher rudder rate amplitudes in fig.5b. Unfortunately, it is not counteracting the motion effectively.

Phase-lead compensation

Hence, in the work presented thus far, the limitation of low-pass filters reported in Baitis [1] and Van Amerogen [5] for RRS is confirmed. Similar effects were also found when low-pass filters were applied to RLS.

To counteract the unfavourable phase-lag incurred by low-pass filters, phase lead compensation filters can be used. Typical responses of such a filter is shown in fig.6a and 6b, which is defined by the transfer function:

$$T(j\omega) = \frac{s\tau_f + a}{s\tau_f + b} \qquad\qquad 0 \le a \le b$$

However, for the present study, the relatively low phase lead and the relatively high amplitude attenuation would make this filter rather ineffective. Instead, an all-pass filter will be used. This can be achieved by making a=-1 and b=1, giving a transfer function:

$$T(j\omega) = \frac{s\tau_f - 1}{s\tau + 1}$$

The amplitude and phase response are shown in fig.7a and 7b.

From an preliminary work, it was found that applying this all-pass filter to RRS with low-pass filter did not produce significant improvements. At close examination, it was apparent that this seemingly low performance stemmed from the fact that the improvement could only be small as the high frequency content in the roll feedback signal was quite low in the first place.

The all-pass filter was applied to the RLS case with Butterworth low-pass filter with the expectation of an increase in effectiveness of RLS. Unlike the RRS case in the last section, the resulting phase relationship between the motions, moments and the rudder are difficult to follow because of the complex phasing between RLS, sway, roll and yaw. This means that it would be easier to find out how the phase-lead compensator affects the motion by varying the amount of phase-lead until the lowest r.m.s. motions are obtained. The amount of phase-lead can be adjusted by changing the cut-off frequency of the filter.

In the first instance, a Butterworth filter with cut-off frequency at 1.1 rad/sec was used. This only introduced a small amount of phase-lag to the signal and did not affect the motion greatly. The cut-off frequency of the all-pass filtered was then selected between 1.4 to 0.3 rad/sec at a few discrete frequencies. The resulting r.m.s motions are shown in fig.8a,8b and 8c. The 0.3 rad/sec cut-off has produced the best overall performance compared to those resulting from higher frequency cut-off. Even so, this performance is only a slight improvement on the original unfiltered case. Although the rudder r.m.s. rate responses in fig.8c show little variation with different cut-off frequencies, there are in fact differences in the rudder response spectra especially near the roll natural frequency. The rudder response near the roll natural frequency is quite sensitive to the amount of phase-lead introduced.

To improve the feedback signal further, a Butterworth filter with a lower cut-off frequency was then used to suppress a wider range of the undesirable frequency. The cut-off frequency was located at 0.9 and 0.7 rad/sec. Again a selection of cut-off frequencies for the lead compensators were used to give varying degrees of phase-lead. The resulting r.m.s motions are shown in fig.9(a-c) and fig.10(a-c). In general, the motion amplitudes have been reduced further except with the phase-lead compensators having the lowest cut-off frequencies. At these frequencies, motion amplifications were evident. This was probably due to the over-compensation in the phase advance. The best performance was derived from the combination of Butterworth cut-off at 0.7 rad/sec and all-pass at 0.5 rad/sec (fig10a-c). Very similar resulting motions were also found in fig.9a-c with the phase-lead cut-off frequency at 0.3 rad/sec. However, looking at the rudder rate amplitudes, those in fig.10c would be more desirable as it is more effective at a larger range of wave angles. Judging from these results, it may seem possible to improve the performance further if the cut-off frequency of the butterworth filter is lowered still. However, this would very likely to incur complications as the cut-off frequency would be too close to the roll natural frequency.

In fig.11a,b, a comparison is made of the filtered RLS with the RRS and RLS results from the last report (Tang[4]). Although the filtered RLS case has reduced the motions quite substantially compared to the unfiltered case, it is still not as effective as the RRS with normal rate feedback. The comparisons in terms of the response spectrum are shown in fig.12a,b.

The roll response near the natural frequency has been successfully reduced by the filters even to a level lowered than the RRS case. However, it is quite evident that at lower frequencies, motion amplification has been brought about by the resulting phasing of the filters. This is also apparent in the rudder response at these frequencies when filters have been used (fig.12b). One the whole, the rudder response has been improved with these filters to achieved lower r.m.s. motions.

Concluding remarks

It has been shown that low-pass filters can incur undesirable phase-lags into feedback signals, which will degrade the performance of rudder induced motion stabilisations. This phase-lag effect can be reduced quite substantially by phase-lead compensation using an all-pass filter, thus improving the effectiveness of the RLS strategy. Despite this improvement, the performance of RLS is still inferior to RRS with simple roll rate feedback.

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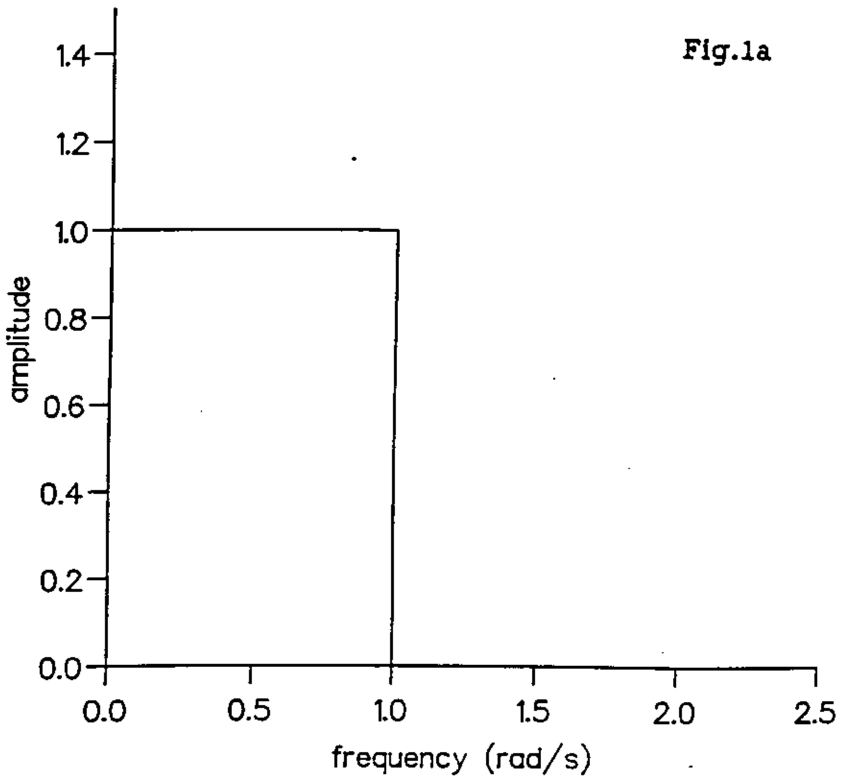
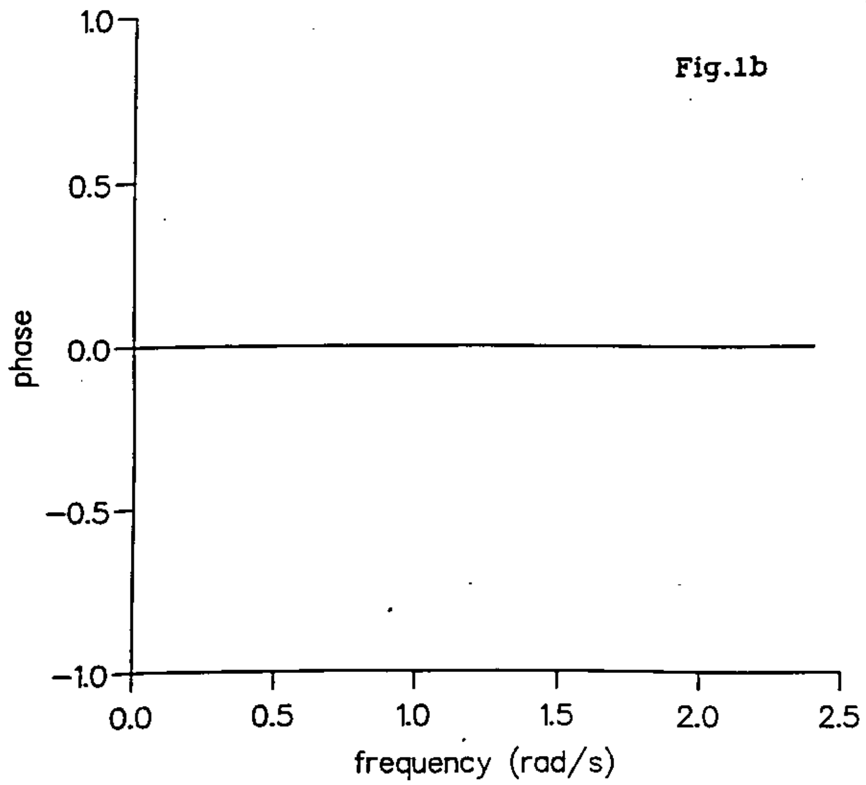
Response of an ideal low-pass filter



Response of an ideal low-pass filter



Response of a linear low-pass filter

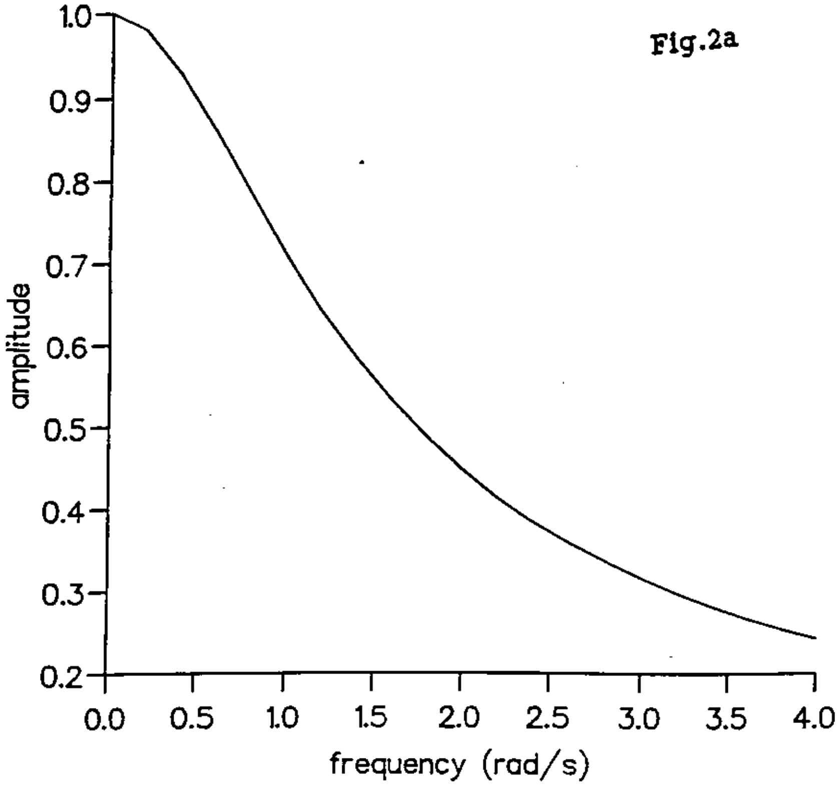


Fig.2a

Response of a linear low-pass filter

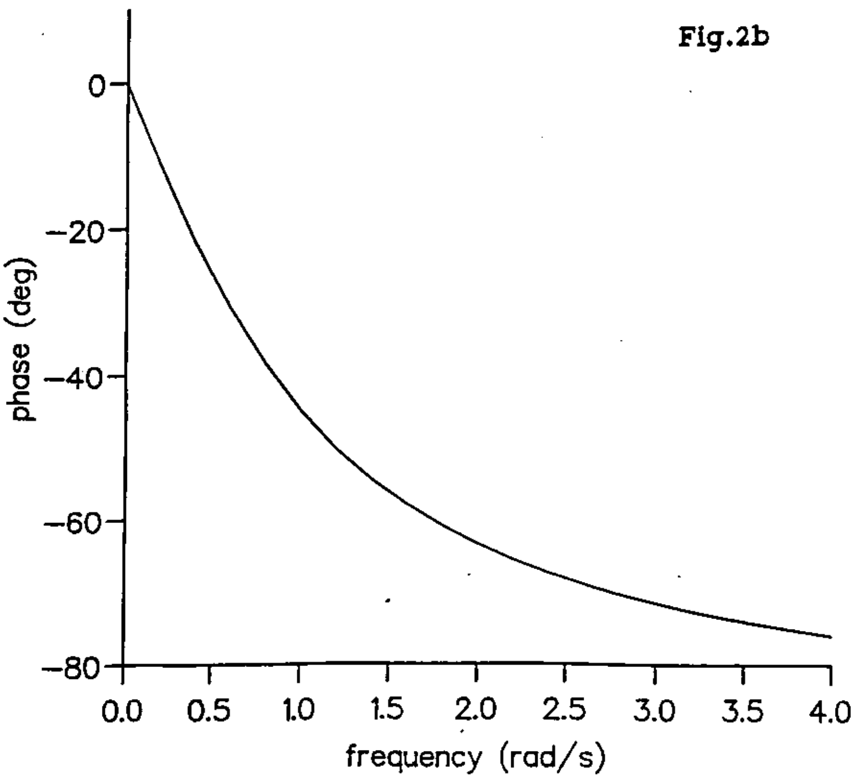


Fig.2b

Butterworth n=5 filter response

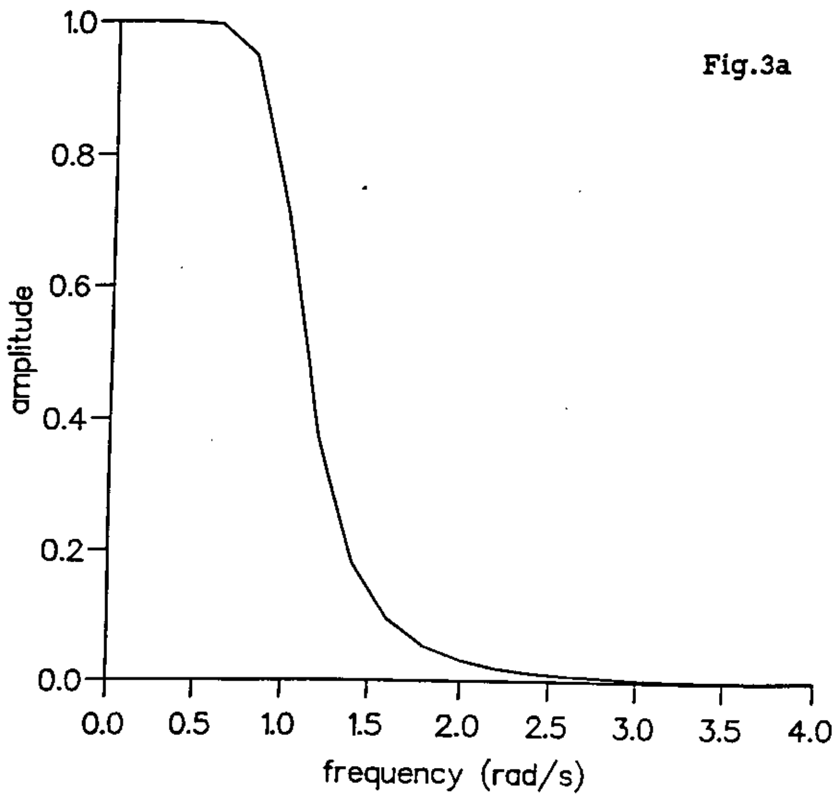


Fig.3a

Butterworth n=5 filter response

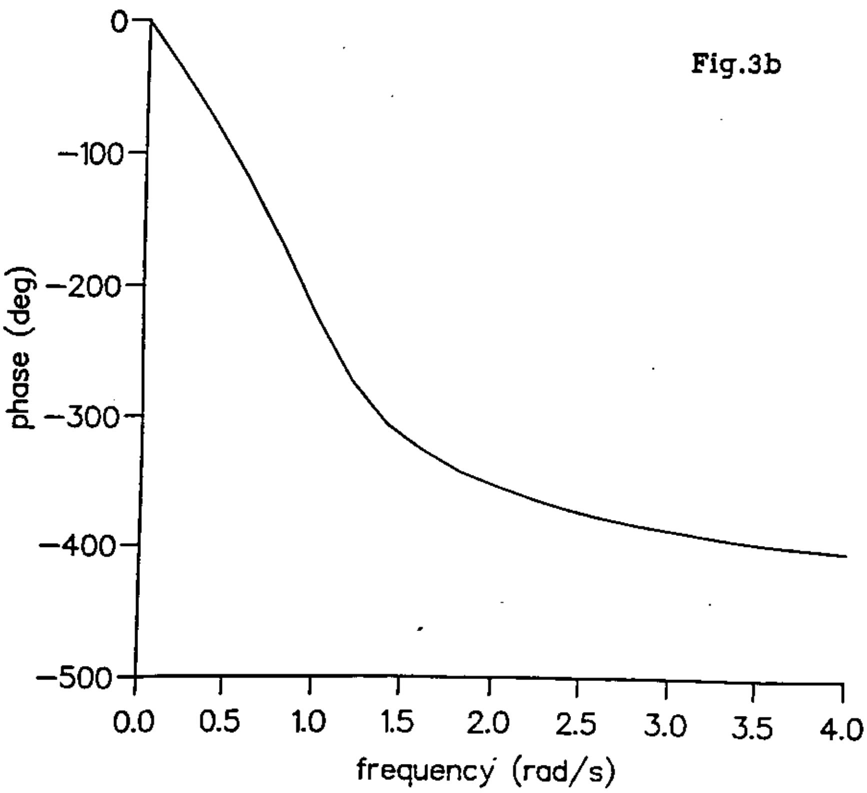


Fig.3b

Roll responses with filters

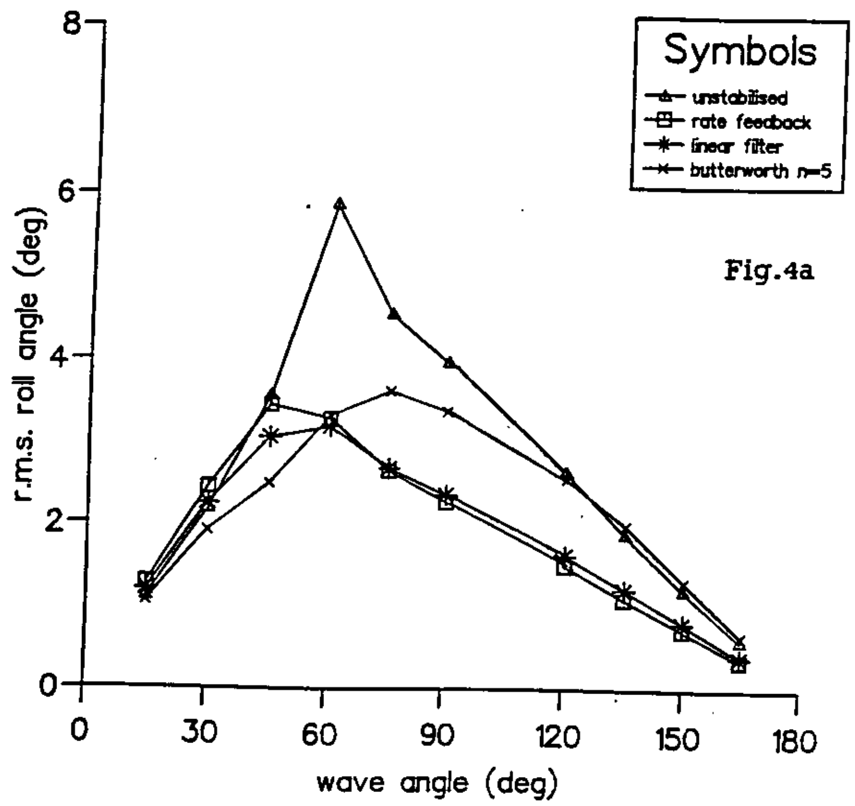


Fig.4a

Rudder rate responses with filters

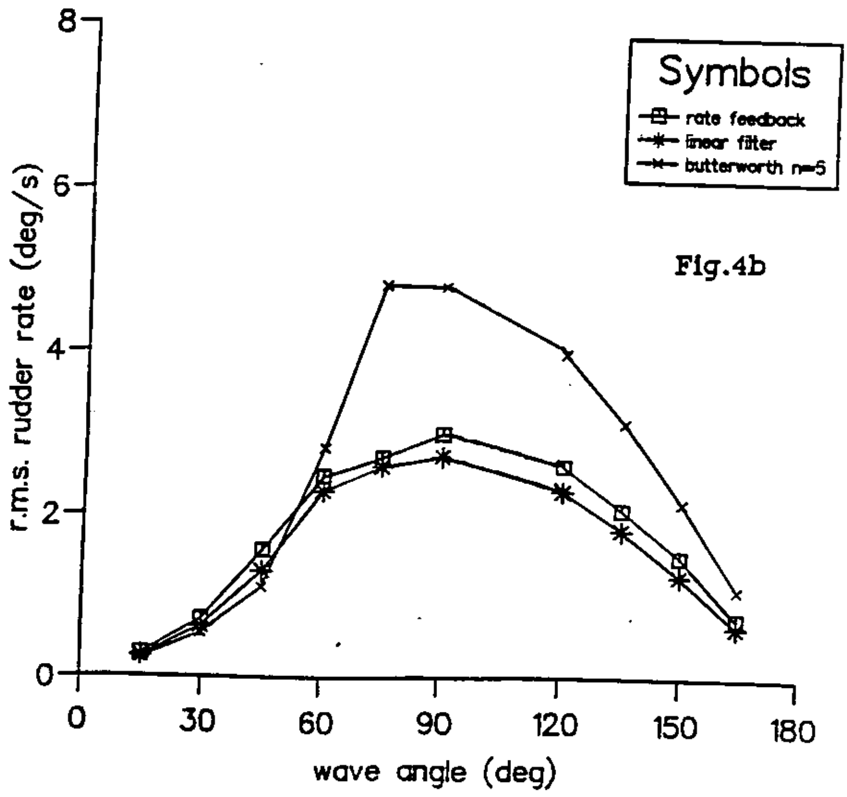


Fig.4b

roll response with butterworth n=5 filter
(varying the cut-Off frequency)

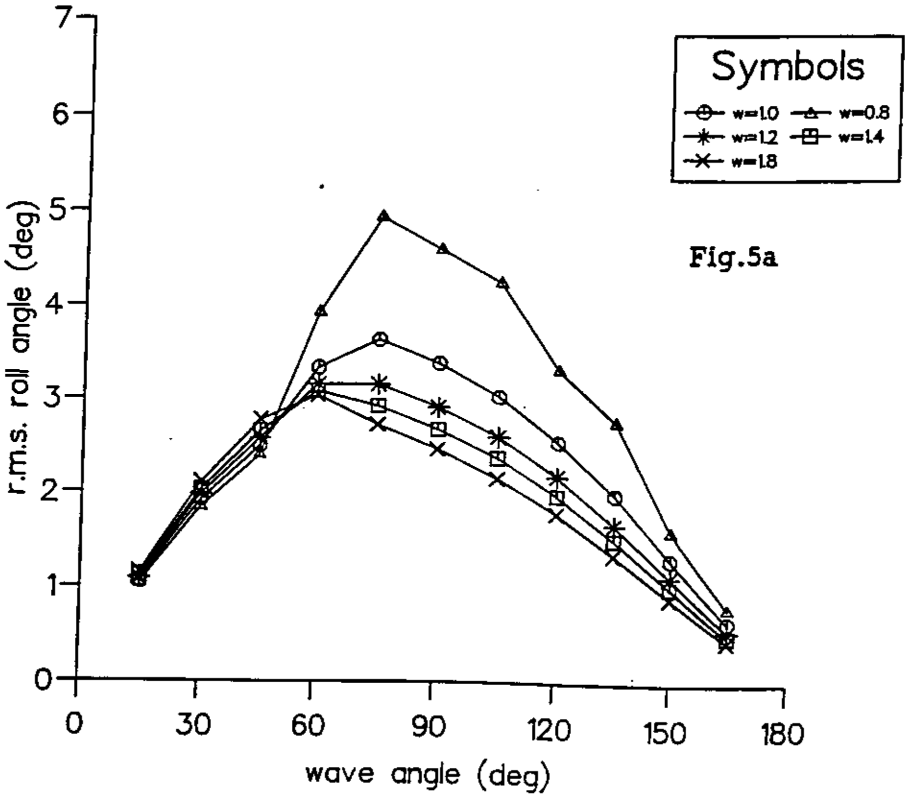


Fig.5a

rudder rate response with butterworth n=5 filter
(varying the cut-Off frequency)

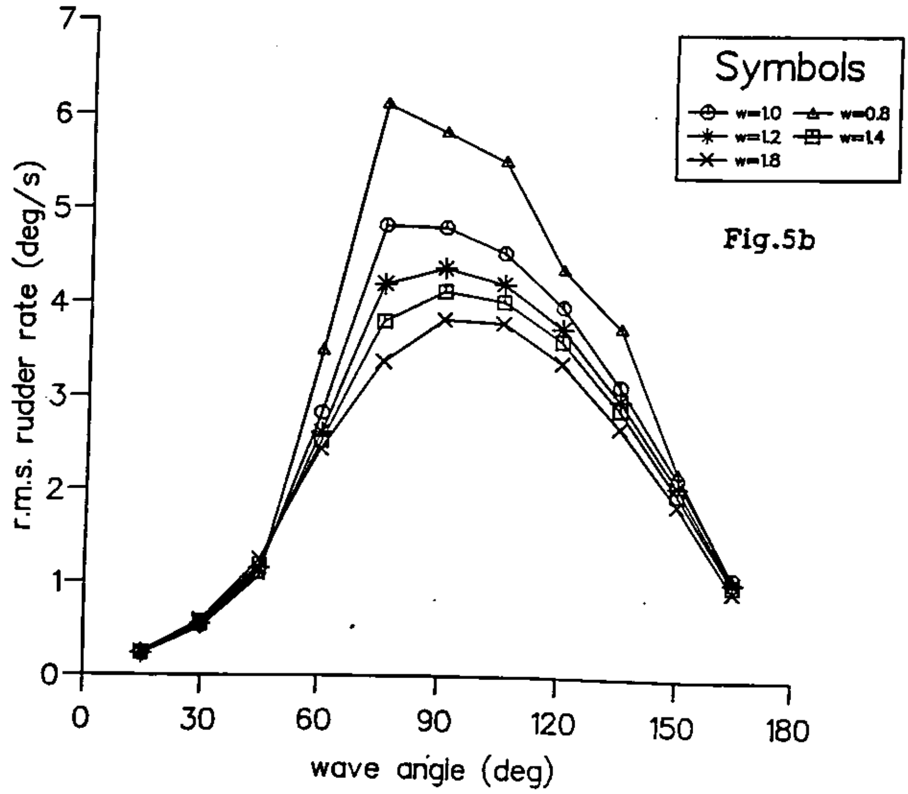
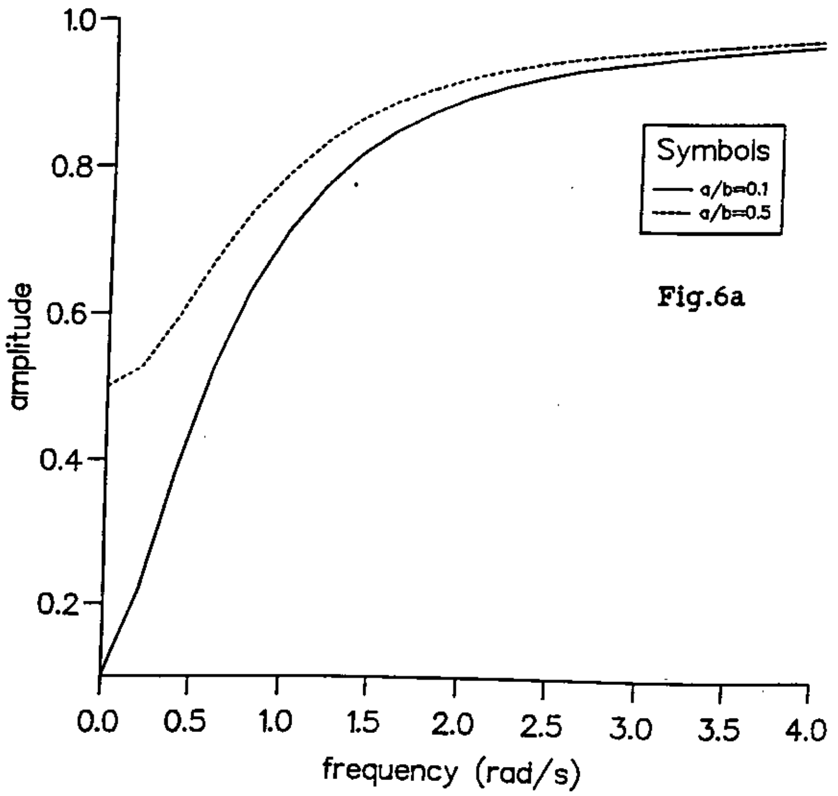
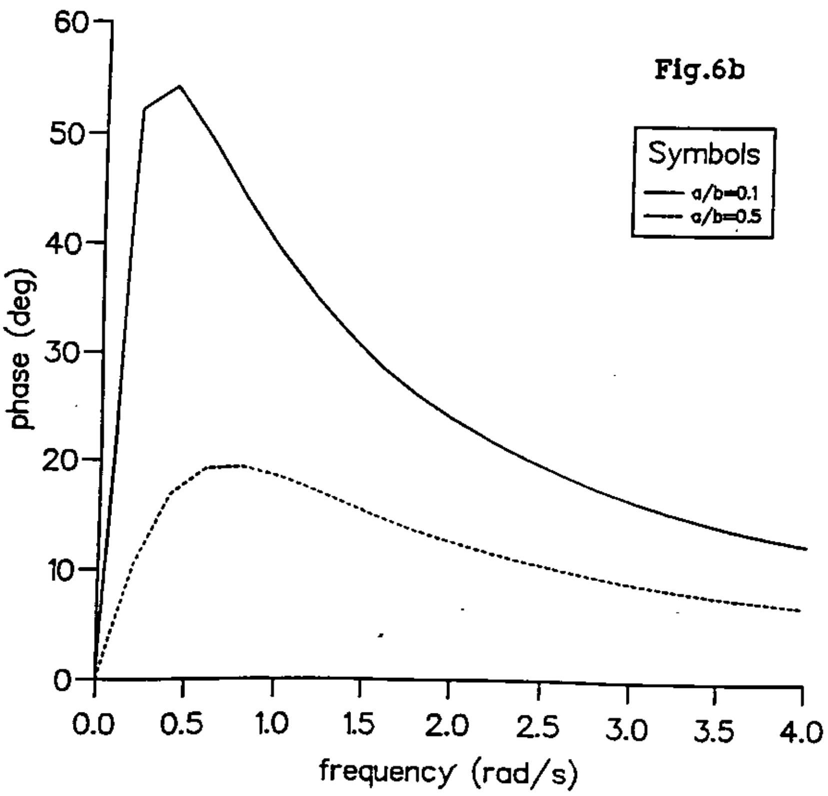


Fig.5b

Response of a lead-compensation filter



Response of a lead-compensation filter



All–pass filter response

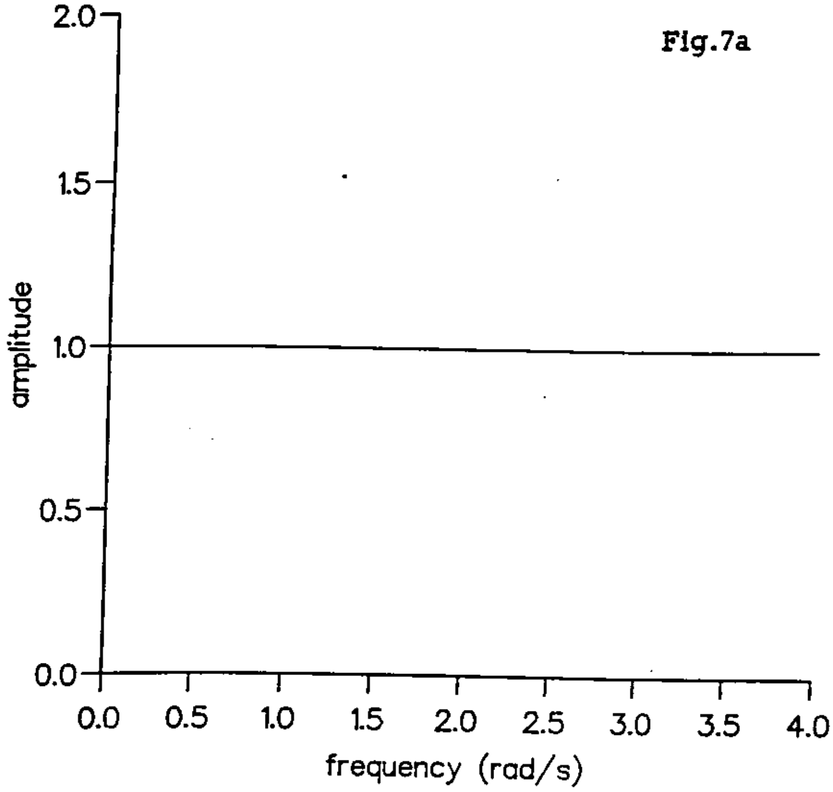


Fig.7a

All–pass filter response

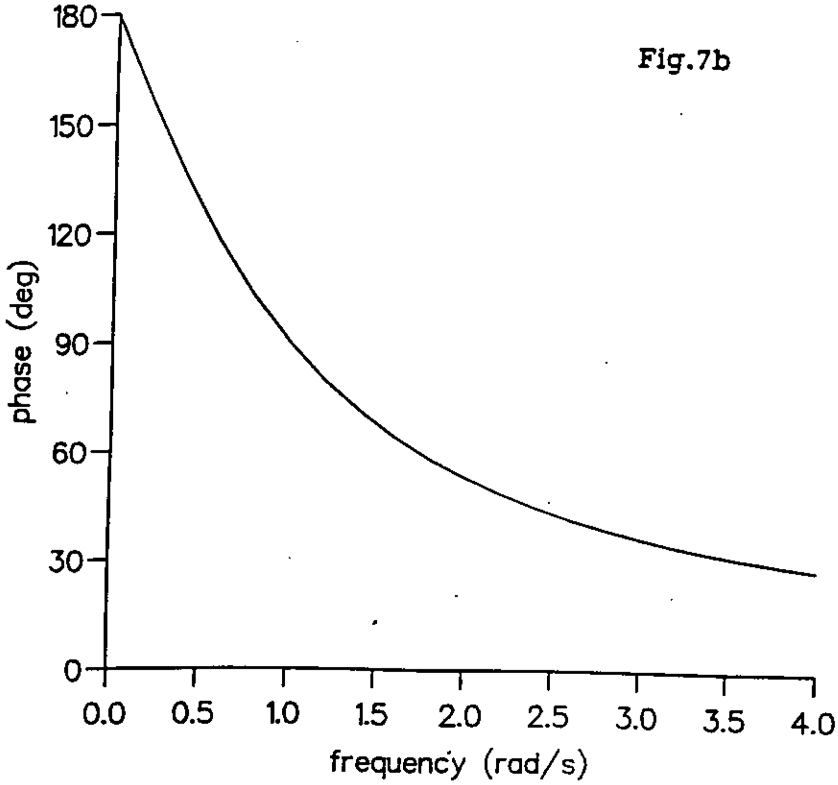
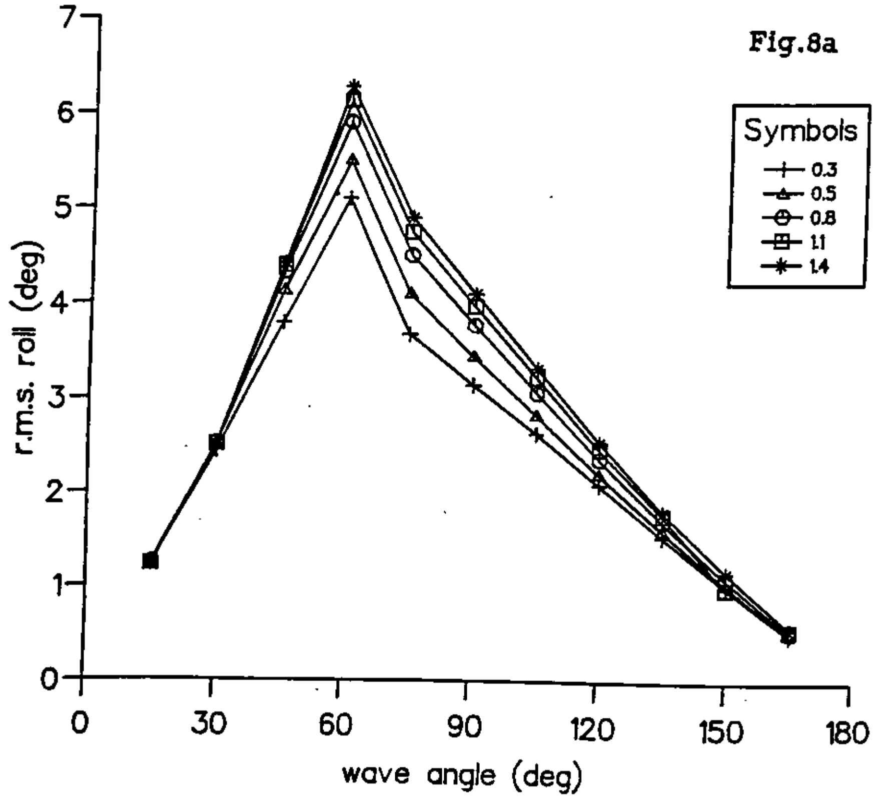
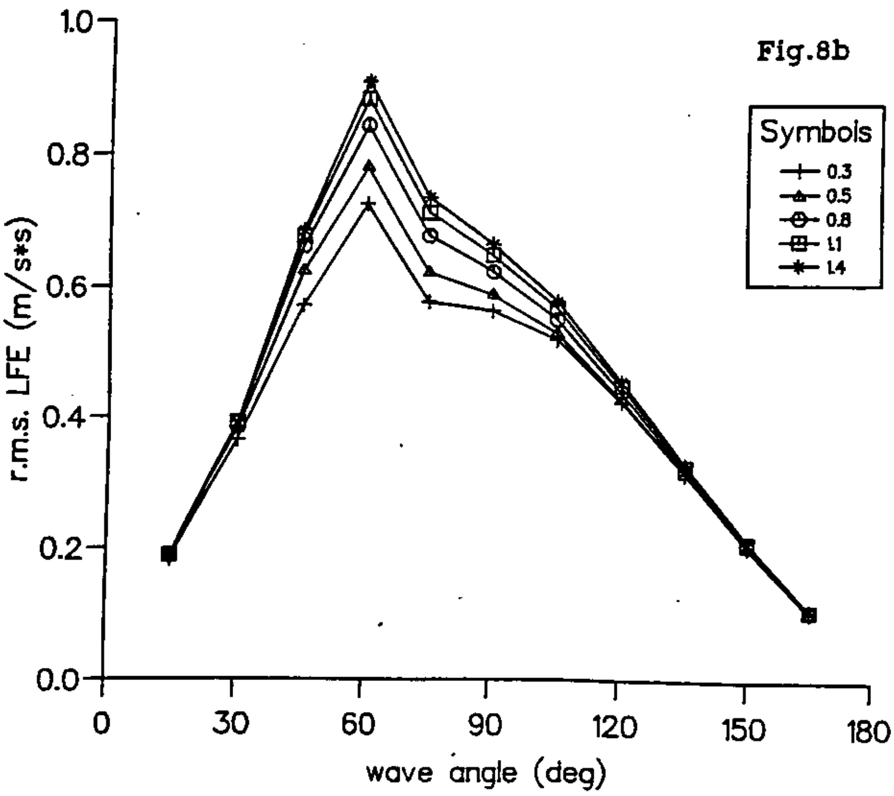


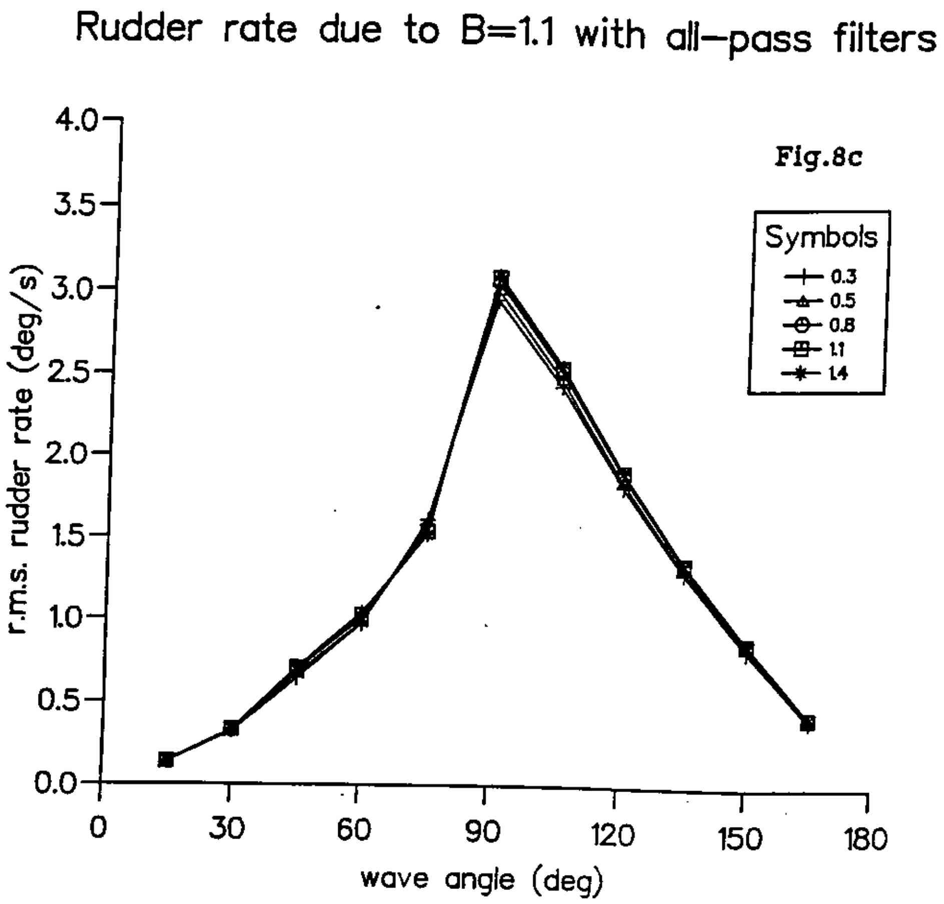
Fig.7b

Roll response due to B=1.1 with all-pass filters

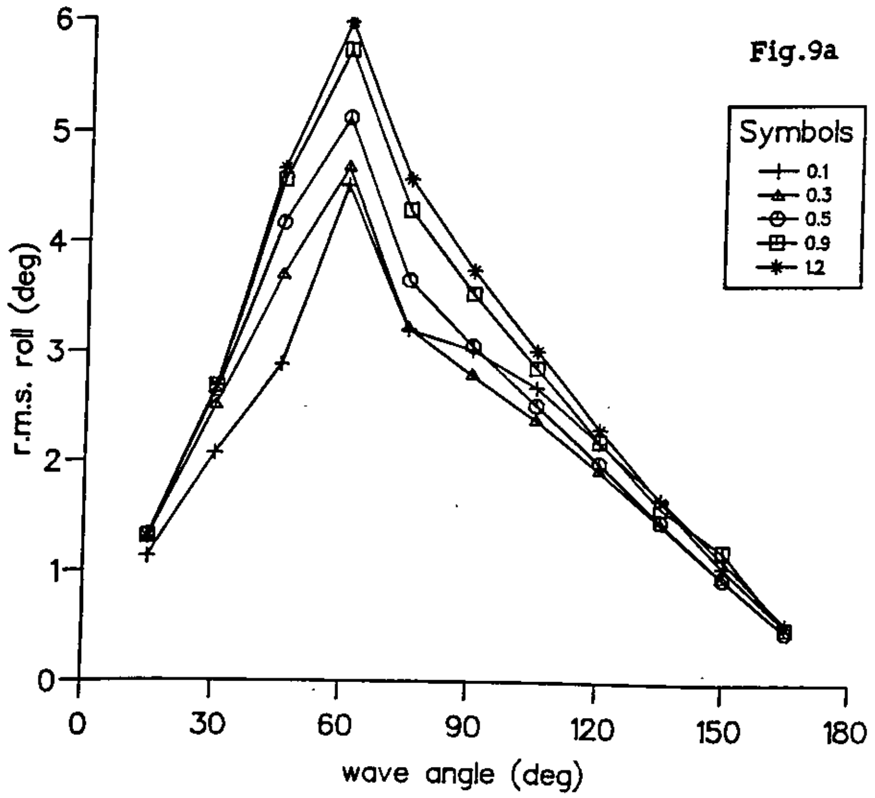


LFE response due to B=1.1 with all-pass filters

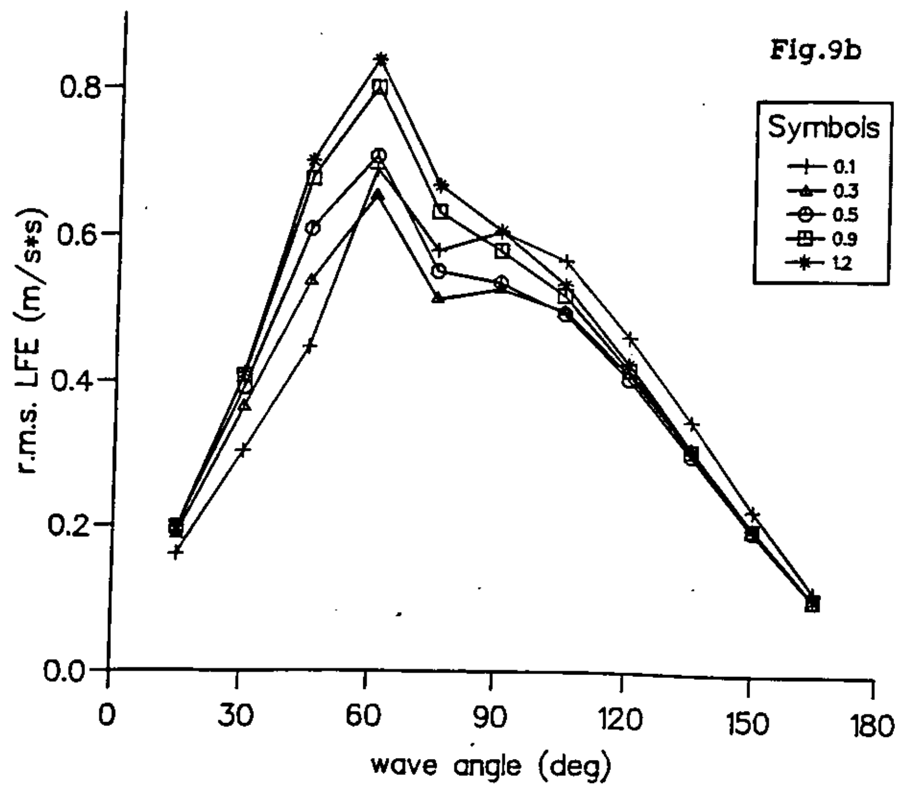



Rudder rate due to B=1.1 with all–pass filters
Fig.8c
Symbols
0.3
0.5
0.8
1.1
1.4
r.m.s. rudder rate (deg/s)
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
0
30
60
90
120
150
180
wave angle (deg)

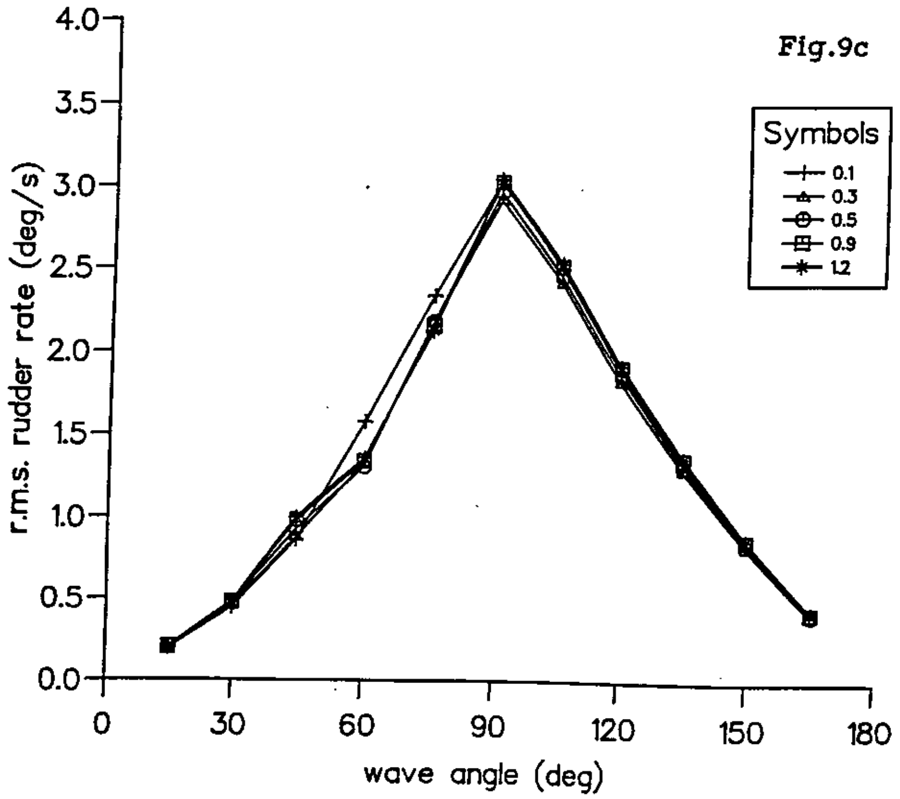
Roll response due to B=0.9 with all-pass filters



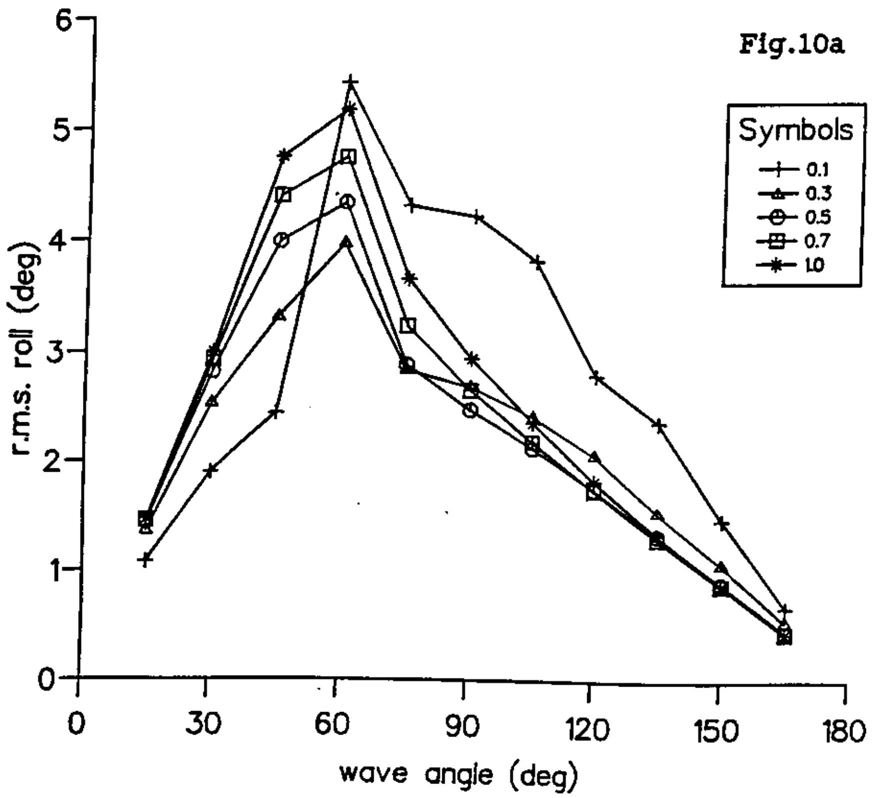
LFE response due to B=0.9 with all-pass filters



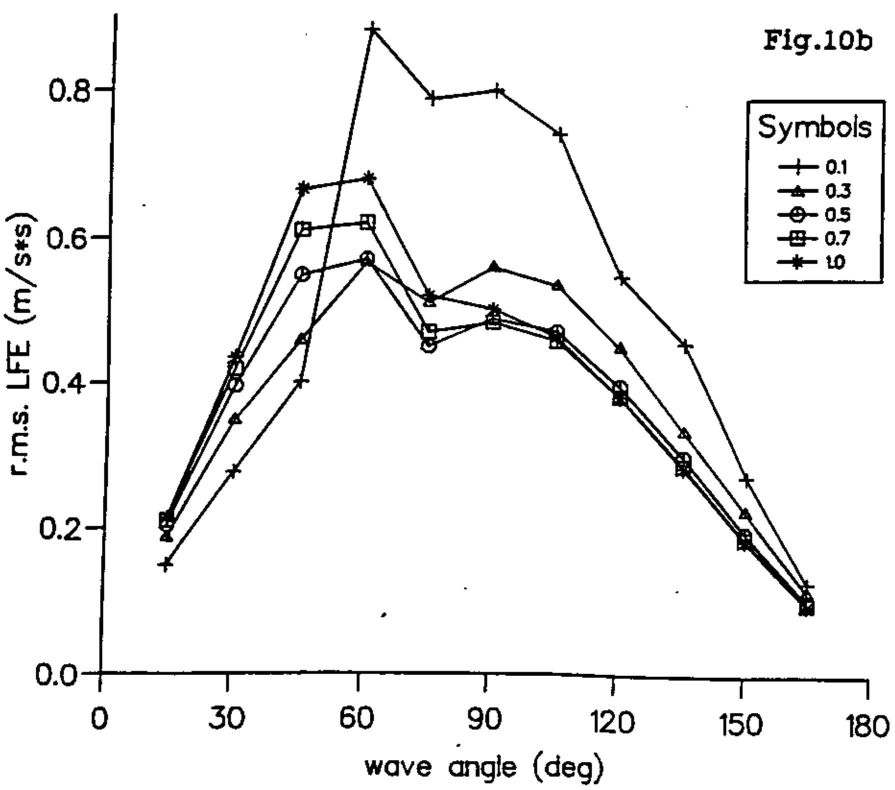
Rudder rate due to B=0.9 with all-pass filters



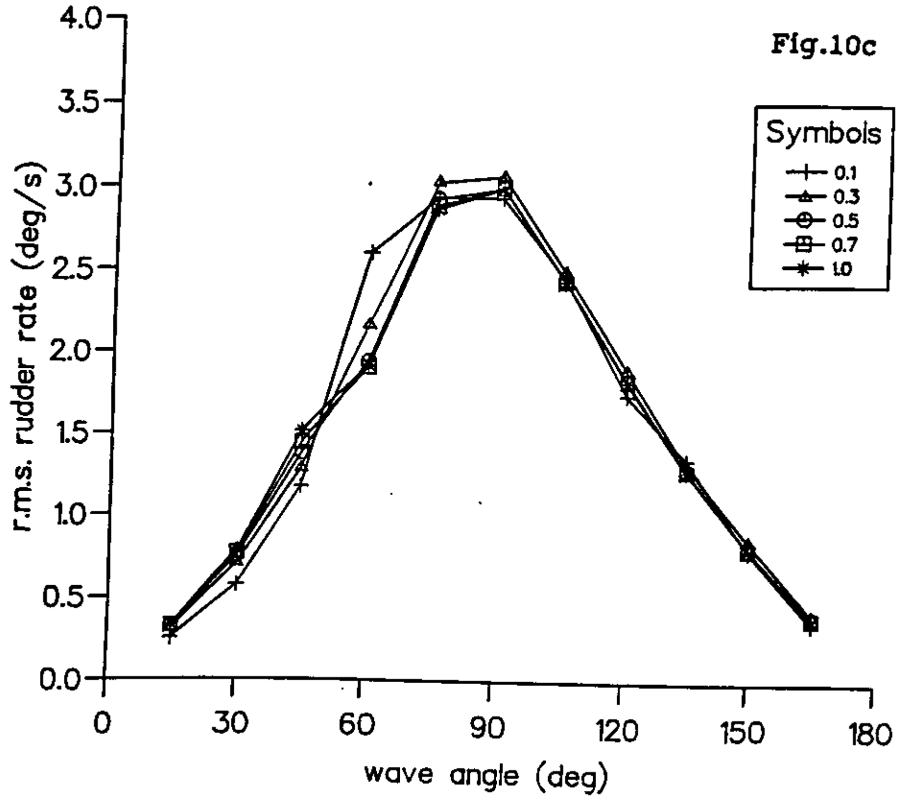
Roll response due to B=0.7 with all-pass filters



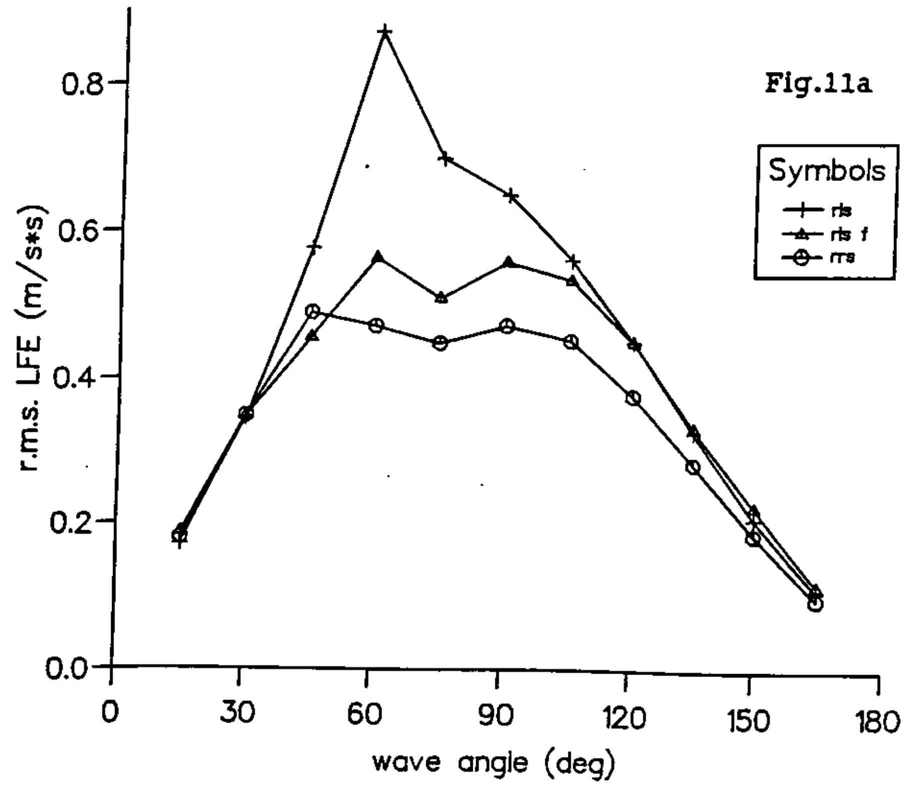
LFE response due to B=0.7 with all-pass filters



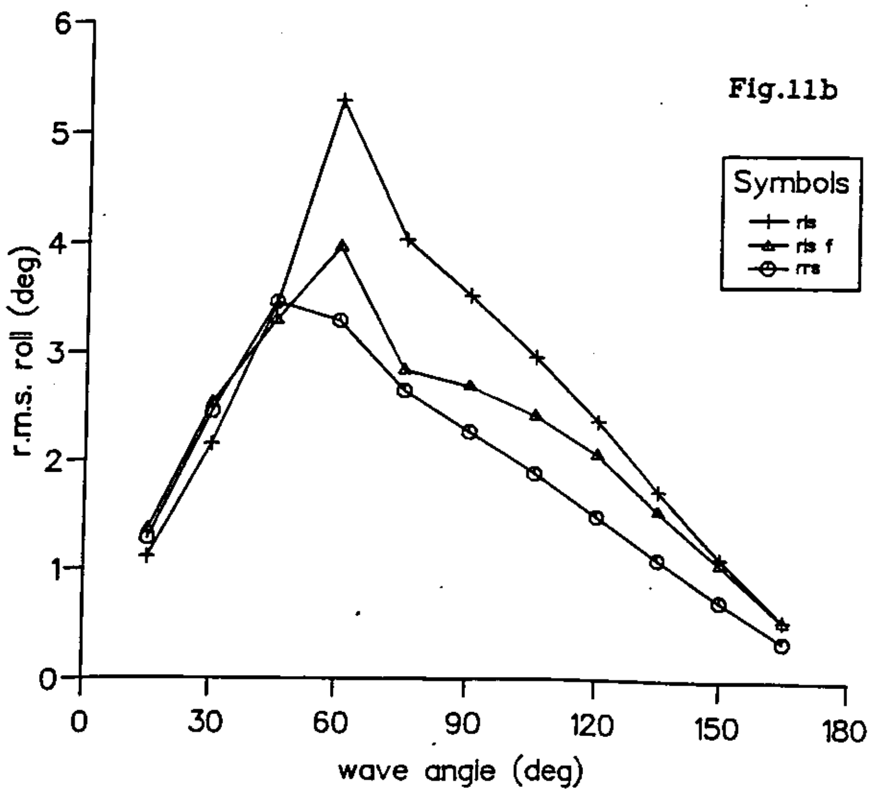
Rudder rate due to B=0.7 with all-pass filters



LFE response with and without filters



Roll response with and without filters



Roll response spectrum

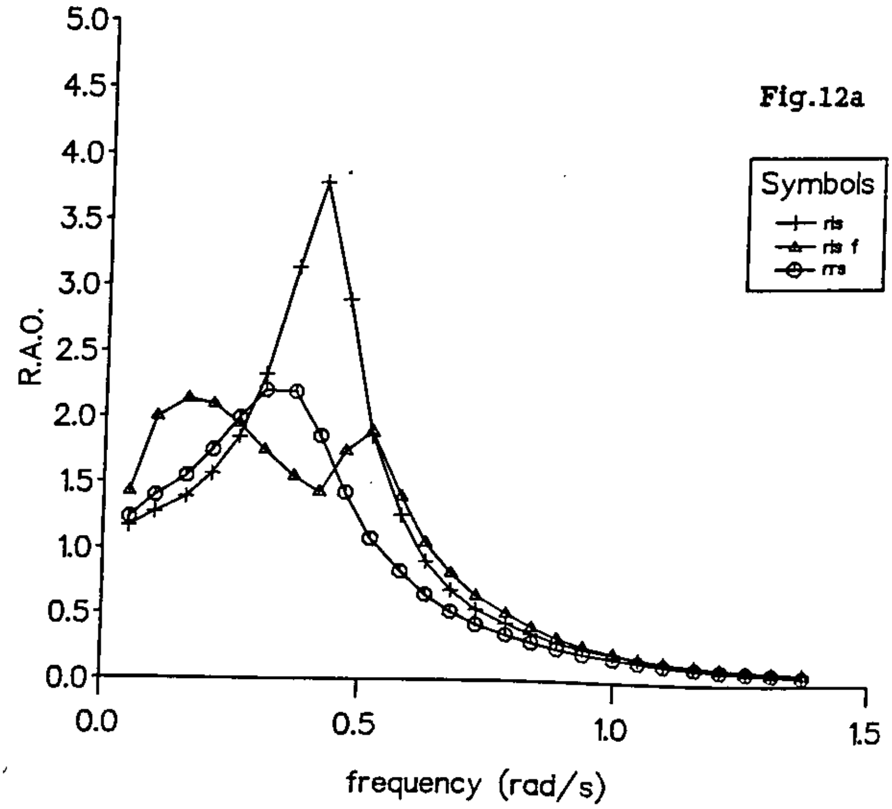


Fig.12a

Rudder rate response spectrum

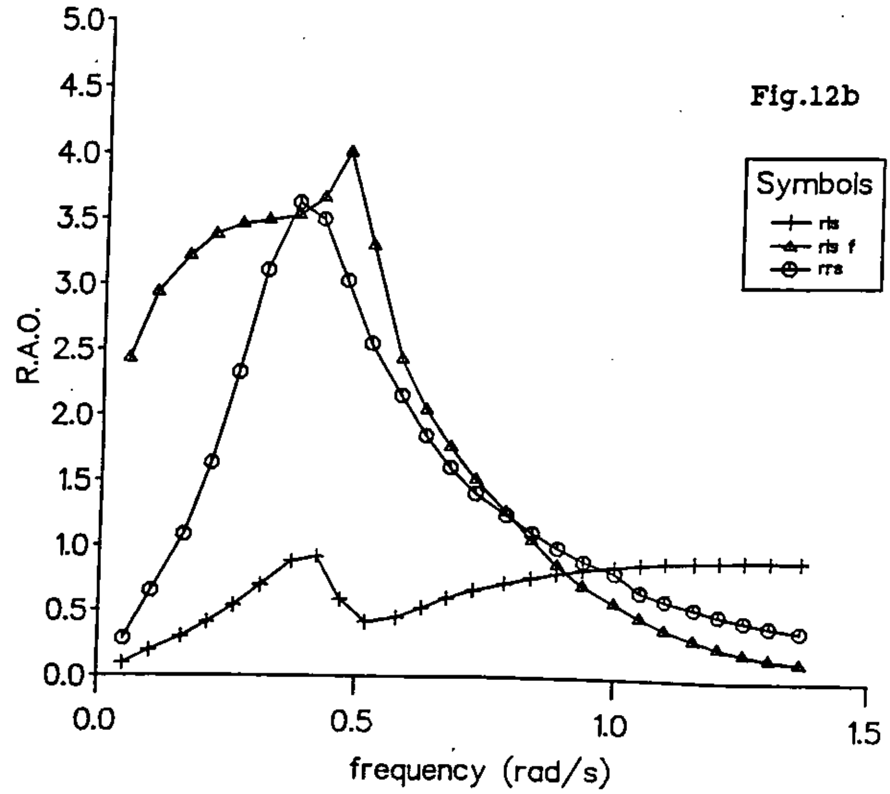
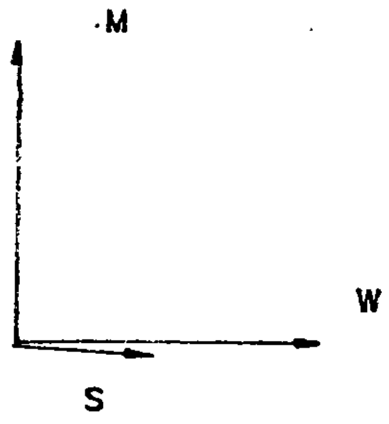


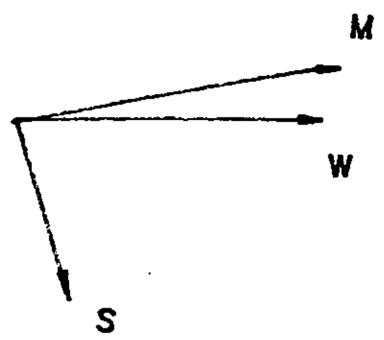
Fig.12b

Dig. 1a–1c Vector diagrams for simple roll rate feedback stabilisation

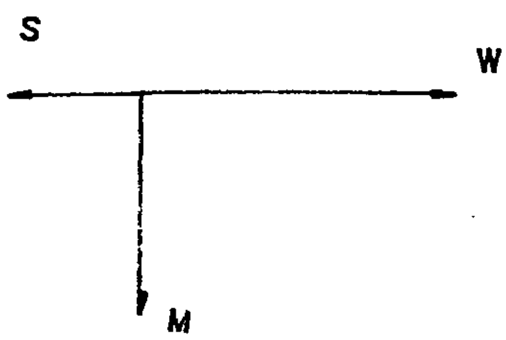
The vectors are not drawn to scale



(1a)
low frequency



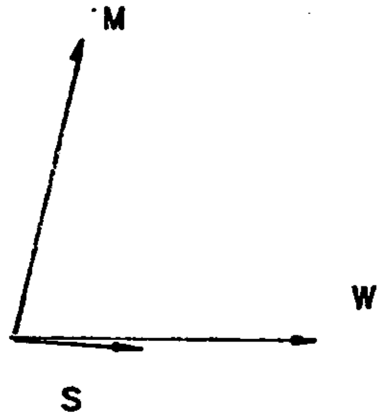
(1b)
near resonance



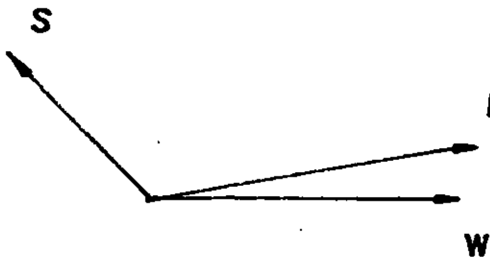
(1c)
high frequency

Dig. 2a–2c Vector diagrams for simple roll rate feedback with filter

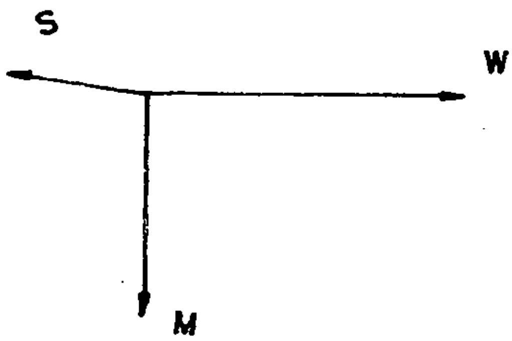
The vectors are not drawn to scale



(2a)
low frequency



(2b)
near resonance



(2c)
high frequency

PART II

A/ LFE part I trials

A1/ Introduction

A forced roll/LFE trial was conducted on the 15th of June 1992 onboard HMS Lancaster. This was the first part of the LFE stabilisation trials, in which data for the motion and fin response were collected. At the same time, the sea-trial also provided an opportunity for the trial team to gain some working experience with the new fin controller as well as the crew members onboard ship. The motion data derived have been used to compared with the numerical simulations in this part of the report.

A2/ Trials procedure

The trials consisted of three parts:
a/ the lurch test to locate the natural roll frequency,
b/ the motion response over a range of forced roll frequency, and
c/ the roll decay tests.

Originally, to perform the lurch test, two crew members were needed in the engine room to set the initial fin angles. Once a steady heel was achieved, the fins were discharged to their normal parked condition, which would excite the ship to roll at its roll natural frequency. However, due to the recording equipment failure and the demand on man power, this test procedure was abandoned. An alternative approach was tried. In this case, the ship was forced oscillated with the fins using square sine waves with the longest period (40 seconds) available in fin controller (CCU). Although the ship took longer than forty seconds to complete the decay motion, an estimate of the natural roll period should not be greatly affected. Unfortunately no useful data was derived from these tests. Due to the constraint of time, no further effort was made to obtain an estimation of the roll natural frequency.

The frequency response test was greatly enhanced by the built-in forced roll function of the CCU, which was found to be robust and easy to use. The procedure consisted of dialling in a sequence of requests on the console of the CCU, specifying the fin motions, then activating the requests and cancellations after a length of time. A typical example of the sequence is illustrated as follows:

31 20	forced roll period:	20 seconds (05-40 sec)
32 16	forced roll amplitude:	16 degrees (01-28 deg)
33 01	wave form:	sine
20 05	execute the requests	
20 11	cancel the requests	

(For the lurch test, the 33 01 was replaced by 33 03 for square waves.)
In general, more than ten cycles of motion were recorded for the subsequent analysis. As the resolution of the forced period parameter is one second, it is quite likely that the exact roll natural frequency cannot be selected.

For the roll decay tests, the ship was first forced oscillated near the natural roll frequency till a steady state was reached. The fins were then parked to allow the roll motion to die out. This was carried out with three different ship speeds, and the roll damping coefficients were derived.

In this trial, seven channels of time-history signal were recorded from the following devices: a three component accelerometer near the hangar, a lateral accelerometer near the workshop, the roll angle signal from the ship's gyro and the fin input/output signal from the CCU. The positions of the accelerometers are given in table 1 below. From these transducer time-histories, the amplitude and phase of the signals were derived. During each run, the rudder was amidships and the ship speed was 15 knots except for the case c above. The displacement of the vessel was 4248 tonnes.

Table 1 : Accelerometer positions

	X (m) from midship	Y(m) from C.L.	Z(m) above USK
3-component accelerometer (hangar)	21.44	0.35	10.30
lateral accelerometer (workshop)	4.92	-0.34	8.51

A3/ Results and Discussions

1/ roll data

The roll amplitude and phase comparisons are given in fig. a1 and a2. In fig.a1, there is a difference of about 7% in the natural roll frequencies and about 15-20% difference in the amplitude response near the resonance frequency between the predicted and the trials data. Apart from possible measurement errors, these differences could be attributed to the GM value assumed in the simulation, as the GM during the trial was not known. In comparison to the GM, these differences should be less sensitive to the 'errors' in the roll damping and the fin lift coefficients of the numerical model. Therefore, if the assumed GM value is higher than the true value, the natural frequency would be expected to be higher and the amplitude response lower, as is the case in fig. a1.

The phase variation with frequency in fig.a2 between the two sets of data follows a very similar trend. The relatively large difference at low frequencies should not be taken too seriously as these measurements are very sensitive to the sea condition, which was not totally calm. Also, being at night, no effort was made to run the ship in head sea conditions as the sea state was less than two. The main point to note is that at low frequencies, both sets of data show non-zero phase values. This suggests the presence of some motion interaction effects, which are probably caused by the sway forces of the fins.

On the whole, both the roll amplitude and phase response comparisons show good agreements. Despite having only one pair of fins, the forced roll response of this ship shows that this design is as effective as some of the older designs, such as those used in the Leander (wide beam) class, which were installed with two pairs of fins.

2/ lfe data

The LFE amplitude and phase response are presented in fig. a3 and a4. The overall trends in both cases are similar to the corresponding roll data. In fig.a3, the LFE amplitudes at the workshop (trial W) are slightly lower than those at the hangar (trial H). The difference is due to the difference of about 3m in their vertical positions, which gives rise to different roll accelerations. The measured LFE is predominantly due to the roll angle equivalent acceleration, which accounted for about 90% of the measurements near the roll natural frequency.

Apart from a shift of pi, the LFE phase (fig.a4) and the roll phase (fig.a1) are almost identical. As the accelerometer was calibrated with a negative signal for a starboard roll, the pi shift is evident. The acceleration terms in the LFE signal did produce a difference of a few degrees to the roll case both in the trials data and the numerical simulation. This difference is more apparent at the low frequencies. It should be pointed out that, in deriving these phase responses, a difference of pi was found in cases using the measured data.

In fig.a5, a comparison of the 'sway' accelerations calculated from two different methods is shown: S(t) is by subtracting the roll equivalent accelerations from the LFE in the time histories, from which the r.m.s. value is derived, whilst S(w) is the difference in their r.m.s values without phase corrections. Although the magnitude of these data is fairly small, their comparisons should still give some indication of their credibility. In most cases the S(t) data differ from S(w), which show more scatter. More importantly, the S(w) approach gives negative values at high and low frequency. Therefore, to derive the r.m.s LFE data, as shown in Monk's paper (TRINA 1987), by simply adding the r.m.s roll and lateral acceleration terms together without proper phasing would lead to questionable results.

3/ fin data

Over the range of frequency tested, the magnitude of the fin angles were almost the same as the demanded angles of 16 degrees. With the highest frequency used (1.23 Hz), this demanded amplitude was too high, which resulted in very sluggish performance. In this case, a fin demand of 13 degrees was used. The phase lag incurred by the CCU and the hydraulic system on the fins is shown in fig.a6. To simulate part of this phase response, the hydraulic system of a Type 22 frigate was assumed i.e. $a1=1.0$, $a2=0.318$ and $a3=0.0253$. To account for the CCU system, the following data has been used in order to match the measured data: $a1=1$, $a2=0.1$ and $a3=0.05$. The sum of these phase angles formed the fitted line in the figure.

4/ roll decay tests

Three speeds were used for these tests: 15, 18 and 22 knots. Unfortunately, the time-history for the 22 knots case was not good enough for analysis. The roll decay coefficients for the 15 and 18 knots cases were 0.126 and 0.136 respectively. The corresponding estimated natural roll frequencies were 0.616 and 0.61 Hz. Judging from these figures, the motions should not differ greatly within this speed range.

B/ LFE Signal Conditioning

B1/ Design history and background

The original specifications for the LFE signal to the CCU (quoting from Brown Brother's proposal) are as follows: a synchro input would be used and the signal should be about 14 bits. This was all the design information that was provided by Brown Brothers Ltd. At a later stage, some additional information about the components required for generating such a signal were also given during discussions over the telephone, but on the whole, Brown Brothers was rather reluctant to be very specific. This lack of information has introduced additional difficulties to the design of the LFE signal conditioning. Nonetheless, a design was developed which was outlined in the last report.

During the visit to Brown Brothers, the hardware was briefly put to test. It was found that the fin response to the accelerometer signal was very stiff. Apart from this, no other undersirable effects were found. Two main reasons have been put forward to explain the stiff response. Firstly, the resolution of the system was restricted by the 14 bit ADC chip. The CCU has a resolution of 10 degrees per revolution and a maximum range of +/- 30 degrees. This means that a 16 bit ADC would be more appropiate as it allows for more than two revolutions in the synchro output. Secondly, the CCU is only tuned for a frequency around the roll natural frequency and a high frequency input signal would saturate the response. Therefore, moving the accelerometer by hand as was done during the test could have caused such a stiff response. Apart from these two points, some additional information about the CCU system on the whole was also deduced from the discussions during the test. To satisfy these additional criteria, modifications to the original design were required.

B2/ Design modifications

Fig.b1 illustrate the signal path for LFE. Whilst the overall signal path is the same as the one described in the last report, modifications to the individual circuits have been made to suit the 'new requirements'. A brief account of these modifications are outlined below.

The band-pass filter in fig.b2 has a bandwidth between 0.05 to 1 hz as compared to a bandwidth of 0.002 to 10 hz in the old circuit. This reduction in bandwidth is to eliminate further the unwanted signals, especially those at high frequency. However, adjustments may be needed if the phase lag associated with this new bandwidth was found to be higher than expected. After filtering, the voltage of the signal has been adjusted to about 2.7 volts, which corresponds to about 31 degree/rev in the Digital to Synchro (D to S) output. Fine adjustment would be needed to tune it down to 30 degree/ rev with the CCU. The voltage gain has therefore been made adjustable for this purpose. The time base circuit in fig.b3 is effectively the same as before. However, to accommodate the new 16 bit ADC chip, some fine tuning of the circuit has been carried out. The conversion frequency is still about ten times faster than the CCU.

The main modification of the whole LFE unit is to replace the 14 bit with a 16 bit ADC. The sole function of this 16 bit ADC chip is to provide two extra bits, which work as counters for a 14 bit signal. These two extra bits are not actually used in the main circuit. By allowing the digital signal to clock up twice, they can provide up to eight revolutions in the D to S output, whilst the digital signal is still of 14 bit accuracy. The new circuit for this chip is given in fig.b4. As the CCU only has a range of six revolutions corresponding to +/- 30 degrees of roll, the two extra revolutions from the 16 bit chip would be ignored, should the roll amplitude be higher than 30 degrees.

A wiring diagram for the different circuit boards is given in fig.b5. A test circuit board has been included in the unit to faciliate future testings. The circuit boards are housed in an aluminium rack as one complete unit, with one cable coming in from the accelerometer and one cable going out to the CCU. The cabling for the accelerometer and the D to S is shown in fig.b6.

At this stage, the whole unit has been tested and trimmed to the requirements available. Bench tests with the CCU will be needed to verify these modifications. Adjustments may be required to fine tune the unit for the sea-trials.

C/ CCU tests

A mechanical oscillator, which serves as a roll table, has been designed and constructed. The main purpose of this oscillator is to provide a means to generate known signals at a range of frequencies and amplitudes with a transducer, such as a potentiometer or an accelerometer, suitable for the CCU. This would allow the following objectives to be met in preparation for the sea-trials:

a/ check the modifications described in the last section thoroughly over the range of frequencies of interest,

b/ establish a procedure for setting the CCU gains for the sea-trial,

c/ assess the gains derived for LFE stabilisation,

d/ compare the fin response due to simulated roll and LFE signals.

The design of the oscillator was based on a scotch yoke mechanism, which has been used widely for producing sinusodal oscillations for ship motion studies. However, in order to simplify the driving mechanism, a tangent oscillator was adpoted. Therefore, above ten degrees of amplitude of oscillation, the generated signal would depart progressively from an exact sine wave. The oscillator proposed should have a frequency range between 0.2 to 0.05 Hz (5 - 20 sec) and an amplitude range between about 4 to 30 degrees. A d.c. motor has been used as the prime mover in the oscillator. The electronic circuits required for controlling the frequency of excitation and for interfacing the transducer signals to the CCU have been designed and constructed as an integral part of the mechanism.

An overall view of the mechanical design is given in fig.c1. This mechanism has been constructed and preliminary tests show that it performs well between oscillation periods of 5 to 16 seconds. Below 16 seconds, the mechanism does not run smoothly as there are slight mis-alignments in the set-up. In time, the system should be able to work down to 20 seconds after a good run-in. Also, at these low frequencies, the motor torque is low. Therefore, a simple voltage controller has been designed to cope with the variation in the loading. This is accomplished by velocity feedback from an optical encoder mounted on the drive-shaft of the mechanism. The circuit design is given in. fig.c2. To facilitate the selection and the determination of the test period, a display unit for the oscillation period has been included.

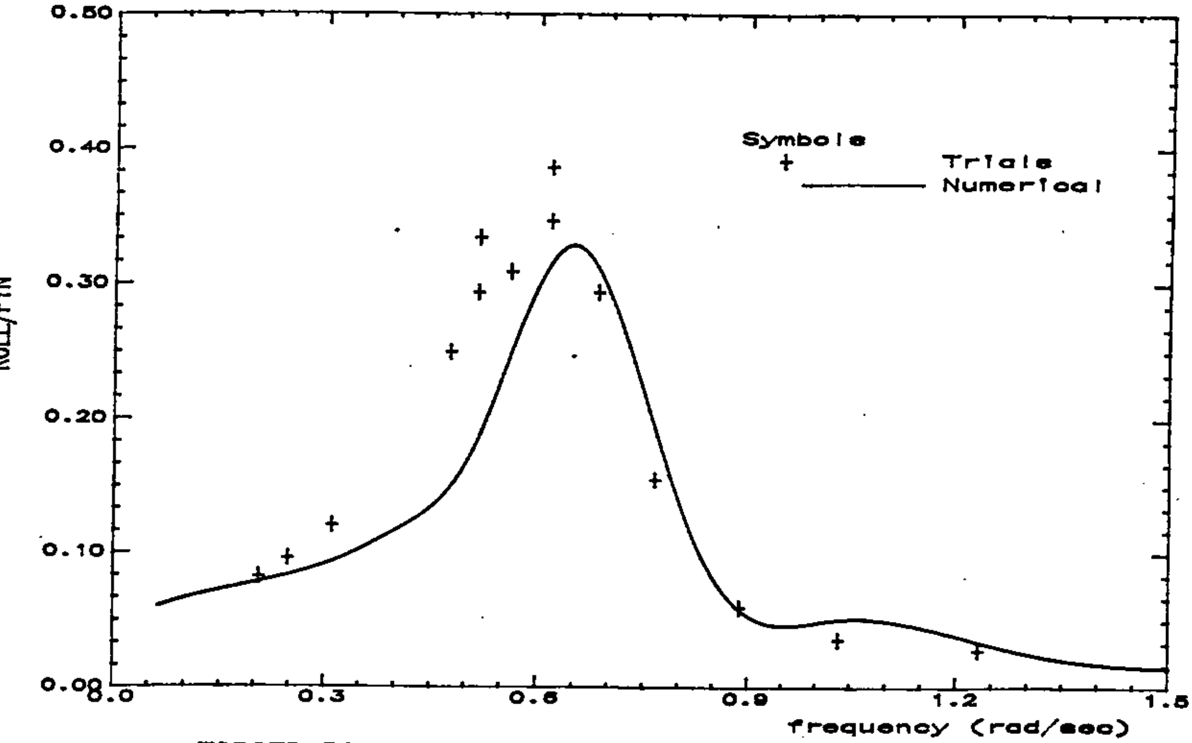
The circuit diagram for this unit is shown in fig.c3.

To simulate the test signals for the CCU, an accelerometer and a rotatry potentiometer are mounted on the rotating arm of the mechanism. Signals from these transduers are fed into the LFE unit, which is interfaced to the CCU. By varying the amplitude and frequency of the oscillator, the fin responses due to these two input signals are then compared, which would give an indication of the LFE stabilisation stategy. Likewise, the most desirable gain setting could be selected for LFE.

For item d mentioned above, it was assumed that the 'sway' terms in the LFE would be about 30% of the roll angle and about 90 degrees forward in phase. This simulated LFE signal would be more representative of the LFE in a seaway than a pure sinusodal signal. The phase angle and the signal amplitude can also be varied to give different combinations of a simulated LFE demand. The circuit diagram for this phasing is given in fig.c4, which is basically a modified all-pass filter. The acceleromter signal from the oscillator will be fed into this circuit first before connecting to the CCU.

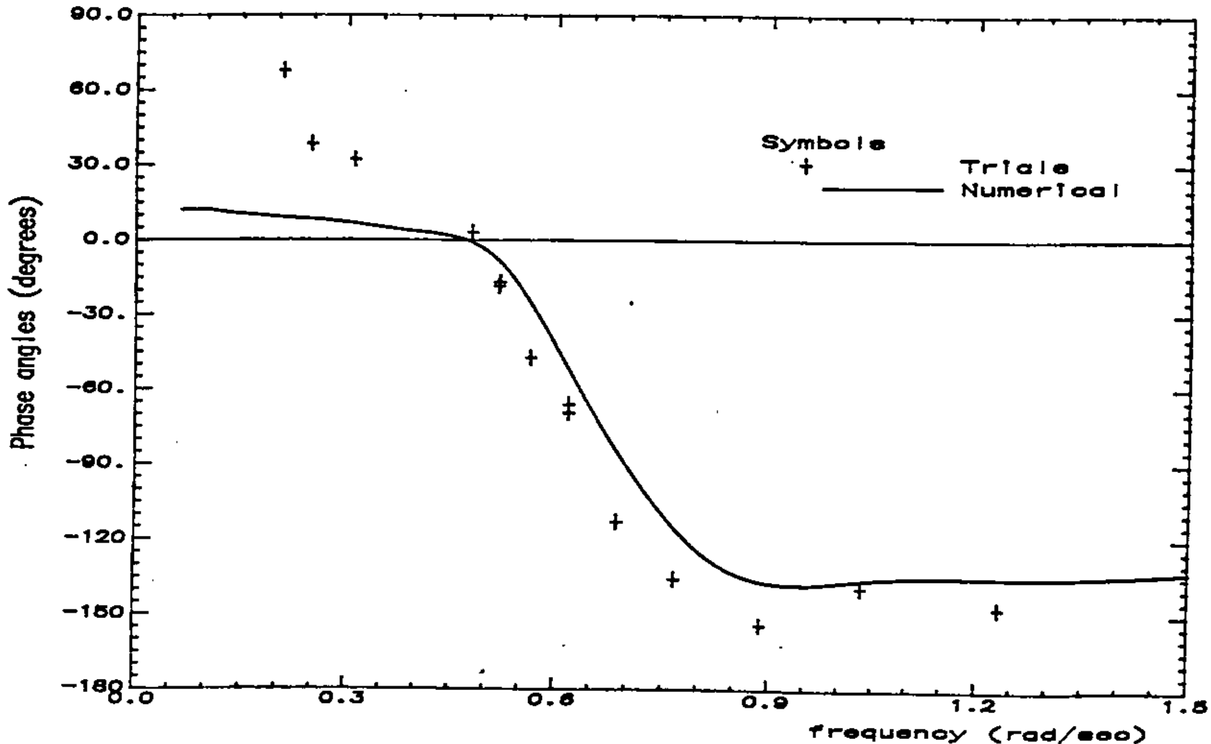
The above plan shall be carried out as soon as the CCU at Sultan is available for testing.

Fig. a1



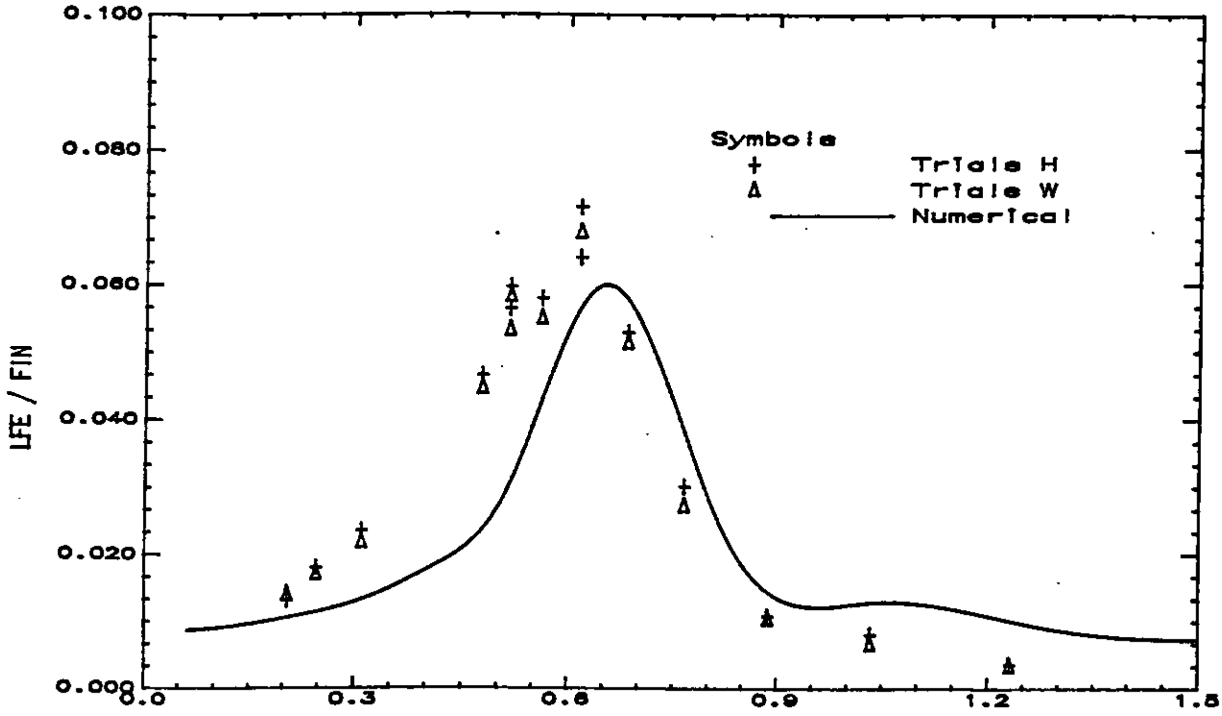
FORCED ROLL TRIALS AT 15 KTS : Roll Response

Fig. a2



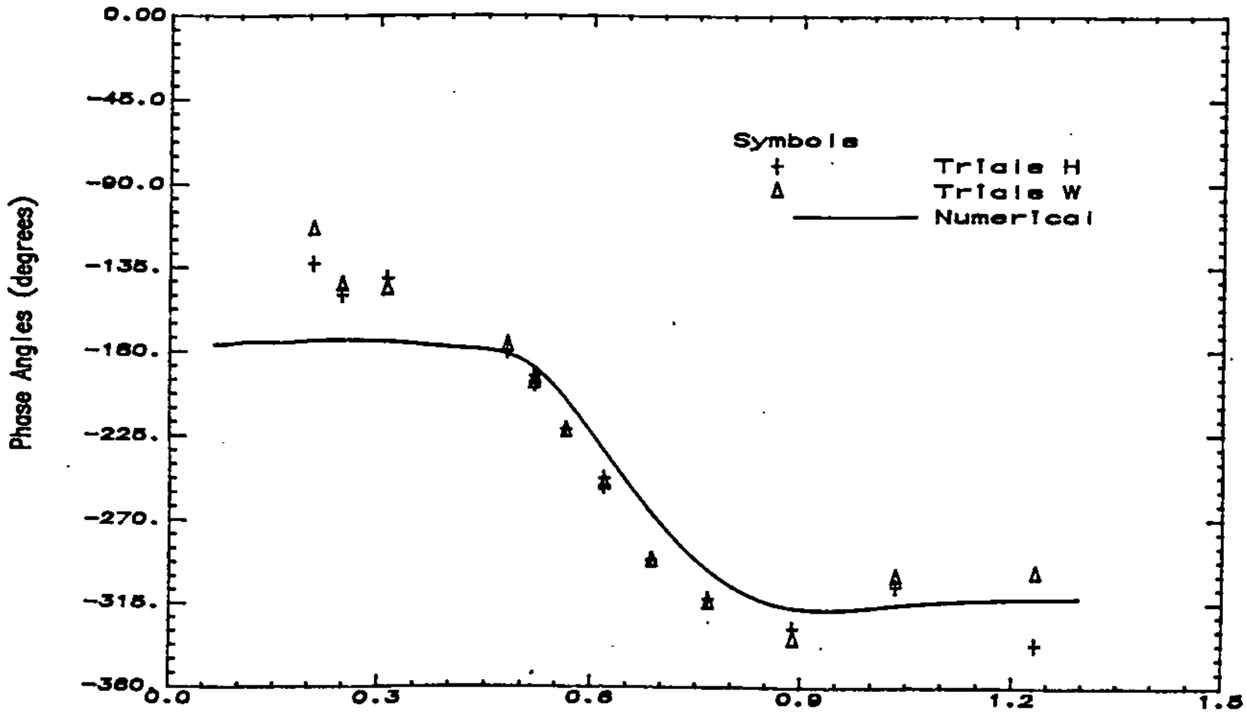
FORCED ROLL TRIALS AT 15 KTS : Roll Response

Fig. a3



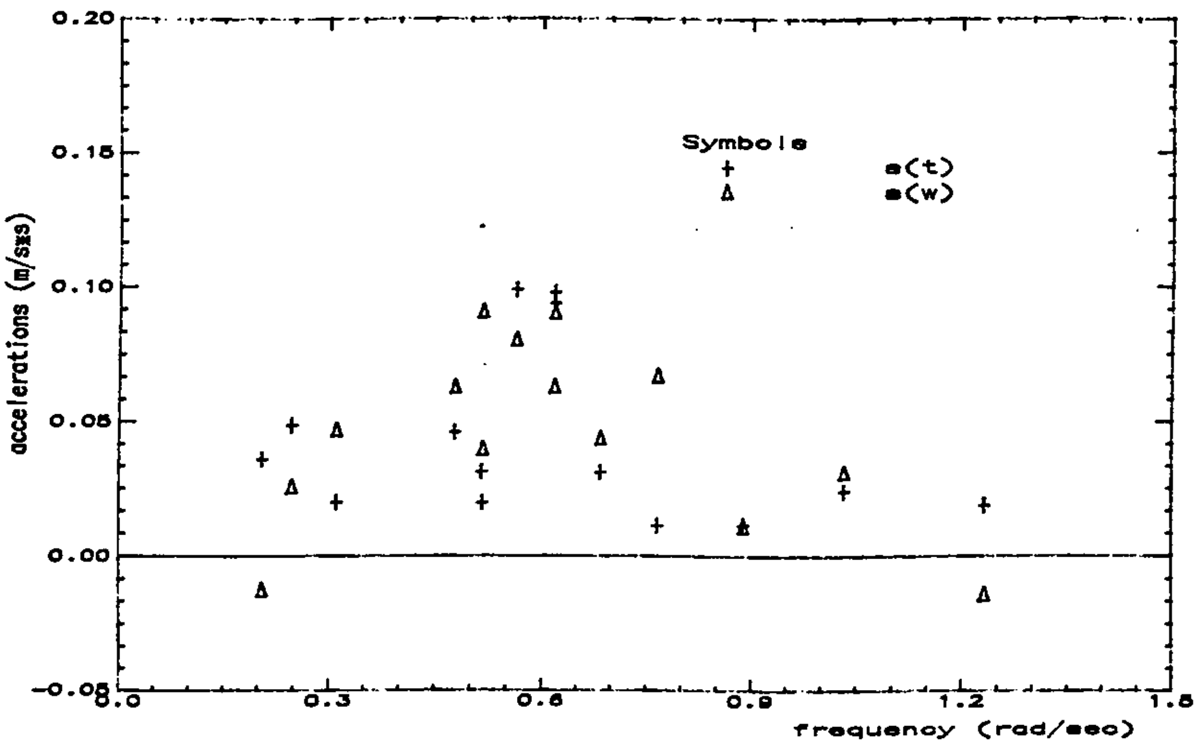
FORCED ROLL TRIALS AT 15 KTS : LFE Response

Fig. a4



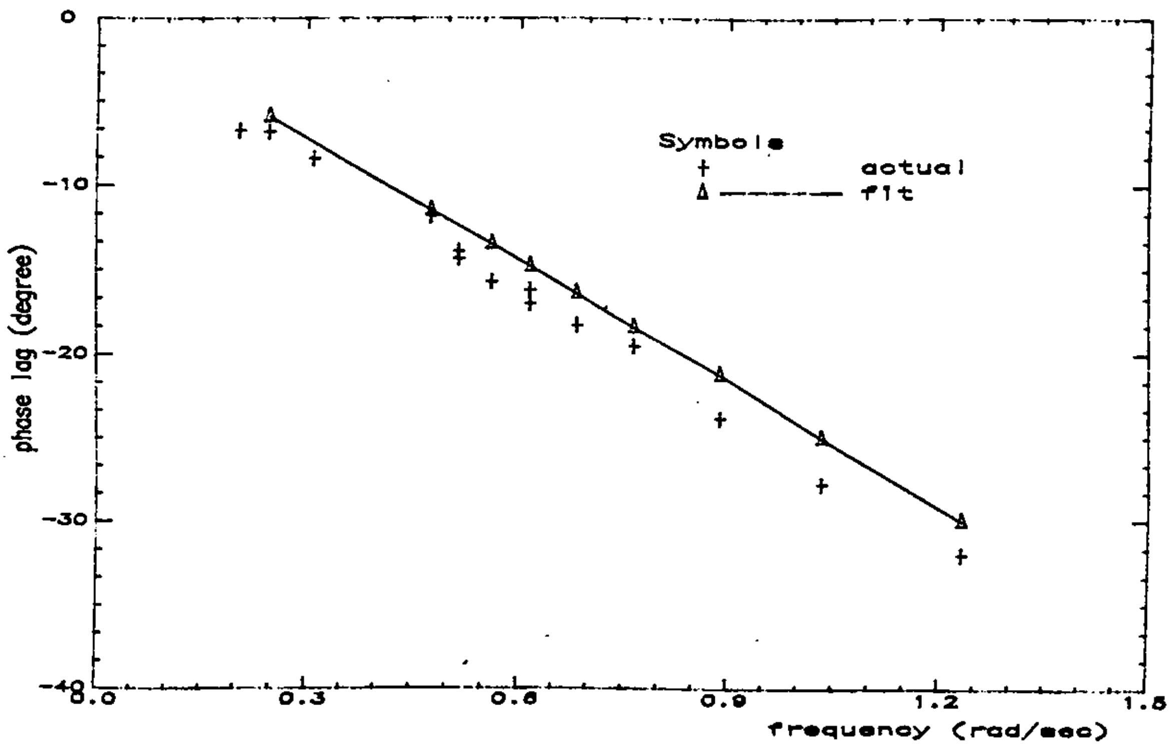
FORCED ROLL TRIALS AT 15 KTS : LFE Response

Fig. a5



FORCED ROLL TRIALS AT 15 KTS : 'Sway' term

Fig. a6



FORCED ROLL TRIALS AT 15 KTS : Fin Phase Lag

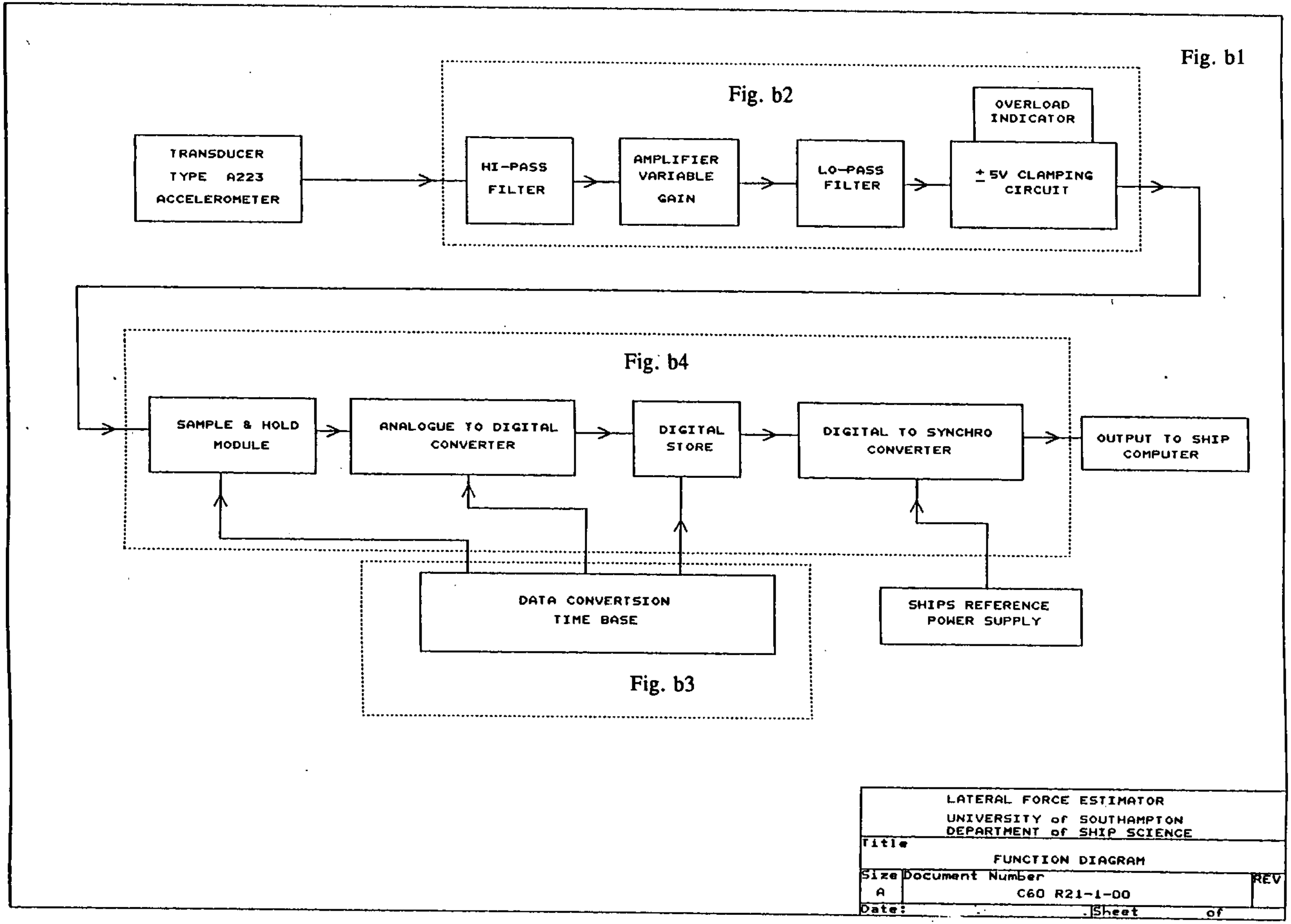


Fig. b1
Fig. b2
TRANSDUCER
TYPE A223
ACCELEROMETER
HI-PASS
FILTER
AMPLIFIER
VARIABLE
GAIN
LO-PASS
FILTER
OVERLOAD
INDICATOR
± 5V CLAMPING
CIRCUIT
Fig. b4
SAMPLE & HOLD
MODULE
ANALOGUE TO DIGITAL
CONVERTER
DIGITAL
STORE
DIGITAL TO SYNCHRO
CONVERTER
OUTPUT TO SHIP
COMPUTER
DATA CONVERTSION
TIME BASE
Fig. b3
SHIPS REFERENCE
POWER SUPPLY
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
FUNCTION DIAGRAM
Size
A
Document Number
C60 R21-1-00
REV
Date:
Sheet
of

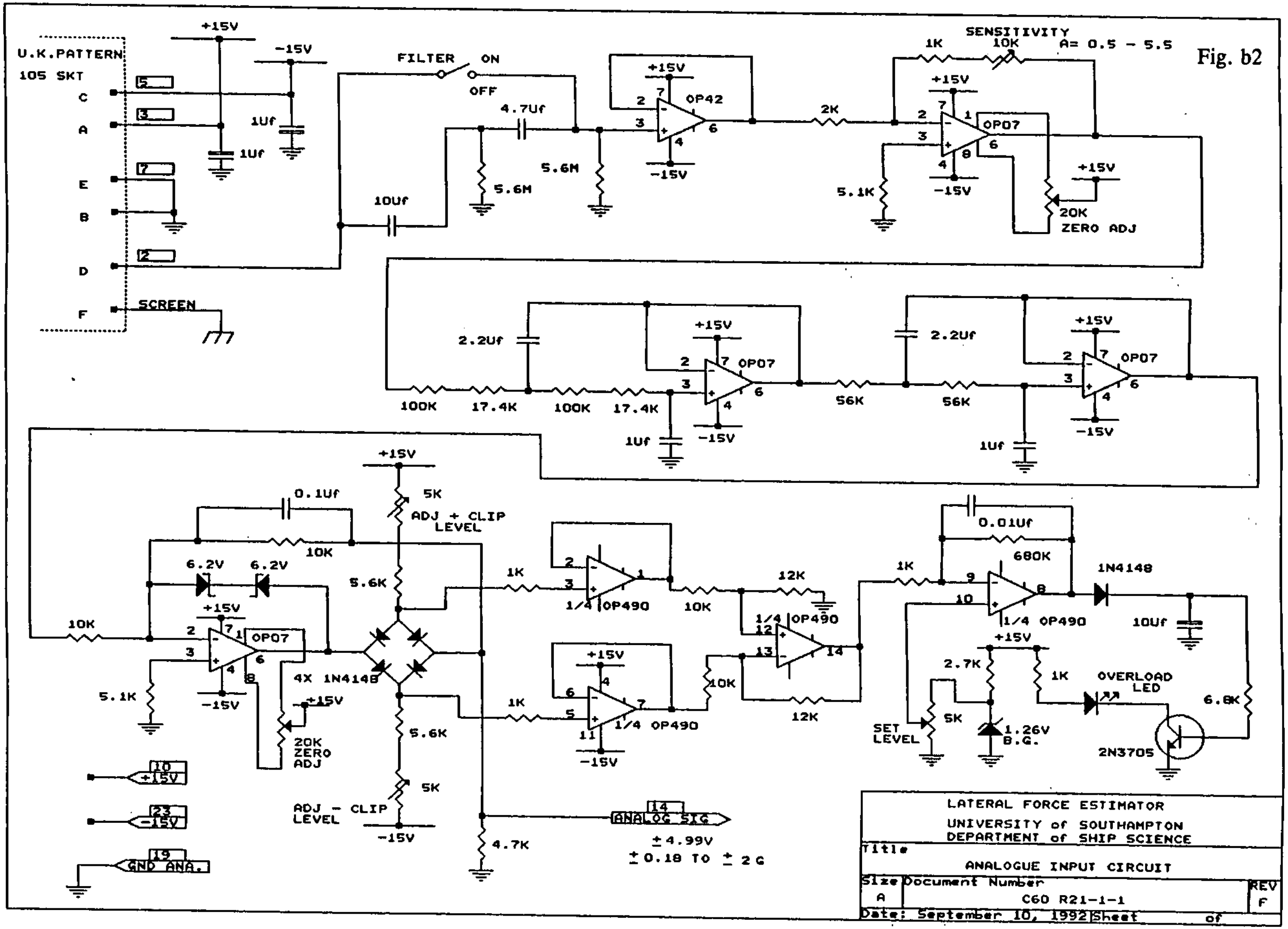
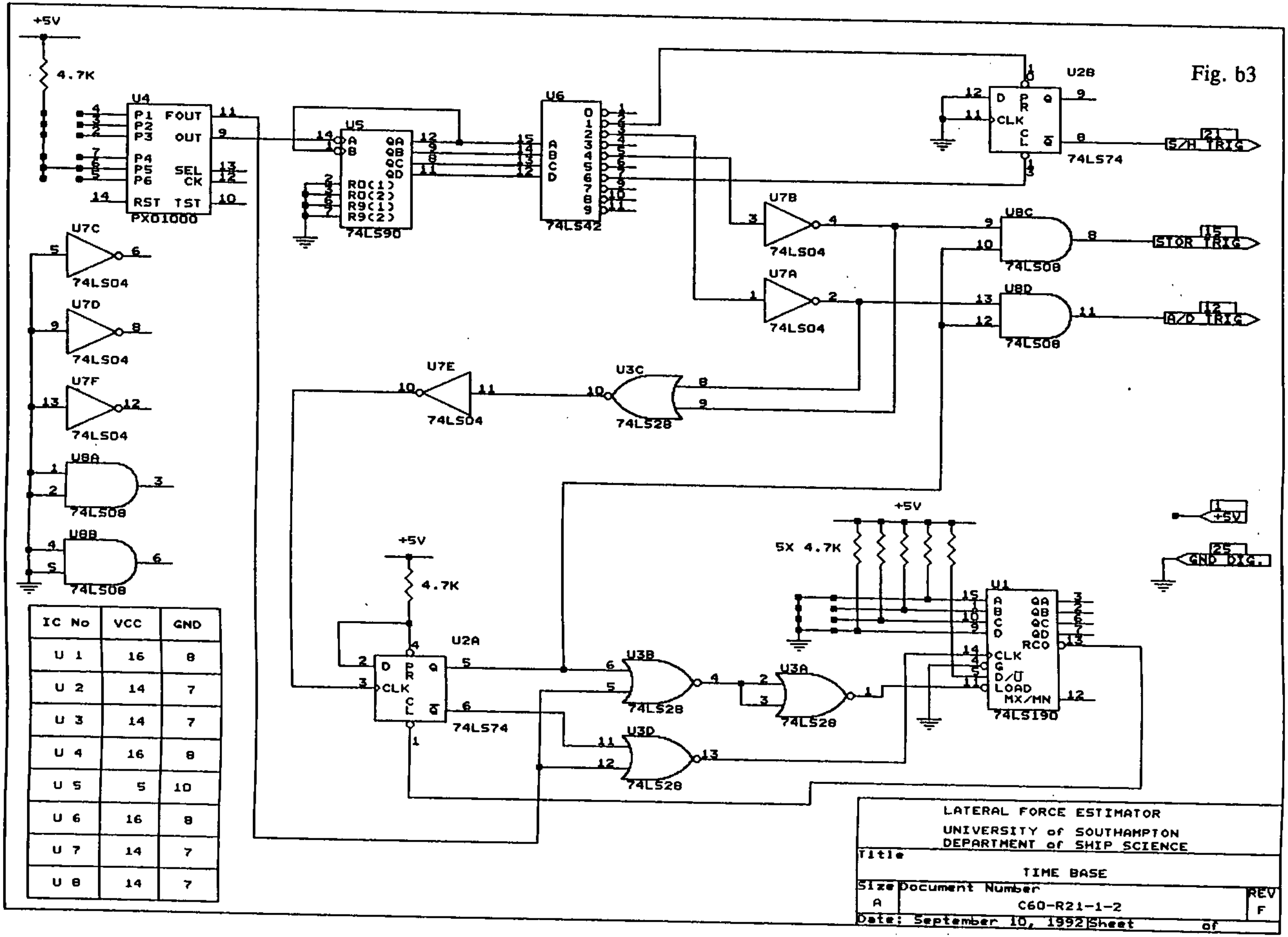
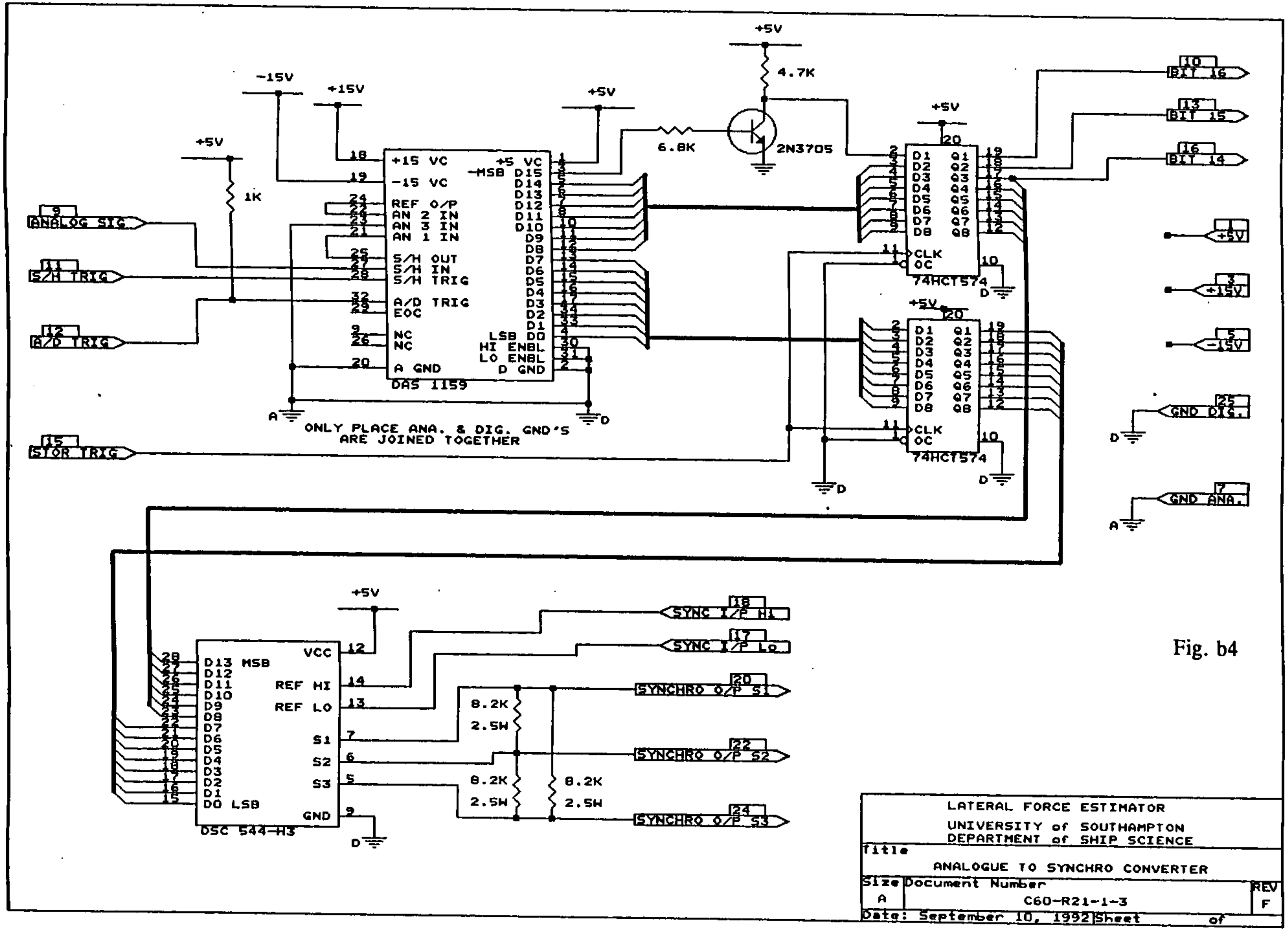

U.K.PATTERN
105 SKT
FILTER
ON
OFF
SENSITIVITY
A= 0.5 - 5.5
ZERO ADJ
SCREEN
ADJ + CLIP LEVEL
ADJ - CLIP LEVEL
4X 1N4148
OVERLOAD LED
SET LEVEL
1.26V B.G.
2N3705
ANALOG SIG
±4.99V
±0.18 TO ± 2 G
GND ANA.
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
ANALOGUE INPUT CIRCUIT
Size
A
Document Number
C60 R21-1-1
REV
F
Date: September 10, 1992
Sheet
of
Fig. b2

Fig. b3



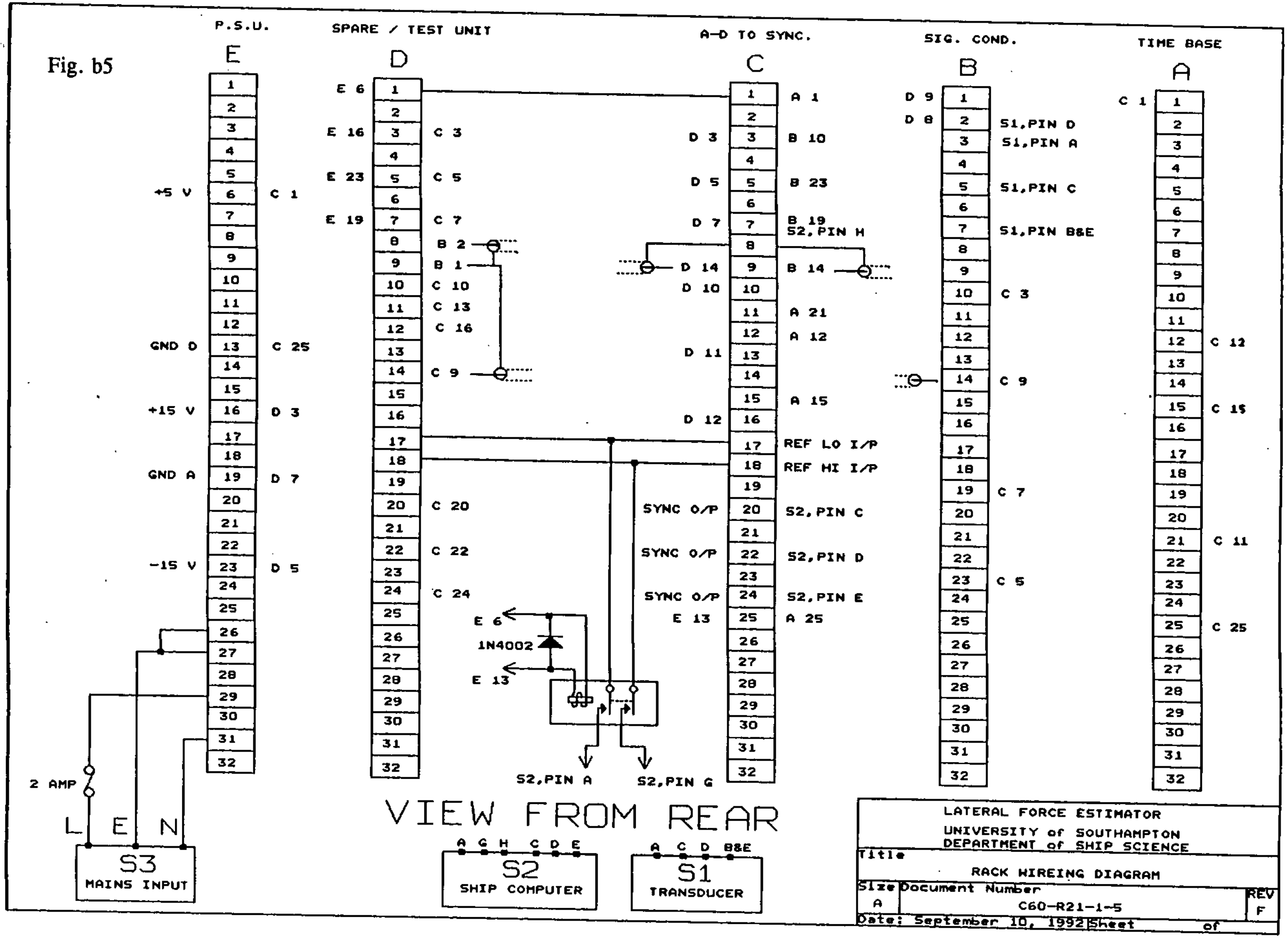
IC No	VCC	GND
U 1	16	8
U 2	14	7
U 3	14	7
U 4	16	8
U 5	5	10
U 6	16	8
U 7	14	7
U 8	14	7



ANALOG SIG
S/H TRIG
A/D TRIG
STOR TRIG
+15 VC
-15 VC
REF O/P
AN 2 IN
AN 3 IN
AN 1 IN
S/H OUT
S/H IN
S/H TRIG
A/D TRIG
EOC
NC
NC
A GND
+5 VC
-MSB D15
D14
D13
D12
D11
D10
D9
D8
D7
D6
D5
D4
D3
D2
D1
LSB D0
HI ENBL
LO ENBL
D GND
DAS 1159
ONLY PLACE ANA. & DIG. GND'S ARE JOINED TOGETHER
+5V
+15V
-15V
1K
4.7K
6.8K
2N3705
74HCT574
74HCT574
CLK
OC
BIT 16
BIT 15
BIT 14
+5V
+15V
-15V
GND DIG.
GND ANA.
VCC
REF HI
REF LO
S1
S2
S3
GND
D13 MSB
D0 LSB
DSC 544-H3
8.2K
2.5W
SYNC I/P Hi
SYNC I/P Lo
SYNCHRO O/P S1
SYNCHRO O/P S2
SYNCHRO O/P S3
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
ANALOGUE TO SYNCHRO CONVERTER
Size A
Document Number C60-R21-1-3
REV F
Date: September 10, 1992
Sheet of

Fig. b4

Fig. b5


P.S.U.
E
SPARE / TEST UNIT
D
A-D TO SYNC.
C
SIG. COND.
B
TIME BASE
A
+5 V
GND D
+15 V
GND A
-15 V
C 1
C 25
D 3
D 7
D 5
E 6
E 16
E 23
E 19
C 3
C 5
C 7
B 2
B 1
C 10
C 13
C 16
C 9
C 20
C 22
C 24
D 3
D 5
D 7
D 14
D 10
D 11
D 12
SYNC O/P
SYNC O/P
SYNC O/P
E 13
A 1
B 10
B 23
B 19
S2, PIN H
B 14
A 21
A 12
A 15
REF LO I/P
REF HI I/P
S2, PIN C
S2, PIN D
S2, PIN E
A 25
D 9
D 8
S1, PIN D
S1, PIN A
S1, PIN C
S1, PIN B&E
C 3
C 9
C 7
C 5
C 1
C 12
C 15
C 11
C 25
E 6
1N4002
E 13
S2, PIN A
S2, PIN G
2 AMP
L
E
N
S3
MAINS INPUT
VIEW FROM REAR
A G H C D E
S2
SHIP COMPUTER
A C D B&E
S1
TRANSDUCER
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
RACK WIREING DIAGRAM
Size
A
Document Number
C60-R21-1-5
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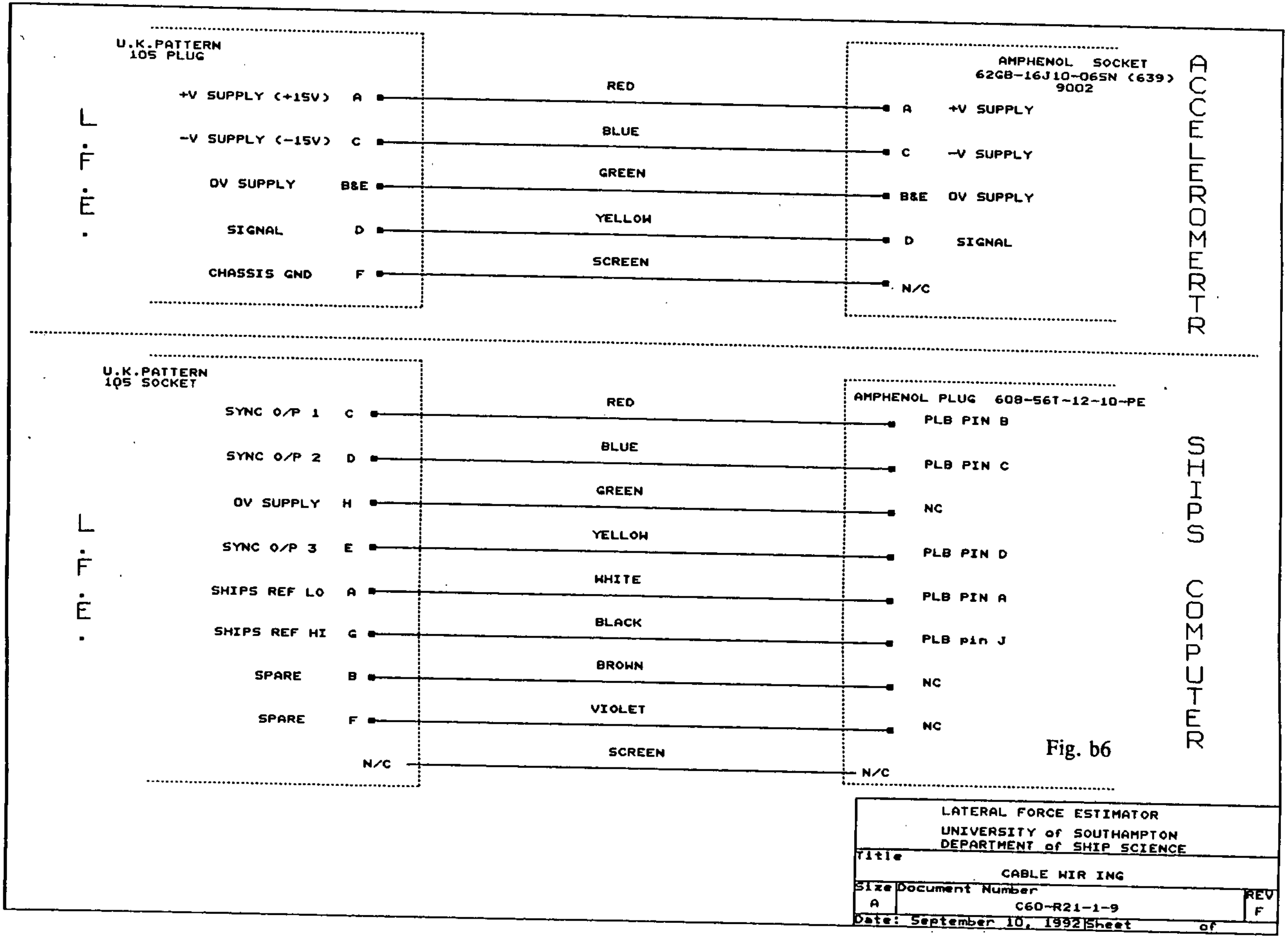


U.K.PATTERN
105 PLUG
L.F.E.
+V SUPPLY (+15V) A
-V SUPPLY (-15V) C
OV SUPPLY B&E
SIGNAL D
CHASSIS GND F
RED
BLUE
GREEN
YELLOW
SCREEN
AMPHENOL SOCKET
62GB-16J10-06SN (639)
9002
A +V SUPPLY
C -V SUPPLY
B&E OV SUPPLY
D SIGNAL
N/C
ACCELEROMERTR
U.K.PATTERN
105 SOCKET
L.F.E.
SYNC O/P 1 C
SYNC O/P 2 D
OV SUPPLY H
SYNC O/P 3 E
SHIPS REF LO A
SHIPS REF HI G
SPARE B
SPARE F
N/C
RED
BLUE
GREEN
YELLOW
WHITE
BLACK
BROWN
VIOLET
SCREEN
AMPHENOL PLUG 608-56T-12-10-PE
PLB PIN B
PLB PIN C
NC
PLB PIN D
PLB PIN A
PLB pin J
NC
NC
N/C
SHIPS COMPUTER
LATERAL FORCE ESTIMATOR
UNIVERSITY of SOUTHAMPTON
DEPARTMENT of SHIP SCIENCE
Title
CABLE WIR ING
Size A
Document Number C60-R21-1-9
REV F
Date: September 10, 1992
Sheet of

Fig. b6

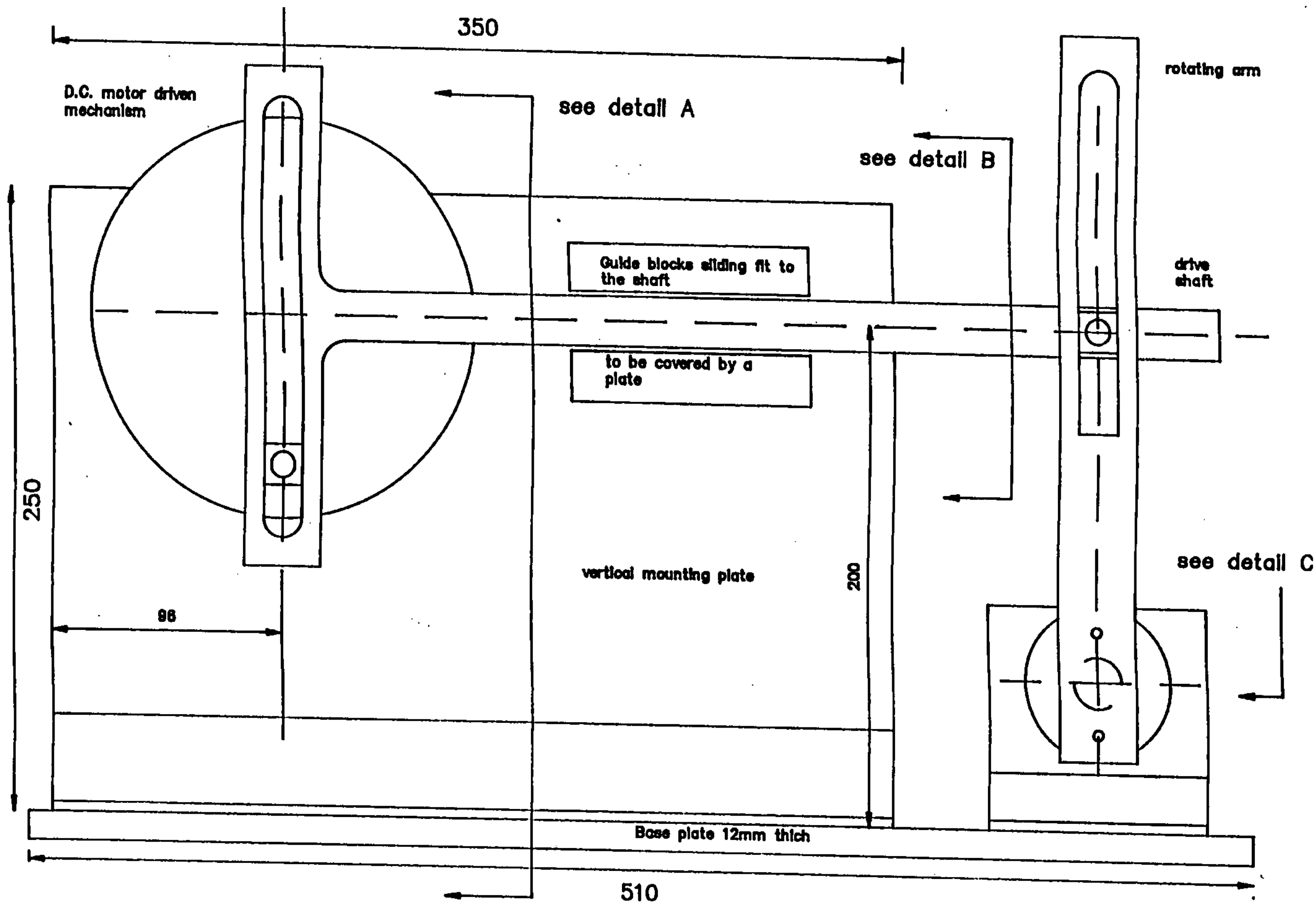
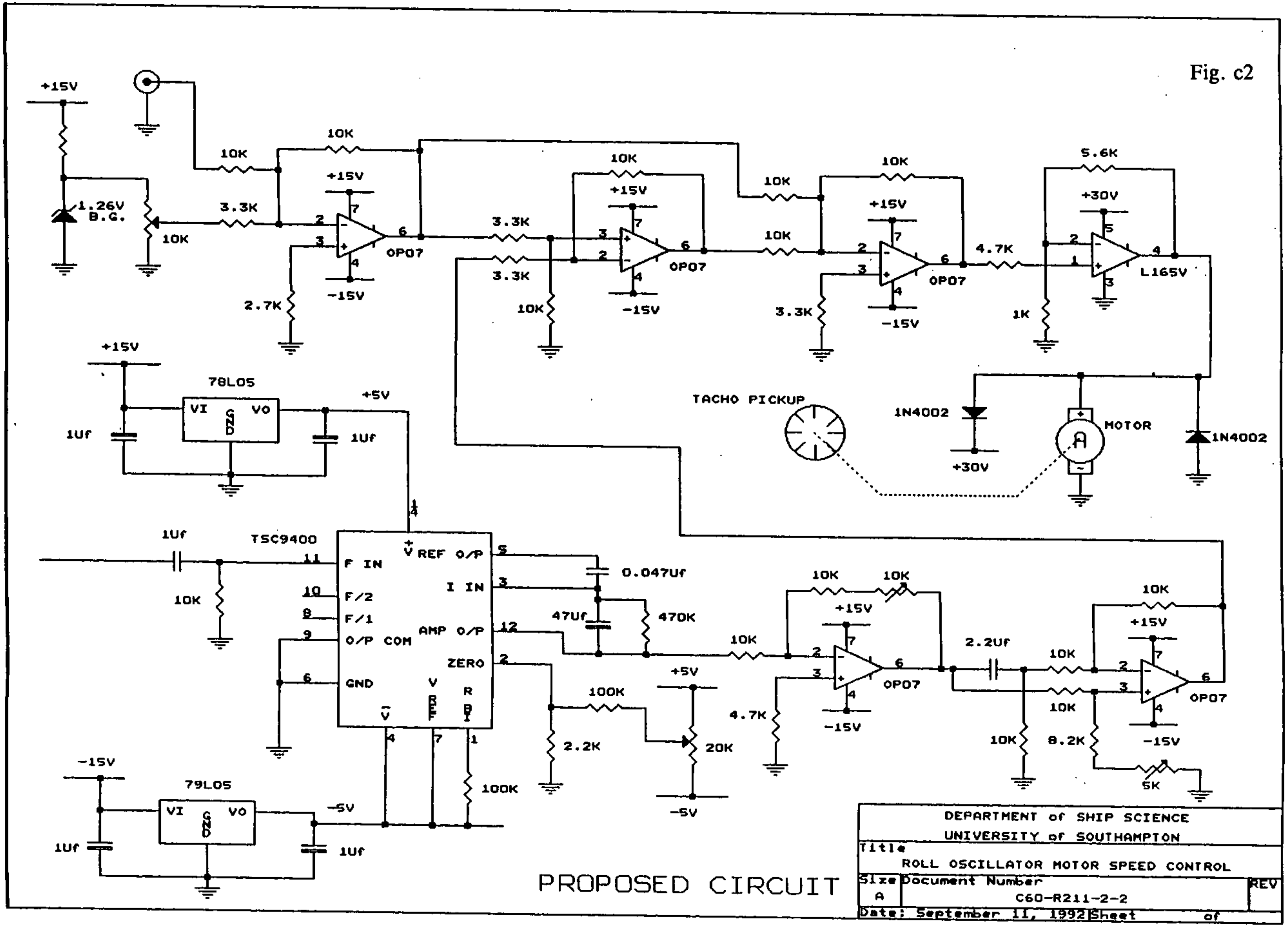


Fig. c1 signal generator for the active fin system scale 2 : 1

Fig. c2



PROPOSED CIRCUIT

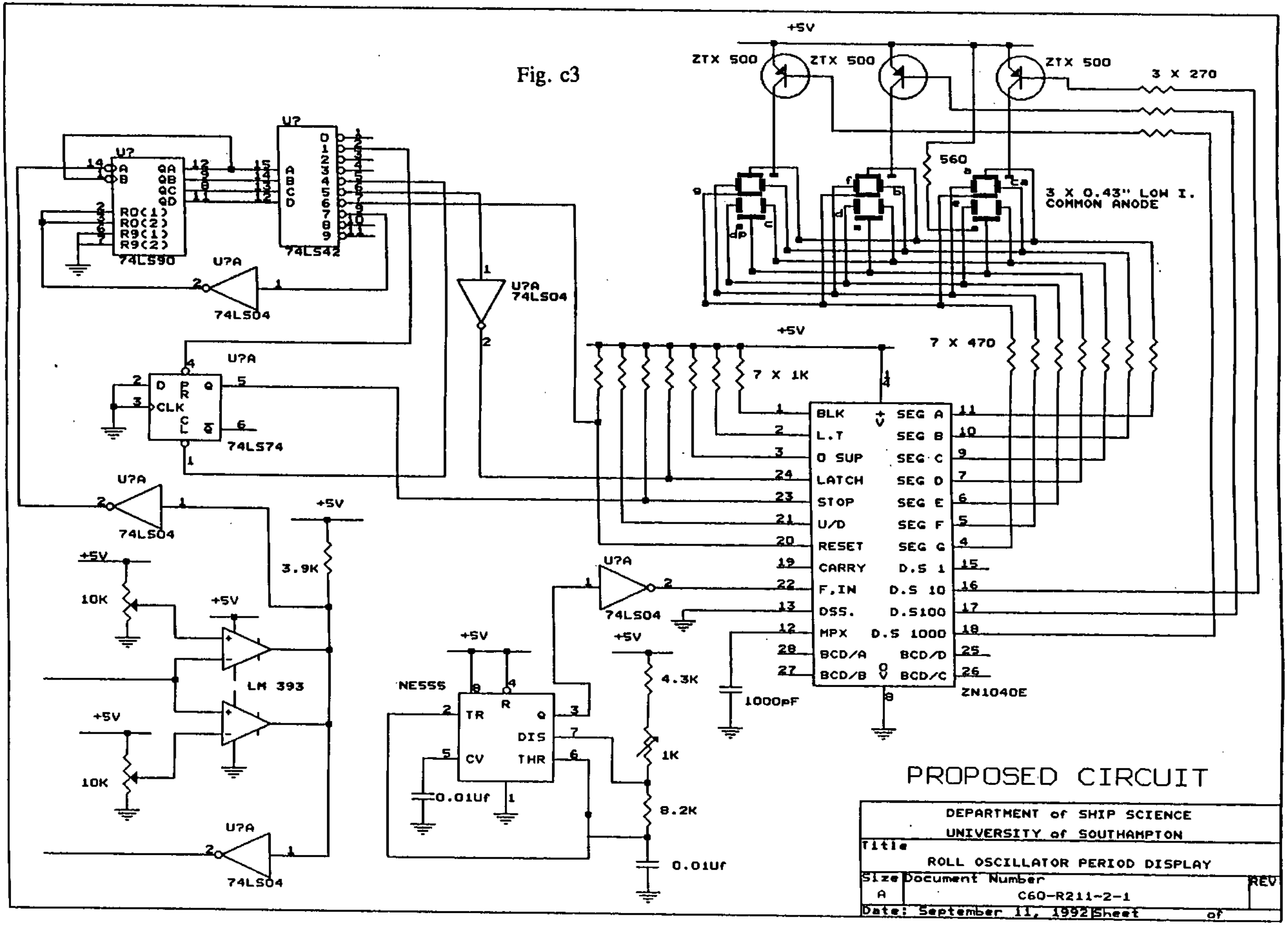
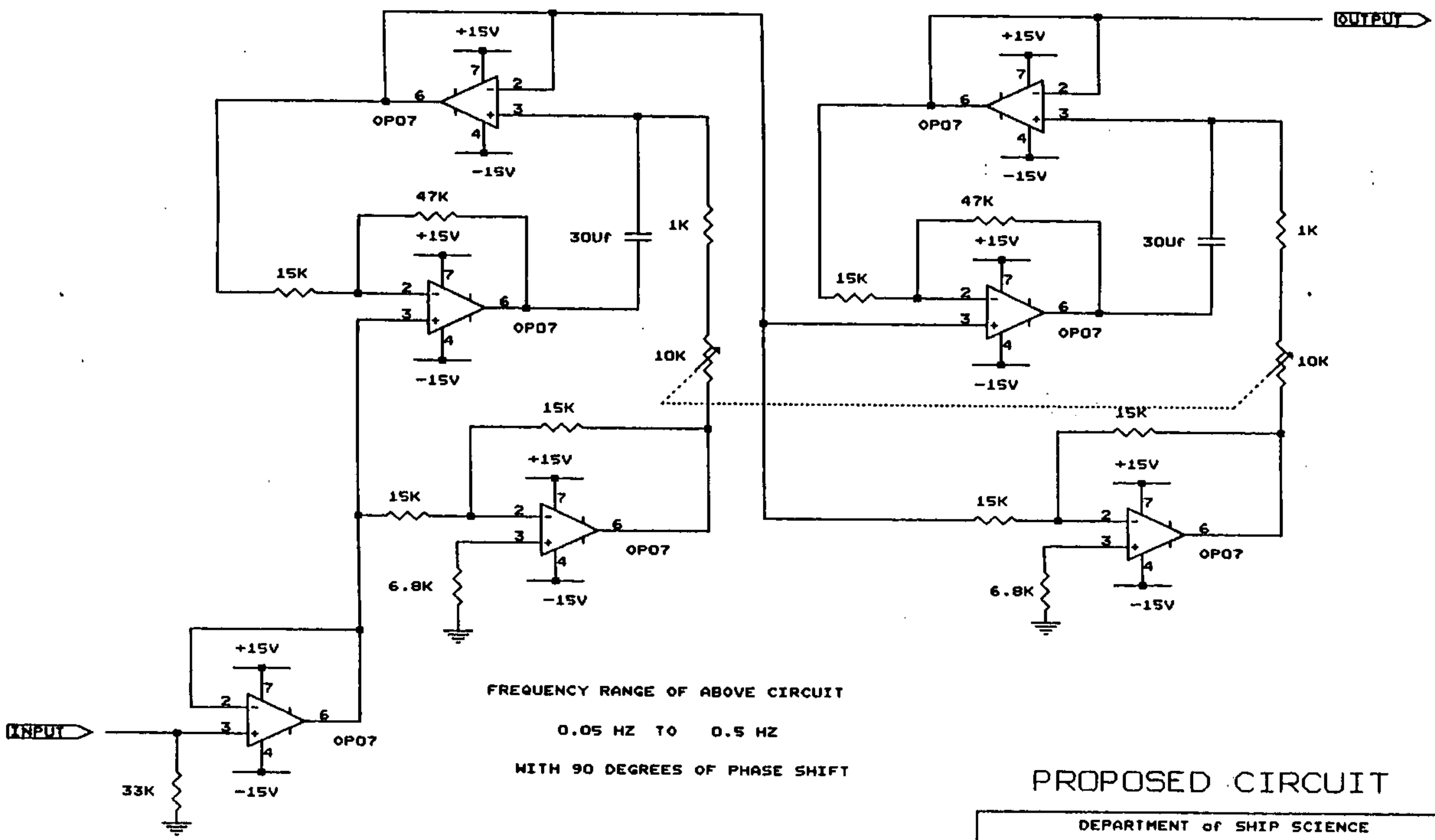

PROPOSED CIRCUIT
DEPARTMENT of SHIP SCIENCE
UNIVERSITY of SOUTHAMPTON
Title
ROLL OSCILLATOR PERIOD DISPLAY
Size A
Document Number C60-R211-2-1
REV
Date: September 11, 1992
Sheet of
ZTX 500
3 X 270
560
3 X 0.43" LOW I. COMMON ANODE
7 X 470
7 X 1K
+5V
ZN1040E
BLK
L.T
0 SUP
LATCH
STOP
U/D
RESET
CARRY
F.IN
DSS.
MPX
BCD/A
BCD/B
SEG A
SEG B
SEG C
SEG D
SEG E
SEG F
SEG G
D.S 1
D.S 10
D.S100
D.S 1000
BCD/D
BCD/C
1000pF
74LS90
74LS42
74LS74
74LS04
U?A
LM 393
3.9K
10K
NE555
4.3K
1K
8.2K
0.01Uf

Fig. c3

Fig. c4



FREQUENCY RANGE OF ABOVE CIRCUIT

0.05 HZ TO 0.5 HZ

WITH 90 DEGREES OF PHASE SHIFT

PROPOSED CIRCUIT

DEPARTMENT of SHIP SCIENCE UNIVERSITY of SOUTHAMPTON		
Title: 90 DEGREE PHASE SHIFTER		
Size: A	Document Number: C60-R211-2-3	REV
Date: September 11, 1992	Sheet	of