2 ANALYSIS

2.1 Timber cargoes exported from Finland, general

The total exports of sawn timber from Finland in 2001 were 7 182 000 m^3. The transportation takes place in ships that do not usually have their own lifting equipment. The deliveries to the Mediterranean and to North Africa take place as tramp cargo. The vessel types are often traditional. The share of deck cargo is about 1/3 of the entire cargo.

Sawn timber is exported from several ports. These days a considerable share of the production capacity is located inland, close to the supply of raw material. The most important changes concerning the exports of sawn timber include continuous shipping around the year, artificial drying and standardised package size.

The introduction of sawn timber into scheduled line traffic has been slow. The trafficking is still often done as tramp cargo pickups. One reason for this is the fact that trade to the North African countries has grown (more than 20% of the total exports of sawn timber). The delivery terms are often traditional and the buyers take care of the deliveries by using large tramp cargo vessels. The imports of sawn timber in many West European countries also take place through large importers and timber companies that usually manage the transportations themselves with vessels specialising in the transportation of sawn timber.

The auxiliary services affect the efficiency and service level of the port. A visit of a vessel to the port requires a considerable number of different activities. The shipping agents handle the documents related to the port visit. The forwarding agents act as representatives of the sender or receiver of the goods for tasks including: coordination of transfer and handling of the goods, customs formalities, payment of fees and charges, provision of the necessary transportations, drafting of the transportation documents.

The forwarding agent can act either as an agent, which means it does not carry the cargo carrier's responsibility, or, as an independent cargo carrier. The transportation takes place according to the agreed shipping terms. The stevedoring company loads and/or unloads the vessel according to agreement and handles the goods in the port area. The export handling of sawn timber takes place along the same principles in the various ports. The port is a significant cost factor in the transportation chain.

The most common dimensions of sawn timber with regard to thickness and width are defined e.g. in RT 21 10626 in Finland. The maximum allowed deviations in the cross section measurements are in compliance to standard EN 336. The thickness and width dimensions are the nominal dimensions of the timber at 20% humidity. The humidity of sawn timber used for framework structures is normally 15-24% of the dry weight. The most common lengths of sawn timber range from 1800-5400 mm at 300 mm intervals so that the length of the side boards ranges from 1800-4200 mm and the length of the board in the middle from 3900-5400 mm. Sawn timber is a general term for timber sawn at least on four sides. The normative quality classification of sawn pine and spruce timber are based on the joint Nordic classification guidelines for sawn timber. The average volume weight of dry timber is 0.52 t/m^3 for pine and 0.42 t/m^3 for spruce (absolutely dry wood).

Packaged sawn timber is delivered both length packages and as truck packages (one end uneven). These days almost all sawn timber exported by ships is wrapped lengthwise. The most common package size is $1.15 \cdot 1.15 \cdot 1.80 - 6.00m$. The weight varies according to the humidity. The tying into a package is done with steel bands without ground timber. Lengthwise wrapped sawn timber packages can be hooded at request from the buyer. It is often necessary to hood the packages in order to keep the sawn timber usable, since the buyer often stores the packages in open air. Typical hood materials for sawn timber include cardboard/paper hoods reinforced with artificial fibre webbing and coated with weatherproof material, or the material is weatherproof plastic (clear or coloured). The friction coefficient of the hood material varies. Some sawmills use a friction coating on the hoods. It is a demanding task to optimise the relation between sufficiently strong protective properties on the one hand, and the packaging costs and the need for post processing on the other. Environmental friendliness and suitability for recycling have a large significance.

The sawn timber packages are stored in the ports and at the sawmills either in open air and/or shelters piled on top of one another. A forklift truck usually does the handling. Transportation to the ports takes place either by train or by trucks. The stevedore companies handle the goods for in-port storage for shipping or the goods can be delivered directly alongside the ship. The shipping terms determine the method of delivery.

The sawn timber packages are loaded on board by a crane (lo-lo) or lift wagons, trailers or trucks can transport them, and also sto-ro loading is used. Depending on the size of the vessel, the stevedore may use machines in the hold of the ship (lo-lo, sto-ro). When loading by crane, the most common lifting devices are polypropylene ropes or belts of artificial fibre that travel with the cargo. The use of steel wires is rare. The lifting devices are selected according to the weight of the goods and the lifting method. The transportation of sawn timber in containers is growing.

The necessary information of the loaded cargo is delivered to the Master of the vessel (Manifest of Cargo, Loading Order, special properties, identifications, quantity etc.). The stevedore loads the vessel according to the approved cargo plan drafted by the vessel. The cargo plan includes the placement of the cargo on board.

2.2 Cargo-securing manual and loading

The cargo-securing manual gives e.g. the following instructions concerning a cargo of sawn timber:

Instruction: The shipper of the goods shall provide at least the following information to the Master of the vessel:

- Number of packages and total volume of cargo
- Actual humidity of the cargo (if the humidity is greater than the transportation humidity, where the density is 0.435)

- Dimensions of the packages classified into standard length packages and packages of varying length
- Friction coefficients for material pairs cargo/cargo, cargo/steel and cargo/wood for packages covered with waterproof coatings.
- Nominal loading volume, types and densities of wood listed for the loading

The Master of the FJORD PEARL did not get the information of the friction coefficients. On the other hand, the cargo-securing manual does show, how the friction coefficients are used. The calculations in the cargo-securing manual are based on the stability angle of a package stack, which in turn depends on the friction coefficient and the tipping limit of the stack. Impacts of changing conditions have not been given.

The placement of the cargo created vertical friction surfaces between the packages in addition to the horizontal friction surfaces between the packages and between the packages and the surface under them. The surface under the cargo was level only on the hatch covers and on deck. The situation was complicated further by the lifting ropes left around the packages and by their protective hoods.

Enclosure 4 list generally available friction coefficients that are given for dry or wet wood and metal surfaces. The static stability angle, 17°, of the package stack in the cargo securing manual of the FJORD PEARL corresponds to a friction coefficient μ = 0.3. When this is compared to the commonly used friction coefficients, it can be concluded that this figure includes ample safety margin. Various sources give μ = 0.4 – 0.5 as the static friction coefficient for the material combination wet wood/steel, when there are no ropes or hoods between the surfaces. The investigation has not succeeded to clarify, how the lashing calculations were carried out on FJORD PEARL.

In this case friction coefficients in winter conditions should have been known, because these might become so small, that they determine the stability angle of the package stack. The ICHCA (International Cargo Handling Co-Ordination Association) recommendation quotes 0.1 as the friction coefficient for snowy and icy wood/metal surface combinations. The IMO recommendations concerning timber deck cargo include a warning of lower friction on an iced cargo hatch. However, no practical procedures for controlling the situation have been given either in these guidelines or the cargo-securing manual. During the loading the ice and snow was removed from the timber packages and from the hatch covers in order to maintain as high a friction coefficient as possible.

The deck cargo of the vessel got partially wet in the prevailing wind and waves although it had been covered. The cargo was cold; its temperature during loading was -20°C. During the voyage the outermost sections of the deck cargo warmed to the air temperature, -4°C. The cargo was coldest in the holds. It is probable that frost formed on all surfaces reducing the friction. The water temperature was about +1°C. There is no data of the humidity of the air. It is probable that the seawater that came into contact with the cargo froze at least in some places. Thus, it is possible that there was both water and ice at the same time between the tiers. The tiers did not come into close contact with each other in all parts of the deck cargo because of the ropes in between. There were gaps that possibly let in water that froze. A combination of friction, lubrication and glueing ensued.

The investigation team decided to commission a series of friction tests that would replicate the above winter conditions as far as possible. On the vessel, the following material combinations are the most important: wood-rope-metal and wood-metal. These correspond to the situation at the hatch covers. In between other tiers, the material combination was wood-rope-hood. Some of the tests were conducted on an iced steel plate frozen to - 22°C. The measurements were repeated while the ice gradually melted at room temperature. Figure 22 presents the effect of the thaw. The surfaces were initially icy; the surface temperature was -2-4°C. In the end the surfaces had thawed out completely.

Enclosure 4 includes the results of this test series. The friction coefficient diminishes radically when the thaw progresses and begins to grow when the ice has completely melted away.

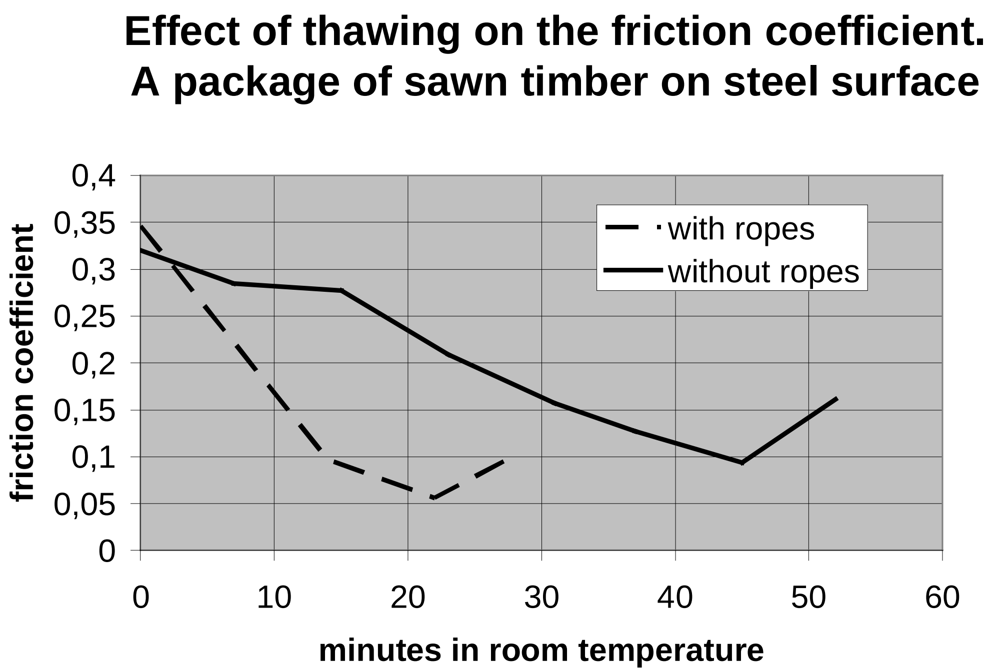


Figure 22. Effect of warming on the friction coefficient in the friction tests performed at the Technical Research Centre of Finland

Based on the observation and studies on the behaviour of the friction it can be concluded: FJORD PEARL sailed in conditions resulting in significantly lower friction coefficients and angles of static stability of the package stack than the initial values used as the basis when securing the deck cargo.

Instruction: It is permissible to stack timber packages lengthwise or lengthwise and crosswise on the hatch covers. If lengthwise and crosswise stacking is used, the second tier is placed crosswise, if there are in total three tiers and the first and third tier crosswise in a stack of four tiers.

The packages on board had been placed according to the three-tier mixed option although some of the cargo had been stacked in four tiers. The Master had selected this option because the fourth tier was less than half the length of the deck cargo. The way of stacking has an effect on the stability angle of the package stack.

The figure 4.2.2 of the cargo-securing manual should be corrected so, that packages should be stowed lengthwise along the sides also in the tier placed athwartships.

Instruction: When stacking packages lengthwise in two or three tiers, uprights must be mounted on the sides of the vessel prior to starting the loading. If the deck cargo is covered with protective sheets, a gap of 30-50 mm should be left between the cargo and the uprights. The packages are covered with protective sheets at request of the receiver of the goods.

The uprights were mounted although the packages were stacked lengthwise crosswise. According to the instructions, no uprights are needed if the topmost tier of a cargo stacked lengthwise in two or three tiers has been stepped in. The uprights were mounted to improve work safety (in case of toppling of the timber stacks prior to securing them).

Instruction: The cargo must always be secured tightly. At sea, the securings are checked twice a day or more frequently if needed. If a storm warning is issued, an extra check must be performed.

The last tightening had been performed six hours before the accident at which time the warning for storm winds had already been noted. It is not clear whether this check was a normal check according to the inspection scheme or an extra checkround because of the storm warning. By the time of the accident, the tension forces in wires had decreased. The tightening force was not measured. Therefore, it was not known how near it was to the design value. Investigation did not clear up this value.

In addition, the IMO guidelines state that:

Instruction: Air pipes and ventilation shafts shall be well protected.

The air and sounding pipes and ventilation shafts had been constructed according to the regulations. The shifted deck cargo broke a section of the railing, which resulted in a break of the air and sounding pipes in the respective area.

Instruction: The timber deck cargo must be kept free of snow and ice during the loading.

It is known that the stevedore removed snow by blasting air and ice by chiselling. Some snow remained on the sawn timber packages and on the hatch covers since it was snowing during the loading.

2.3 Loading of the vessel and wintry conditions

In the winter water, snow, ice and moisture accumulate on the sawn timber packages during the transportation chain. Some of the packages are stored in open air, some in covered partly open shelters. The wind deposits water and snow between the packages, in the gaps and on the surfaces under the hoods. The vessel may also be snowy and icy during the loading.

During the winter months, the temperature can range from severe colds to several degrees above zero during the loading and during the voyage. When this factor is combined with the wind, rain and snow and the spray and waves at sea, the result can be an increase in the weight and a considerable reduction in the friction coefficient. Sailing in ice covered waters shakes the cargo a lot more than a passage in open water. The dark and the cold inhibit human activity.

The Master should receive data on the physical properties and storing of the sawn timber packages for planning of the securing of the deck cargo. In addition, the conditions at sea have an effect on the friction coefficient. By monitoring the status of the cargo it can be estimated whether the situation is turning critical with regard to the friction. This should be considered when navigating the ship. The corresponding instructions should be included in the cargo-securing manual.

Figure 23. Timber packages *stored in open air, work safety should be considered in the storage (photo by the Finnish Institute of Occupational Health)*

2.4 Estimated list-friction coefficient ratio

2.4.1 Calculations based on the cargo-securing manual

Calculations, performed by the investigation team according to the cargo-securing manual, give the total tightening force of 4592 kN. Calculations are shown in enclosure 2. The maximum heel angle for the vessel is 34° and the dynamic stability angle of the package stack is 16.5°. The weight of the deck cargo in need of securing is estimated to 1250 t.

2.4.2 Prevailing friction coefficients

There were several different horizontal glide surface combinations. All but the surface lowermost and closest to the side included the lifting rope. Moreover, the ropes could lie on top of one another and across in various ways. Most likely, the package dimensions and forms have varied, and snow, ice and frost have occurred unevenly. In places, the surface pressure may have grown so great that the ice melted which resulted in a considerable reduction of the friction coefficient in that spot. The movement of the vessel caused the cargo to shift slightly and pack. The area of the cargo lying on ice could increase and thereby decrease the average friction coefficient.

During the loading ice and snow was left between the packages. Although the cargo had been covered with protective sheets more ice formed as a combined result of spray, waves and the below zero temperatures. There was momentarily water between the icy cargo surfaces. It can be assumed, that the friction coefficients reduced considerably during the voyage when the temperature rose.

At departure from Pietarsaari the cargo was cold, especially in the hold. During the voyage as the air warmed, frost formed on the surfaces, which reduced the friction.

Due to the factors above, it is only possible to reconstruct the upper and lower limits for the average friction coefficients and for the heeling angle causing the shifting of the deck cargo.

The results of the calculation are presented in figure 24. The tightening force at the time of the accident is estimated to have been between 40-60 kN. There were about 6 hours between the last tightening and the accident. During this time, the weather conditions were at their worst, which is why the cargo evidently packed further. This resulted in a decrease of the tightening force.

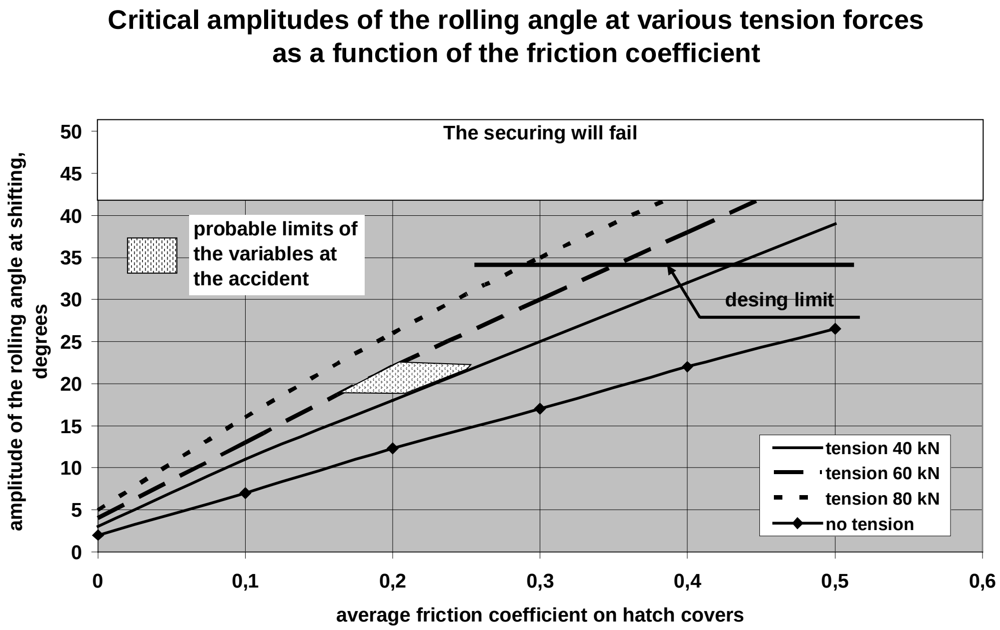


Figure 24. Critical friction coefficient-rolling angle-tightening force combinations

According to the reports, the vessel listed first about 15° to port. The deck cargo did not start to move, but its stability could have been disturbed. It can be deduced that the friction coefficient was at least 0.12-0.17. The friction coefficient at the time of the accident has been estimated within 0.18-0.25 depending on the tightening force. Based on Fig. 24, it can be concluded, that the lashing calculated using the instructions in the cargo-securing manual is adequate if the friction coefficient is more than 0.35-0.4.

2.4.3 Effect of the place of the cargo

Rolling, yawing and pitching are movements that occur around the rotation axis. The forces due to rotation are directly proportional to the distance of the part of cargo from the rotation axis.

The forces caused by pitching and yawing are small compared to the force components created by the mass and affecting the cargo at the bow and stern when the vessel is listed. According to enclosure 3, the transversal force in the first tier is mass x 0.25 and in the fourth tier mass x 0.38. This effect is at the most a 6% increase in the static forces.

In the uppermost tier the friction force and the shifting force, on the other hand, are created by the mass of one tier of deck cargo only. The downward force of the securing wire is about 90% of the tension power because of the slanted position of the wire in the upper corner. The forces created by the mass of the deck cargo are the smallest compared to the tension forces. The situation changes further down. The downward force reaches its maximum in the following tier but the forces resulting from the mass are still growing.

Due to the great range of local variance of the friction coefficient, it is difficult to estimate where the deck cargo shifted first. The fact that the cargo shifted most in the stern of the vessel may attribute to the possible sliding of the stern down the wave slope when the vessel tossed.

It is unclear whether the cargo shifted first in the hold or up on deck. There was no extra tension force in the hold but the static force was created by the weight of the sawn timber packages themselves. The stacks have swayed slightly. On the other hand, the friction coefficient may have been greater than on deck. The effect of the conditions of the hold on the friction properties of the cargo surfaces is difficult to estimate. The cargo in the hold was initially very cold, -22°C.

By comparing the friction coefficients of the various surface combinations it can be concluded that the tier of the cargo on the hatch covers was most critical from the point of view of shifting.

2.4.4 Evaluation of the load on the securing wires of the cargo

According to the reports the wires remained intact. In the first stage the securing wires were stressed due to the rope friction when the deck cargo shifted. Another stress peak was created when the shifted cargo came to rest on the securing wires. Before the final stop some of the kinetic energy of the deck cargo had been consumed on the following: breaking of the uprights, heeling of the vessel, friction between the securing wires and the cargo, internal changes of shape in the sawn timber packages and friction between them.

The elongation of the securing wires must be less than 5% at 80% of the break load. This limit is 110 kN. It is possible that the rope friction caused a stress near this value. The effect of the stopping of the deck cargo is estimated to have been smaller.

The crew looked at the securing wires in connection with the repair of the damages of the vessel. Some of the wires were cut. No damage whatsoever was found in the securing wires.

It is possible, however that some changes have occurred in the securing wires which is why their monitoring should be increased.

2.5 Evaluation of the accident situation of the vessel

2.5.1 Degree of danger of the situation of the vessel

Under the prevailing circumstances the FJORD PEARL was subjected to several danger factors:

- Turning the vessel against the wind from a downwind position with a 22° list was a demanding operation. In addition, the vessel was rolling violently to both sides of this equilibrium. This turn presented an obvious risk of further damage.
- The deck cargo could have shifted further. The rolling might have snapped some of the wires. The officers could not have a clear picture of the condition of the cargo securing equipment after the shift of the deck cargo. Some of the deck cargo could have been left hanging outside the ship and some could have found its way into the propellers. The shifting deck cargo could have broken more air pipes and more of the gunwale and the railing. An additional list would have been dangerous.
- The pumps could barely stabilise the situation. A disturbance in the pumping would have led to an increase in the list.
- According to the weather reports the weather was improving but the warning for icing was still standing. It would have been impossible to remove the ice in the existing conditions, so the deterioration in the stability caused by the ice had to be taken into account. The icing and the list prevented the windlass from being used.
- It was difficult to control the listed ship. Only the port side diesel tanks could be used. The vessel travelled for about 13 hours without a pilot.
- The lowermost scuttles of the starboard side living quarters and one of the thresholds leading into the living quarters were under water.

2.5.2 Stability of the vessel at a listing angle of 26°

After the accident the ship was steered in head wind and waves. The cross-waves caused the vessel to roll a few degrees. According to enclosure 3 it can be stated that if the vessel remains against the wind the gusts do not present a danger.

If the vessel ended up in a side wind, she would roll 10° and could list to about 40°. This is quite near to the limit when the deck cargo will go over board.

The formation of ice has two effects. Firstly it raises the centre of gravity, which again decreases the stability. Secondly, more ice forms on the listed side, which increases the list further. The following looks at the effects of an extra 200 tonnes of ice. The ice is placed at a height of 12 m and at a distance of 3 m from centerline on the listed side. The new listing angle of the vessel would be about 30° and she would roll to about 38° in head wind.

The passage of the vessel to Utö was fortunately against the wind, so her rolling was limited to less than 5°. The passage from Utö to Turku was travelled in a side wind but the wind was from the listed side and from the coast. Moreover, the wind was calming down.

In summary it can be stated that the load condition of the vessel at departure from Pietarsaari was such, that her stability was enough at a permanent list of 26°. The design calculations had been performed for a wind up to about 43 m/s, according to the regulations.

2.5.3 Effect of flooding on the vessel

The following evaluates the effect of the flooding resulting from the damage presented in Section 1.3.7. Water got into the holds apparently through the sounding pipes and/or because of the leaking seals between the hatch covers and hatch coamings. Fuel tank 2 was midships and about half full, so the water pooling there did not heel the ship but increased the draught and the stern trim slightly. The fuel became unusable. The pipes leading to the other fuel tank remained intact. The water pooling into the holds remained at the edges of the holds in the corners and thus had a tilting effect increasing the draught. The dry tank is situated at the side and the water pooling into it tilted the ship and increased her draught.

The diameter of the air pipes was 133 mm, of the sounding pipes 57 mm and their broken ends lay about 2.5 m below the water surface. The rate of inflow through the pipes can be calculated from the known formula v = $k\sqrt{2gh}$, where water runs out of a small hole on the side of a barrel. The flow rate of the water calculated in this way is = $\sqrt{2 \cdot 9{,}81 \cdot 2{,}5}$ = 7.0 m/s. The area of one air pipe is 139 m^2 and one sounding pipe 26 m^2. It is estimated that the total effective area of the holes for the leaks in pipes and in the sealings of the hatch covers was 450 m^2. When the flow resistance is included by using a coefficient of k = 0.3 the resulting flow is about 340 t/h. Two pumps that yielded

100 t/h each compensated this. Thus, the pumping volume would not have been enough. However, according to the reports the situation was stabilised. This can be explained as follows: The water flowing in had to push the air out, which makes the assumption of the above formula that the air pressure in the tank was the same as the air pressure outside it false. In all probability, the holes for the leaks were not clean and uniform holes. The total size of the leak holes is therefore only an estimate. The vessel was lucky in the sense that no bigger holes were created.

The flooding of the tanks would not yet have been fatal as such, but the flooding of three holds would have sunk the ship in due course. On the other hand, the leaking into the holds was slow. The situation could be controlled with two pumps, but a disturbance in one pump, for example, would have resulted in a flooding rate of about 100 t per hour. An amount of 17 t of water equals an increase of 1 cm in the draught, so in one hour the draught would have increased by about 6 cm. The list would have grown by 3°. The basic hydrostatic and stability curves are not known but it can be estimated that the situation of the vessel would have turned critical in the next few hours had one of the pumps broken down and no pumping assistance from the outside been available.

2.6 Evaluation of the rescue activities

The crew of the ship showed a high level of seamanship by controlling a difficult situation.

There were about 7 hours between the forewarning of a danger situation sent by the vessel and the time of the accident. The forewarning according to the IAMSAR Agreement could have sent considerably earlier. By sending the forewarning earlier the Master of the ship would have ensured the safety of his crew and could have expedited the initiation of the rescue operation had something untoward happened.

During the rescue operation evacuation of the vessel with sufficient equipment was prepared for. All activities were undertaken as quickly as possible in practice. Contact was maintained to the vessel. Two pilots because of the difficult situation piloted the ship. The situation was analysed successfully, for example towing assistance was obtained. The unloading of the timber cargo was organised flexibly.

3 CONCLUSIONS

3.1 Shifting of the cargo of the FJORD PEARL

The chain of events leading to the shifting of the deck cargo can be divided into the following phases:

1. Turning and the resulting list in the wind and waves.
2. The friction limit was exceeded due to the list and the accelerations and the cargo started to move.
3. The cargo shifted, gained kinetic energy and the ship listed further.
4. The wires stopped the cargo, the kinetic energy was absorbed, and e.g. the uprights broke. Air pipes were broken; the corner of the deck sustained a tear, the list of the vessel diminished slightly as the rolling subsided.
5. The vessel gained a new equilibrium; water began to pour in through the holes in the damaged structures.
6. After the pumps started, the vessel was rebalanced.

The vessel started to turn from bow seas into quartering seas. According to the reports of the officers, the vessel was unable to turn at first. The speed had to be increased in order to make her turn. The vessel listed past the critical friction limit and the cargo started to move. This resulted in a toss to 30°.

The friction coefficients were lower than expected due to the weather during the loading and the voyage. The cargo started to move when the list was between 19-22° at which point the static friction coefficient was 0.18-0.25. The friction coefficient of the shifting cargo diminished further. Alarming proof of unexpected development of the friction coefficient of an iced timber cargo stored in open air was obtained in the tests performed for determining the friction coefficient when the cargo warms up. For a cargo like this, the radical reduction in the friction coefficient may lead to unexpected consequences in conditions that should not present any risk for a cargo shift. Therefore, special attention should be paid to the securing of timber cargoes transported in the winter in such a way that the lashings prevent any major shifts that could jeopardise the stability of the cargo. Another solution improving safety would be to avoid transporting sawn timber as deck cargo in winter conditions.

The lashing of the deck cargo had been performed based on the cargo-securing manual. According to the calculations by the investigation team, the tension of the securing wires played a great role in securing the deck cargo. If there is reason to suspect that the securings have become slack, the navigation should pay attention to minimising the motions of the ship. The significance of the securings is emphasised at low friction coefficients. The prevailing tightening force was not known on board, because it was not measured.

It can be stated that the wind and wave conditions did not exceed the design limits. The length of the ship was close to the prevailing wavelength. When the vessel was sailing in head waves she was pitching violently. Her rolling was unpredictable as the waves were met at different angles. When the vessel was travelling in the quartering seas, it was difficult to steer her.

The forces created during the turn contributing to the shifting of the cargo exceeded the securing forces because of the reduction in the friction coefficient. Figure 25 shows one possible scenario of the development of forces.

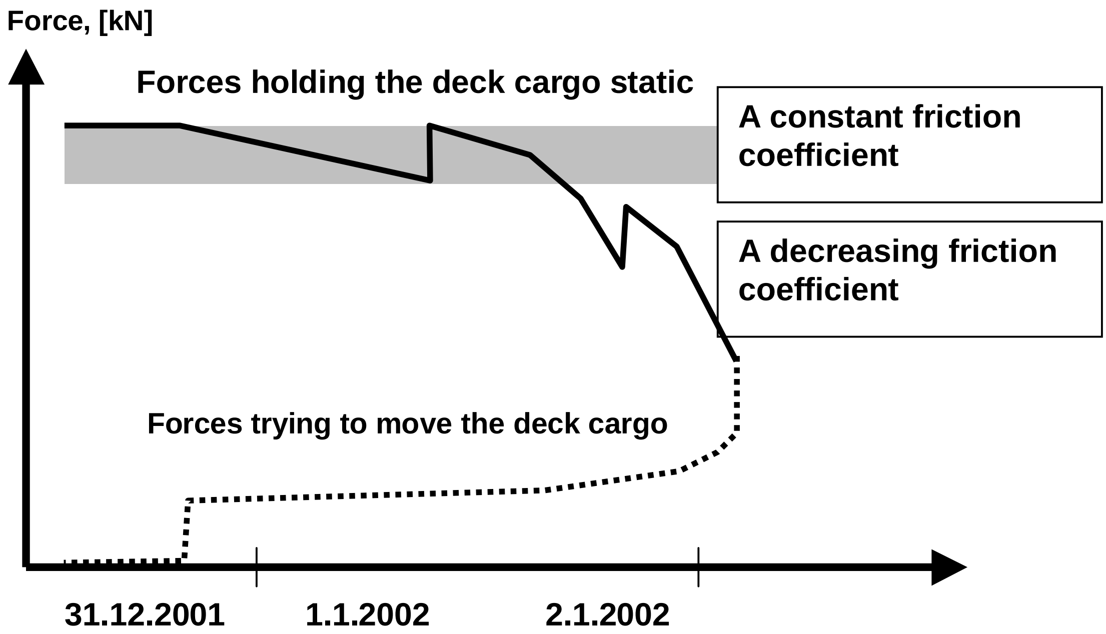


Figure 25. One possible scenario of the history of the forces during the voyage of FJORD PEARL, abrupt changes in the securing force show the moment of tightening

Considering the safety of the vessel she had to turn in head wind as quickly as possible. Had driving her with the wind been continued the cargo could have shifted further and caused more damage to the structures of the ship.

The vessel had to change course at open sea at a time when the wind and the waves were at their maximum. On the other hand, the fact that the wind started to calm down after the accident and was blowing from an advantageous direction contributed to her rescue.

3.2 Underlying factors contributing to the accident

The air pipes were protected according to the regulations but the shifting deck cargo broke some of them. This made the consequences of the accident worse. The seals of the hatch covers were possibly damaged when the deck cargo shifted resulting in water seeping into the holds.

According to the weather reports a standard storm was expected for the open sea area south of the Aland Islands. This presented no reason to deviate from the original passage plan since the reduction in the friction coefficients was not seen as a risk.

3.3 Other factors uncovered during the investigation

Any activity on board a vessel that has listed to 26° and is rolling a few degrees is difficult and trying on the stress tolerance of the crew. It was imperative to get help quickly and to be able to right the ship. In this sense, a speedier request for help would have been in order.

According to the pilots the piloting of the vessel from Utö to Turku was the most demanding task in their careers. The piloting of more than nine hours was taxing both mentally and physically. A dense fog settled during the voyage. Because of the list the radars echoed off the ice on the side of the list, which made the radar image cluttered or broke it off entirely at times. The situation was perilous, especially in the narrow waterways.

The authorities responsible for the safety of navigation should provide all the necessary support for pilots carrying out their task in corresponding danger situations in order to ensure that the accident vessel and her crew reach safety without any unnecessary risks. The use of technical positioning systems, escort by towing and sufficient monitoring support for the pilots must be introduced.

The investigation team has evaluated the securing of the deck cargo based on the friction coefficients between different surfaces in the stow. The method in the cargo-securing manual is based on the stability angle of the stack of sawn timber packages.

According to the cargo securing manual the information about the friction coefficient must be given among the other things to the Master. However, it is not shown, how one should apply the friction coefficients when planning the lashing or how they affect the stability angle of the stack.

Packaged sawn timber is a usual type of cargo in wintry conditions. However, the cargo-securing manual does not contain information of the effect of varying conditions on the friction coefficients or on the angle of stability of the stack.

Moreover, there is no complete example of the determination of the securing forces.

3.5 Measures for increasing the safety

During the investigation it has become obvious, that the tension forces of the securing wires are unknown in praxis. The tension has a profound effect on the stability of the deck cargo and hence on the safety of the vessel. The tension forces should be known during the whole voyage. In addition, the cargo-securing manual should contain means to determine the securing forces at various friction coefficients and heeling angles. Then the Master will have better possibilities to asses the safety of the securing of the cargo in varying friction conditions and sea states and plan his navigation measures.

During the investigation it has become obvious, that the way of giving the strength properties of the securing equipment is not appropriate. The situation should be corrected by issuing only the break load of the equipment in question. The final decision should be based on safety factor determined suitable for the situation.

During the investigation it has become obvious, that the friction coefficient of protective hoods of timber packages have a great variation. In order to improve the safety of the total logistic chain, it is of vital importance, that the friction properties should get certain tolerances. This should be based on wintry conditions and, which is most important, on the safety of people. Information on friction coefficients of the actual hoods should follow the cargo.

During the investigation it has become obvious, that the influence of the lifting ropes on the friction coefficients of the packages and on the tipping angle of the package stack is not known. This situation could be improved e.g. so, that some institute undertake the promotion of a national or international research project of this subject.

4 RECOMMENDATIONS

The safety risks of winter conditions that have become known in the course of this investigation should be minimised. Reporting of the identified factors to the Master of the vessel receiving the transportation should form into such a routine that the Master will automatically receive such information. It is the recommendation of the investigators that:

1. The freight carriers forward the friction information published in this report with the cargo to the Master of the ship loading it.

During the investigation it has become obvious that the cargo-securing manual of the vessel does not carry sufficient information and a clear calculation example for varying winter conditions. The investigators have observed a safety risk in the drawings on the placement of sawn timber that should at least be noted. It is the recommendation of the investigators that:

2. The information of friction coefficients of sawn timber packages in winter conditions and a clear calculation example should be included in the cargo-securing manual.

3. Figure 4.2.2 in the cargo-securing manual should be corrected in such a way that sawn timber packages are placed lengthwise at the sides.

A ship loading timber cargo in wintry conditions sails in most cases to warmer weather conditions. Then the friction coefficients of the cargo, which has been cold during the stowage, may develop unexpectedly. It is the recommendation of the investigators that:

4. The Finnish Marine Board take measures in order to inform internationally about the considerable decrease of the friction coefficients of timber cargo loaded in subzero temperatures and sailing in warmer zones.

Helsinki 21.10.2003

Pertti Siivonen — Kai Mäcklin

Kalervo Mattila — Olavi Huuska

LIST OF SOURCES

The following sources are filed at the Accident Investigation Board:

1. Cargo securing manual of the vessel, in Russian, partly in Finnish

2. Stability calculations for the vessel, by the Master

3. Extracts of the vessel drawings, showing the distribution of tanks, the air pipe data, the discharge/ballast pipe system and the dispatch note related to these

4. Memoranda drawn up during the investigation

5. Calculations performed during the investigation

6. Photographs

7. Data from the friction tests and their analysis

8. Weather reports from the Finnish Meteorological Institute

9. Maritime data from the Finnish Institute of Marine Research

10. Nautical chart drawings

11. Final report: Safety of Timber Coatings in Stowage Work, Carita Aschan, Erkki Rajamäki, Mikko Hirvonen & Tarmo Mannelin, Finnish Institute of Occupational Health, Department of Physics, Helsinki 21.12.2001

12. The Securing of Vehicles on Roll-on/Roll-off Ships, S R Turnbull and D Dawson, RINA 1995

13. Code of Safe Practice for Ships Carrying Timber Deck Cargoes

14. Cargo Securing Manual, ICHCA, 1998

15. Draft of an European Standard for road vehicles: Calculation of lashing forces, prEN 12195-1

16. Russian rules for the securing of cargo, part 2, volumes 1 and 2, St. Petersburg 1996, 1997, (in Russian)

17. In addition, the following sources have been used in the investigation:

    - Jerzy Matusiak, Vessel Flotation and Stability, Otatieto 557
    - Principles of Naval Architecture, SNAME 1988
    - John R Knott, Lashing and Securing of Deck Cargo, Third Edition, The Nautical Institute
    - Outinen, Koski, Salmi, Principles of the Strength of Materials, Pressus Oy 2000
    - Anderson P, et al: Safe Stowage and Securing of Cargo On Board Ships, Research Report, MariTerm AB, 1982

18. Various Internet addresses include reports on shifted timber cargoes

Enclosure 1. Stability calculations obtained from vessel

На отход: JAKOBSTAD
31/12/01

m/v Pioner Onegi   Date 12-31-2001   Time 00:03:29   File sawngood

Cargo items	Number	Weight	X	Z
Timber cargo hold 1		495.10	41.40	7.57
Timber cargo hold 2		659.77	25.70	5.12
Timber cargo hold 3		1027.48	5.40	5.15
Timber cargo hold 4		983.92	-18.30	5.28
Timber cargo deck F		312.50	24.28	11.48
Timber cargo deck A		533.16	6.43	12.03
Timber cargo deck 4		587.12	-18.48	12.60
Total Timber cargo		4599.05	5.47	7.61
Ballast		1001.60	15.33	1.06
Fuel		383.40	-6.95	1.80
Fresh water		192.50	-28.82	5.19
Oil		26.19	-39.36	3.57
Total liquids		1603.69	3.81	1.77
aw		5.00	-5.00	5.00
Stores		10.00	-55.00	10.00
Supply		75.00	5.00	5.00
Ice		130.15	-6.50	13.61
Deadweight		6292.75	4.94	6.09
Light ship condition		3936.60	-11.35	7.59
Displacement		10359.50	-1.40	6.76

FREE SURFACE CORRECTION (dMz) 1686.6 (tm) / 0.16 m
GRAIN CORRECTION (dMz) 0.0 (tm) / 0.00 m
WATER DENSITY 1.002
Added weight in tanks did not take into account

Signature MASTER [signature] ' 31 ' DECEMBER ~~199~~ 2001
2off. [signature]
Ch. off. [signature]

oner Onegi   Date 12-31-2001   Time 00:06:27   File sawngood

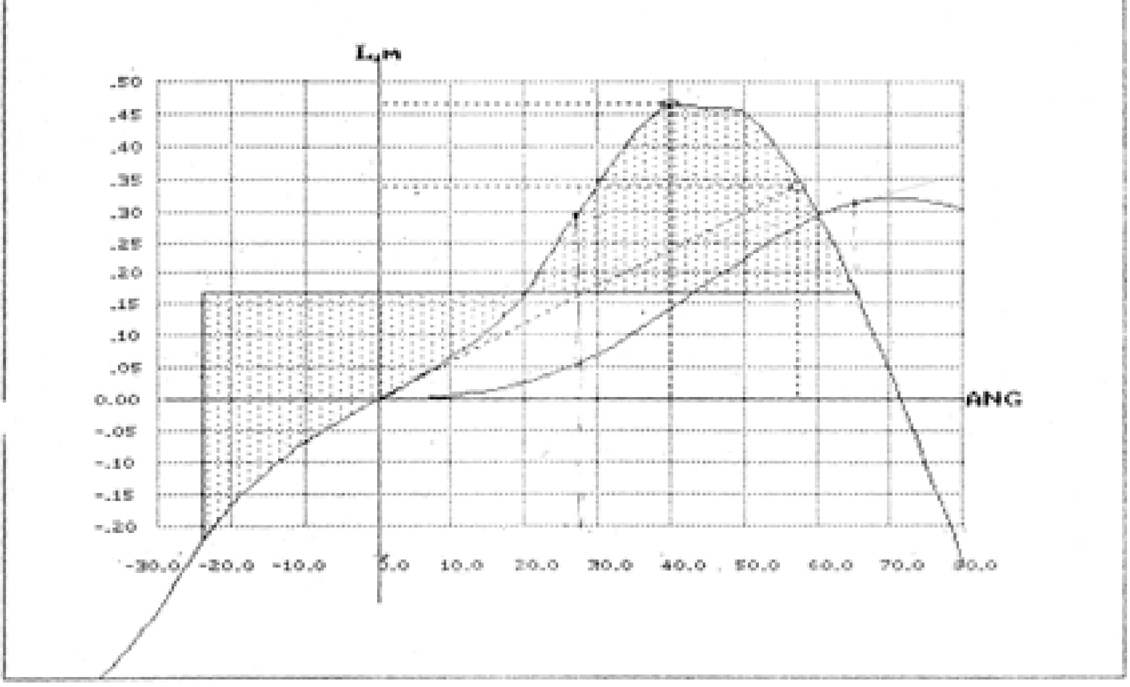
Приход SKIKDA 11/01

ICE CONDITION		
DRAUGHT MEDIUM	7.29	7,01
FORE DRAUGHT MARK	7.04	6,84
AFT DRAUGHT MARK	7.52	7,17
TRIM	-0.51	-0,35
METACENTRIC HEIGHT GMo, m	0.50	
METACENTRIC HEIGHT GM, m	0.34 > 0,21	0,31 > 0,14
Zg , m	6.92 < 7.09	6,87
Maximum Arm	0.47 > 0.25	0,53
Angle of max arm	40.00 > 30.00	49,6
Weather criterion	2.305 > 1.000	2,5
Angle of vanishing, deg	72 >55	75
Area under curve 0-40 , (m*rad)	0.149 > 0.080	0,161

HEEL	0	10	20	30	40	50	60	70
ARMS	-0.00	0.07	0.17	0.34	0.47	0.45	0.30	0.04

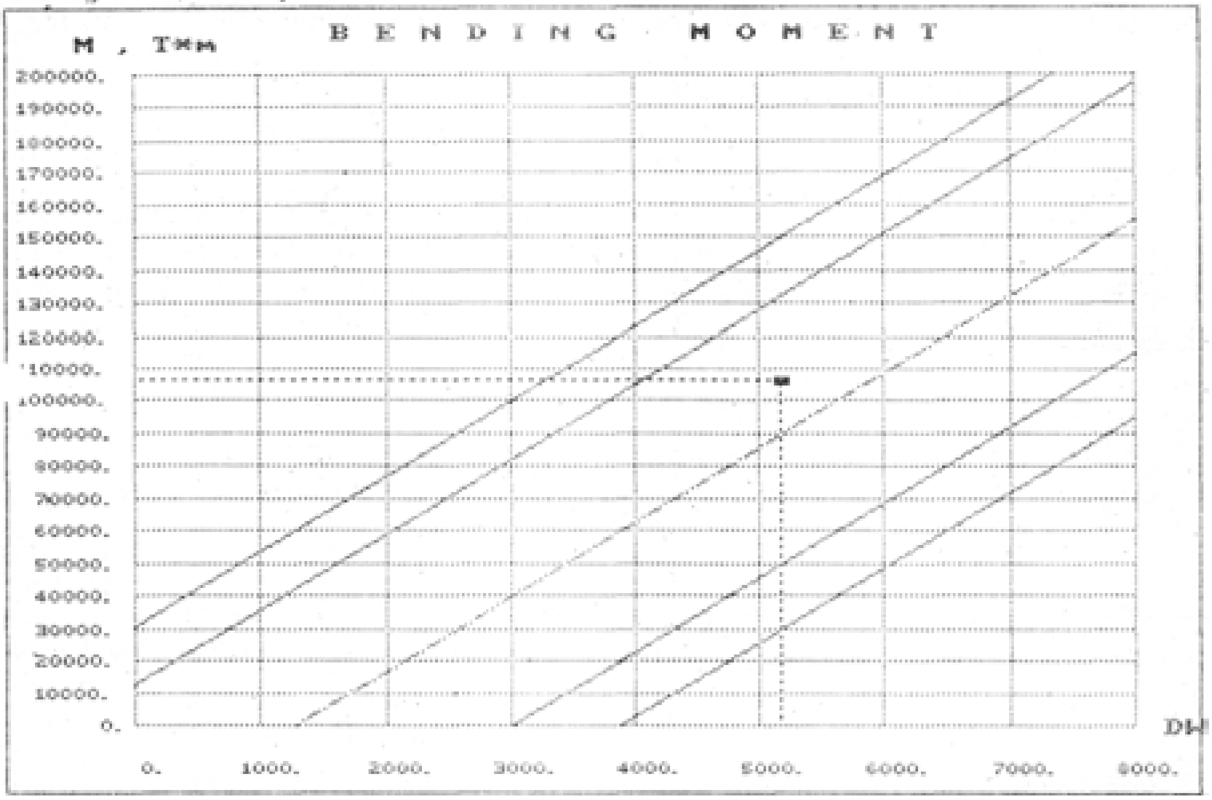
The summary trim and stability are SAFETY

STABILITY DIAGRAM



Pioner Onegi   Date 12-31-2001   Time 00:07:29   File sawngood

Actual moment, tm 106325
Deadweight ,t 5172
Strength is safety



Enclosure 2. Review of the securing of the deck cargo of the vessel

M/S FJORD PEARL, Securing of the packaged sawn timber deck cargo

The securing methods for a deck cargo of sawn timber are still in the development phase. There are inconsistencies in the IMO guidelines and the directions of the national authorities as noted e.g. by /Knott/. In practice, deck cargoes are secured according to the Cargo Securing Manual required by the IMO. Thus, the quality of this manual is very significant. The cargo-securing manual of the FJORD PEARL is quite thorough. However, it was observed in the course of the investigation that the manual needs some additions and comprehensive examples of the calculations.

The aim of the following presentation is to introduce the factors pertaining to the calculations for the securing of a deck cargo of sawn timber packages. The presentation does not aim at giving precise results.

1 The stability condition for a deck cargo

We shall mainly be looking at transversal forces since the longitudinal forces are usually clearly smaller. In this enclosure we will study a static situation. The dynamics resulting from the rolling have been included in enclosure 5. First we shall study transversal gliding.

The initial assumption is that the deck cargo in figure 1 remains static up to a heeling angle of θ_S. The friction coefficient between the deck cargo and the underlying surface is thus μ = tan θ_S. Then, the ship is heeled further to an angle θ_{DIN}. Now, an additional force K parallel to the underlying surface is required to hold the cargo static, giving

$$K = mg(\sin\theta_{DIN} - \cos\theta_{DIN}\ \tan\theta_S)$$

$$\Rightarrow K/\cos\theta_{DIN} = mg(\tan\theta_{DIN} - \tan\theta_S)$$

The following issue is to determine how force K is created. In this case, the pressure of the deck cargo against the underlying surface is increased by over top lashing. The friction between the deck cargo and the underlying surface creates the transversal force K.

2 Solid cargo unit, secured at the corners

We will look first at a solid cargo on the hatch covers when the ship is listed at angle φ. The deck cargo is secured to the deck from its upper corners. The tension forces are F_{t1} and F_{k2} and the difference to the vertical is marked by angle α. N is the supporting force of the underlying surface and μ is the friction coefficient between the cargo and the surface. A free body scheme (VKK) gives the clause for a static body in the direction of the surface of the hatch cover.

$$mg\sin\varphi \leq \mu\ [(mg\cos\varphi + (F_{k1} + F_{k2})\cos\alpha] + (F_{k2} - F_{k1})\sin\alpha$$

To simplify the situation, α = 5° from now on, which gives sin ≈ 0 and cos ≈ 1. The situation on the FJORD PEARL was this.

A cargo of sawn timber packages cannot be secured in this way, but the lashings go over the cargo and/or support posts are used.

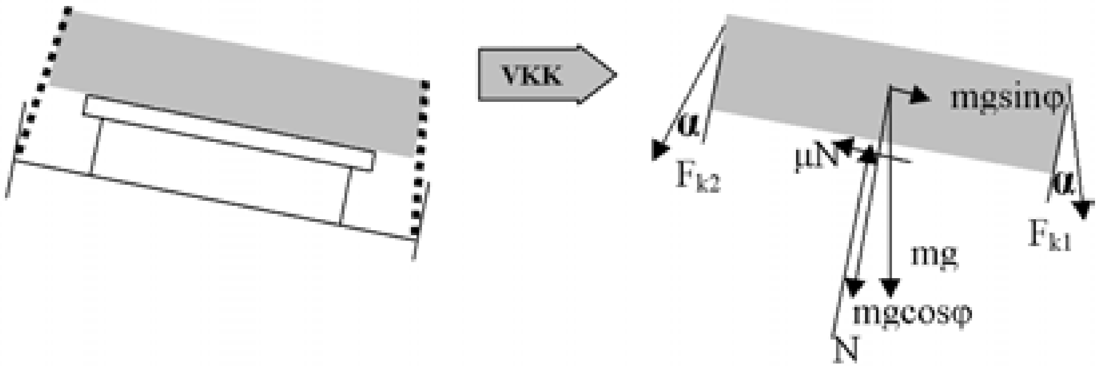


Figure 1. Solid cargo on cargo hatch, secured at top corners

3 Solid cargo unit, over top lashing

The securing of the cargo is changed in such a way that the securing wires go over the cargo, see figure 2. This is called over top lashing (securing based on friction). In addition to the case above, the bends of the securing wires at the corners of the cargo have an effect. The situation corresponds to rope or strap friction. After the first bend, tension $F_{k3,}$ calculated from the formula:

$$F_{k3} = F_{k1}\, e^{\mu 0 \cdot 85 \cdot \pi / 180}$$

After the second bend the tension of the securing wire is:

$$F_{k2} = F_{k1}\, e^{\mu 0 \cdot 170 \cdot \pi / 180}$$

When these tensions are taken into account, the following clause for a static cargo is obtained:

$$mg\sin\varphi \le \mu\,[mg\cos\varphi + F_{k1}(1 + k)]$$

where:

$$k = e^{\mu 0 \cdot 170 \cdot \pi / 180}$$

μ_0 = friction coefficient between the wire and the corners of the cargo

$$F_{k2} = k \cdot F_{k1}$$

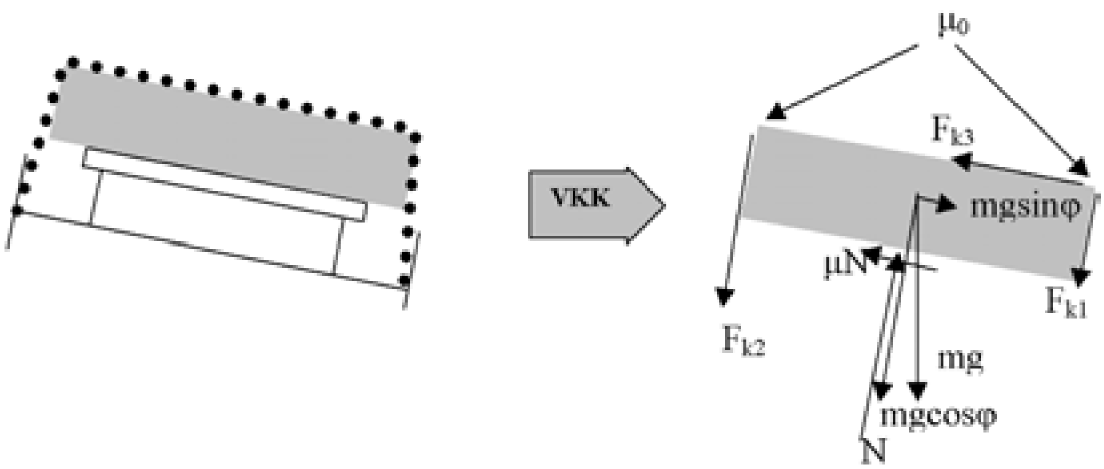


Figure 2. Deck cargo secured with wire going over it

In order to calculate the tension forces of the wires, μ_0 needs to be estimated. This can be assumed to depend on the tension of the securing wires and on the properties of the wire and of the corner of the cargo. When the tension increases, the wire is pressed more strongly against the corner of the cargo. The limit is the strength of the securing wire and the compression strength of the corner of the cargo as well as of the entire cargo.

When the securing wires are tightened, the rope friction has an effect in another direction. If the tensioning place is located on top of the cargo the tension forces on the sides can be calculated by dividing the on top tension force by coefficient $\sqrt{k}$. If the tensioning place is to one side, the force on the other side is calculated by dividing by coefficient k. When the wires are tightened on the top of the cargo, the securing is symmetrical. A symmetrical securing is created if both ends of the securing wire are tightened equally on the sides. A symmetrical tightening is a better option. In the case of the FJORD PEARL, the securing wires were tightened on the top. The investigation team has taken this into account in their calculations by reducing the vertical tension force correspondingly.

When the tightening and shifting cases are combined, one gets if Q marks the tension force just after the tightening:

$F_{k1} = \frac{Q}{\sqrt{k}}$ and the force affecting against the shifting:

$$\mu (1 + k)F_{k1} = \mu (1 + k) \frac{Q}{\sqrt{k}} \approx 2\mu Q$$

Consequently, the force caused by the tension force of the wire and parallel to the hatch cover (force K in Section 1) is nearly equal to 2μ Q. Therefore:

$$2\mu Q/\cos\theta_{DIN} \approx mg(\tan\theta_{DIN} - \tan\theta_S)$$

Term $2\mu/\cos\theta_{DIN}$ is usually slightly smaller than 1. In the cargo securing manual angle θ_S is the static stability angle of the stack, which is normally smaller than the angle corresponding to the friction coefficient μ . Therefore, the multiplier of the term mg will increase. As a result, we get the formula in the cargo-securing manual.

$$Q = mg(\tan\theta_{DIN} - \tan\theta_S)$$

It includes some safety reserve, when Q is the tension force of the wire on top of the deck cargo. On the other hand, this reserve is needed because of the ambiguous situation. The draft European standard for the calculation of lashing forces /prEN 12195-1/ has a similar equation, which uses accelerations instead of angles.

However, one should use the friction angle instead of the static stability angle of the stack, if the former is smaller than the latter, see 4.2.

4 Lashed cargo of sawn timber

4.1 Solid packages

The situation is more complicated; when the body is formed of packages, see figure 3. In addition to the gliding, tipping of the stack becomes possible. Each package should be looked at separately with its free body scheme. This would lead into complicated calculations. Moreover, the amount of initial data is large and the data is inaccurate. The situation could be simplified e.g. so that only the package stacks furthest to the side are considered to be fully under the effect of the vertical pressure of the securing wire. We will study in more detail only the gliding of the cargo. Tipping is taken into consideration based on experimental stability angles of the stack issued in the Russian rules of the securing of cargo.

It is enough to look at the tiers above the level of the hatch covers since the shifting of cargo between the hatches and between the hatch coamings and the gunwale is in practice blocked. If the mass of the stack furthest to the side and above the hatch level is m_1, the mass of the stack on the hatch is m_2, and the corresponding friction coefficients are μ_1 and μ_2, the equation in the direction of the level of the hatch covers is:

$$(2m_1+m_2)g\sin\varphi \le \mu_2[m_2 g\cos\varphi + F_{k1}(1 + k)] + \mu_1[(2m_1 g\cos\varphi + F_{k1}(1 + k)]$$

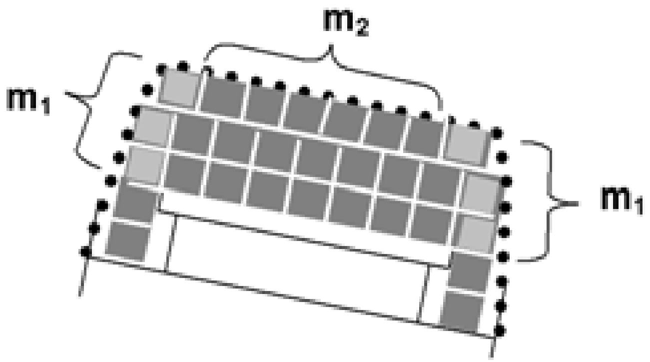


Figure 3. Sawn timber cargo on the hatch cover

The friction coefficients μ_1 and μ_2 are usually different because the materials under the lowermost tiers of the masses are different. In the case of the FJORD PEARL the bottom wood surface of m_2 was lying on steel whereas the lower wood surface of m_1 was lying on a hooded wood surface. In addition, both stacks had the lift ropes in between the surfaces.

This equation has been used in determining the maximum allowable rolling angle for a static cargo at various friction coefficients. Based on the information collected on friction coefficients, $\mu_1 = \mu_2 + 0.15$ was used in the calculations. In addition, the dynamic factors and the wind pressure were included. Both of these factors are studied further in enclosure 5.

The rope friction coefficient is in many cases 0.4-0.5, but because of the reasons listed in Section 3, it is estimated to have been significantly lower in this case. Table 1 presents the estimated friction coefficients and the calculated coefficient k.

Table 1. Estimated friction factor for the wire friction and calculated k value

Tension [kN]	μ_0	**k**
40	0.12	1.428
60	0.16	1.608
80	0.20	1.810

The tension force of a securing wire may increase by 30-40% at the very start of the shift in comparison with the tension force on top of the deck cargo before the shift.

4.2 Deck cargo of sawn timber packages

A package of sawn timber is an example of a cargo unit, where its internal structure must be taken into account. Gliding surfaces between the planks are formed. So far these have been ignored. Planks are stacked mechanically to form a package having a cross-sectional area of $1.1m \cdot 1.1m$. The package is surrounded by two or three steel bindings, which are tightened. Before the bindings are installed a hood often covers the package. Such a package has quite many friction surfaces, figure 4.

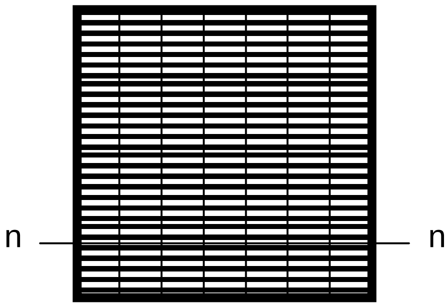


Figure 4. A package of sawn timber, tension forces of the steel bindings compress the package, the uppermost and lowest planks as well as planks on both sides are compressed most, the friction coefficient for dry wood is about 0.5, only one possible gliding surface, n–n, is shown, a package under pressure (by securing force and mass of packages over it) will compress which may decrease the tension in the bindings

When packages are stacked on each other, the stability of the stack comes into picture. Experimentally it is possible to find out the maximum angle, at which the stack is static. This depends on the friction coefficient and on the tipping angle of the stack. Tests have been carried out at various amounts of tiers and at different ways of stacking. The Russian rules give results of such tests for dry wood. Stacks were not lashed and without lifting ropes and protective hoods.

For the packages stacked lengthwise, the stability angle is 23°, 16°, 11° and 0° when the height of the stack is 1, 2, 3 and 4-7 tiers, respectively. These angles are smaller than the theoretical ones for an ideal stack, due to the fact, that in reality, the packages are not similar and the stacking is not ideal. These angles apply only if the corresponding friction angle is greater. If the friction angle is smaller, it must be applied.

The angle θ_S in the cargo-securing manual is the static stability angle of the stack. It is converted to the dynamic stability angle of the stack by taking the rolling into account. The static stability angle of the stacks nearest the case of FJORD PEARL is 17°. Due to the ropes and hoods this angle should be smaller. Instead of $\tan\theta_S$, one should apply friction coefficient, if it is smaller than $\tan\theta_S$.

4.3 The friction coefficient used in the investigation

If we try to consider the deck cargo of packaged sawn timber as one entity instead of one stack, the situation will be too complicated for this study. Therefore, studying the conditions of the shifting of the deck cargo the investigation team has used as a variable the friction coefficient between hatch cover and the package. The interconnection of this coefficient and the angle of static stability of the stacks has not been studied more than above.

5 Tension force of the wire

The following strength data of the securing elements has been collected from the cargo-securing manual. The diameter of the securing wires must be at least 16 mm (requirements of the IMO guidelines on timber cargoes of enclosure E in brackets).

Table 2. Strength data of the securing elements

Name	pcs	SWL, kN	BL, kN (kg)
Securing wire 14 m	88	77.5	More than 133 (13600)
Securing wire 8 m	8	77.5	More than 133 (13600)
Chain 3 m	48	54	More than 133 (13600)
Rigging screw	59	34.3	More than 133 (13600)
Brackets on deck	44	50	More than 138,3 (14100)

BL is the break load; SWL is the safe workload. Instead of SWL, a better term would be MSL, maximum securing load. The actual break load is not known. The securing equipment is Russian. It is possible that the BL is more than 133 kN, may be up to 200 kN.

The only strength value issued for a cargo-securing element should be the break load (BL). The user could then pick a suitable safety factor according to the case. For example: the deck cargo should go over board when the listing angle of the vessel exceeds 40°, but the cargo in the hold should not shift.

The actual tension force remains unclear because it was not measured on board. The initial tightening and subsequent check-up tightenings were performed manually at sea, based on experience. At sea the cargo packs as the vessel moves and vibrates and the tension forces of the wires decrease. Therefore, the wires were retightened twice a day, according to the instructions in the cargo-securing manual. The last retightening before the accident had been performed six hours before, at about 20:00. During this time the securing might have already slackened slightly. It is probable that different wires have had different tension values. This is why the calculations were made for various wire tensions and without tension. The results of the calculations are presented in enclosure 4.

When a cargo of sawn timber packages shifts, it is probable that the perimeter of the load will increase, which, in turn, increases the tension force of the wire and thereby the force resisting for shift. On the other hand, the packages are compressed. The result is a new equilibrium. The true tension forces are unclear since the structure of sawn timber packages is constantly reforming.

6 Calculation of the securing of the deck cargo of the FJORD PEARL

The weight of the shifting cargo is estimated to be 1250 t. The formula gives Q = $1250 \cdot 9.81$(tan 34° - tan 16,5°) = 4639 kN. There were 44 over top securing wires across the ship. In addition, six pairs of diagonal wires had been drawn over the deck cargo of which four may be included due to the diagonal position. This raises the total number of securing wires to 44 + 8 = 52. The resulting necessary tension force is 89 kN. If the calculation is carried out without the diagonal wires, the result is 105 kN.

According to the table 2, these forces seem to be quite high, giving the safety factors of 1.5 and 1.25 for minimum break load. The investigation has not been able to determine the actual tension force nor the break load of the securing equipment.

The calculations made in the course of the investigation are approximate and contain inaccuracies. The fact that the number of package tiers on the hatch covers was in parts three and in parts four has not been included in the calculations. The tension force should be greater in the area of the higher stack or the wires should be placed at smaller intervals. The estimated mass of the shifting cargo, 1250 t, may be inaccurate. Moreover, the underlying train of thought behind the cargo-securing manual is not known. Nor is it known how the cargo securing calculations were made on the FJORD PEARL.

In summary, it seems that the cargo-securing manual requires clarification and additions. The manual does not contain a clear and complete example of the lashing calculations. Data of interconnection between friction coefficient and the static angle of stability of the stack should be added. It has not been stated, that the tension force should be verified nor a method for that. The calculations should base on the break load of the securing equipment and a safety factors according to the actual case. A form for the lashing calculations should be included. Port authorities should check these.

Enclosure 3. Study of vessel stability

M/S FJORD PEARL, study of vessel stability

The stability characteristics of a vessel are determined in such a way that she will not capsize in all design load conditions. The minimum criteria for the stability characteristics are given in the international and national regulations. Under normal operating conditions it is enough to fulfil these stability criteria but this may be inadequate in an accident situation.

The stability of M/S FJORD PEARL with regard to the regulations, the change in the stability characteristics in the waves and the consequences of the cargo shift on the stability are reviewed below.

The momentary rolling of the ship in connection with the shifting of the deck cargo is here called toss.

1 Calculations performed on board

The stability calculations obtained from the ship were appropriate and showed that the stability of the vessel was within the limits of the regulations. The initial metacentric height corrected by the effect of the free liquid surfaces was 0.34 m, whereas the requirement is 0.20 m. The statical and dynamical righting arms are presented in figure 1 below. They have been calculated according to the regulations and fulfil all criteria. The criteria are shown, too. The righting arms calculated on board the vessel after the accident are also included in the figure. The investigation team has performed stability studies for various situations based on these curves.

The weather criterion was calculated and compared to the requirements on board. The weather criterion, which is the ratio maximum heeling moment/capsizing moment was 2.3; which is good. When calculating the weather criterion, it is assumed that the waves cause the vessel to roll from one side to the other around a permanent list caused by constant wind. In addition, the heeling effect of the wind gust is included. The wind speed used in the calculations depends on the area of sailing. In this case the area of sailing was unlimited and the wind pressure about 1100 Pa, which corresponds to a wind speed of approximately 43 m/s. The calculations assume that the centre of gravity of the vessel lies on her centreline.

If the centre of gravity of the vessel does not lie on her centerline, the vessel has a permanent list in addition to which the wind, the rolling and the wind gust all have an effect. The effect of the heeling moment is usually presented by correcting the statical righting arm by an arm corresponding to the heeling moment multiplied by the factor cos (heeling angle). The situations corrected in this way and corresponding to heels of 22° and 26° are presented in figures 5 and 6. Since the vessel is rolling around the new equilibrium, the curve provides an opportunity to study the stability of the heeled ship.

When a deck cargo of timber shifts, this may create extra righting volume (righting moment) to the side. The stability calculations should not be based on this but it is a factor to be considered in the analysis of the accident. The permeability of the extra volume is 25% as that of the rest of the sawn timber on deck. It becomes wet gradually which reduces its righting effect. According to figure 2, the extra volume begins to have an effect at lists of more than 22°. This effect is still small, about 75 tm at a list of 26°.

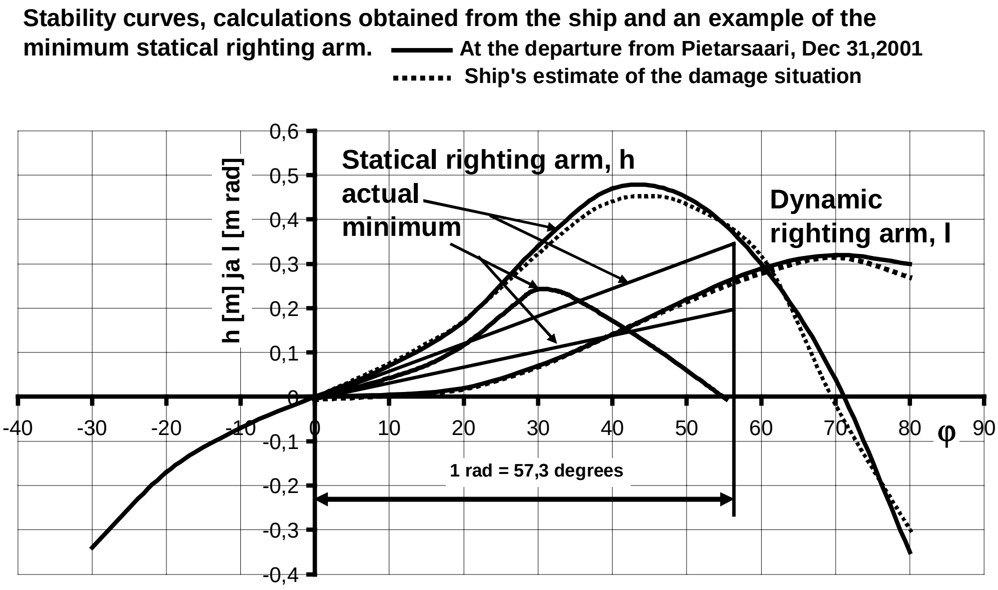


Figure 1. *Stability curves calculated on board the vessel. The figure also includes an example of the minimum curve for the static righting arm. The figure shows the slope of the curves at origin, which gives the initial metacentric height at a heel angle of 1 radian.*

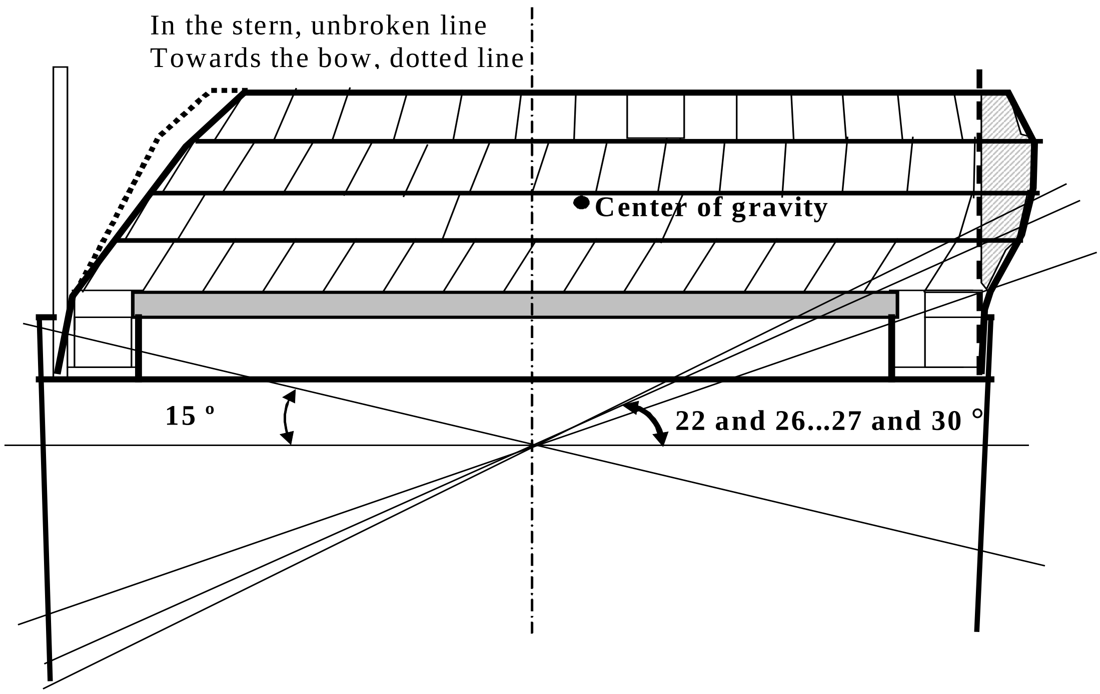


Figure 2. *Shifted deck cargo, the shaded area is the extra volume of the deck cargo compared to the cross curve calculations, the change in the shape of the sawn timber packages may be exaggerated*

2 Factors reducing the stability

The following factors change the above stability curves in an accident situation similar to FJORD PEARL.

1. Position of ship in relation to the wave (contributed to the toss), figure 3, case 1

a: Wave crests at the ends of the ship
b: Still water
c: Wave crest at midship

The stability curves shown correspond to a situation in still water. When the ship is in the waves, the stability is in turns better and worse than in still water. When the ship is in head or following seas so that her centre is on the crest of the wave the situation deteriorates. Normally a ship meets the waves at a varying angles and the stability is not compromised for long periods. FJORD PEARL proceeded in beam seas before the accident and started then to turn in quartering seas.

The speed of the vessel in the direction of the waves was clearly smaller than the speed of the waves. Thus the ship reached the crest of the wave for a short time but many times. It is possible that her stability was momentarily compromised which contributed to the toss of the vessel. On the other hand, the effect was reduced by the fact that the vessel met the waves diagonally.

2. Shift of the centre of gravity of the vessel away from the centre axis (had effect after the cargo shift), figure 3 case 2

Lower curve: centre of gravity of the vessel not on centerline. In the case of timber deck cargo, the shift may result in extra righting moment on the side. The effect is depicted with the dotted line. Figure 5 shows the static righting arm of FJORD PEARL heeled 22°.

3. Change in the draught of the vessel because of the accident (had effect after the cargo shift), figure 3 case 3

According to the calculations performed on board the ship, the draught of the FJORD PEARL was reduced by about 8 cm in the final situation due to the deck cargo, the list and the shape of the vessel. This had a slightly deteriorating effect on the stability.

4. Changes in the free liquid surfaces (had effect after the cargo shift)

Water accumulated in to some narrow tanks. Their deteriorating effect on the stability was very slight. Water started to accumulate also in holds 2, 3 and 4. These were filled with cargo, so no free liquid surfaces were created.

5. Vertical shift in the centre of gravity of the vessel (had effect after the cargo shift)

Some of the water was accumulating in the double bottom, some in the holds, some in the side tanks and some on the deck. Ice formed on the deck, on the structures of the vessel and on the deck cargo. According to the calculations, the centre of gravity was raised by about 5 cm. Thus, the stability curve must be corrected by the factor -5 cm x sin(listing angle).

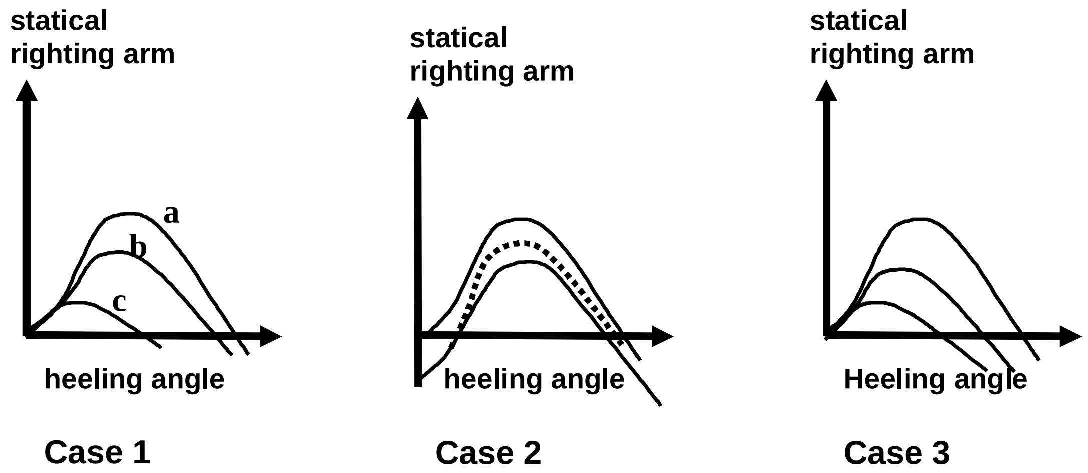


Figure 3. Change of stability curves in the various situations

3 Shifting of the cargo and stability

The shifting of cargo is a dynamic event, which is here handled approximately in separate stages in the following.

According to the reports of the officers of the vessel, she had a list of about 5° to starboard due to the wind. Preceding the toss the vessel listed 15° to port, and then heeled and tossed to 30° to starboard. After a while, she righted to 22° and after a few hours stabilised at a heel of 26-27°.

It can be concluded that the static heeling moment corresponded initially to a 22° list. The momentary toss to 30° was caused by dynamic factors. The end situation, 26-27° corresponded to the new static situation mainly due to the water that had accumulated in the tanks and in the cargo holds.

Tossing of the vessel

The rolling about the longitudinal x-axis through the centre of gravity of the vessel is handled using differential equation of roll, which will include:

- Moments resulting from the moments of inertia and the inertia forces
- Moments resulting from damping
- Static moments
- External moments
- Internal moments

The toss was a result of many ambiguous factors; therefore, the boundary conditions for solving the differential equation based on the above moments are too vague. Instead, the accident is examined with the static stability curve.

Let us first assume, that the deck cargo will not shift. Based on stability curves in figure 4, maximum heeling angle will be known. The shifting of the deck cargo took place before that angle was reached during the toss. We have two possibilities, see figure 4:

1. Maximum heeling angle due to waves and constant wind is about 20°.
2. Maximum heeling angle of due to waves, wind and gust is about 26°.

It can be concluded, that the shift of the deck cargo started at an angle between 16° and 25°, probably at 19-22°. Dynamic effects of the shift tossed the ship over this limit to 30°.

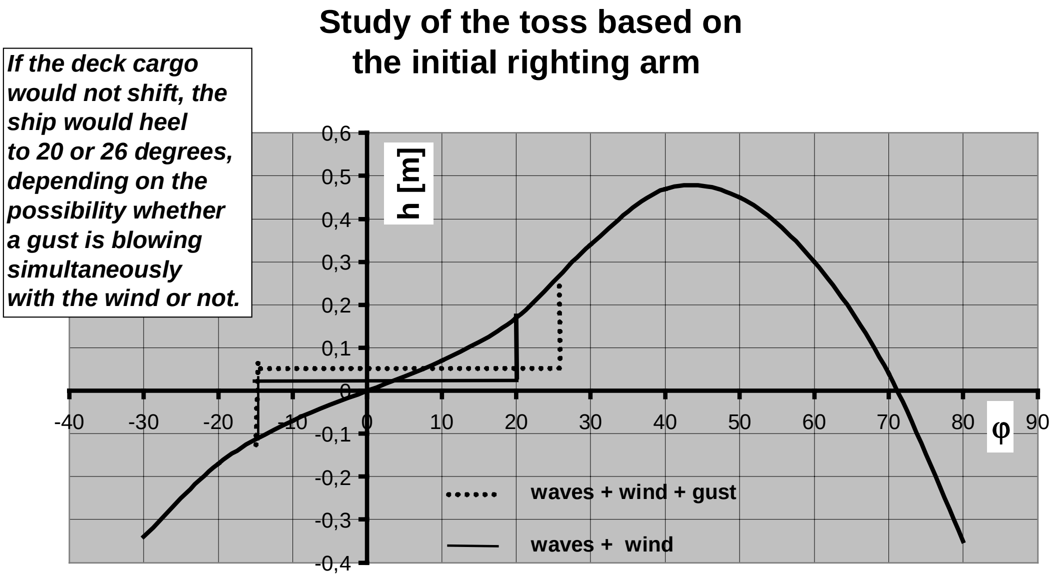


Figure 4. Maximum roll angle to starboard after the rolling 15° to port if the deck cargo had not shifted

Dynamics of the toss

In the waves the vessel heeled 15° to port and with the resulting potential energy and wind, back to starboard to about 20°. There was probably a simultaneous gust, not more than 27 m/s. In addition, the following dynamic factors could have played a role:

- The vessel was in quartering seas and crossed a wave a few times, which may have caused rotation around the vertical axis (yawing).
- A disadvantageous position on the wave slope may have contributed to the tossing of the vessel.
- The roll in the waves may have been quicker than what can be deduced by the rolling period of the vessel.
- As the vessel was crossing the wave diagonally, the righting arm could be reduced which resulted in a greater toss.
- The kinetic energy of the shifting cargo was absorbed by the work required by the additional heeling, by friction, by elongation of the wires, by breaking uprights etc Moreover, the cargo in the hold stopping against the side of the hold and packing there also carried kinetic energy.

Permanent list, situation shortly after the roll

Based on figure 2, the centre of gravity of the cargo shifted sideways about 0.8 m according to the geometric reasoning. Let us estimate that the squeezing and packing of the packages and ropes to the starboard side caused an additional 5 cm shift in the centre of gravity. The wetting of the lowermost packages and the ice and water on and inside them increased their weight and created a heeling moment (app. $8m \cdot 50t$) which is estimated to have caused a shift of 0.2-0.3 m making the total shift 1.05-1.15 m. The heeling moment caused by the shift when the weight of the deck cargo plus the additional weight is 1483 tonnes is approximately $1.1 \cdot 1483 \cos 22°$ = about 1512 tm.

According to the static stability curve corresponding to the initial state, about 2070 tm is needed for a list of 22°. In addition to the shift of the deck cargo, e.g. the following factors increased the list:

- Packing of the cargo in the holds to one side maybe about 35 cm $\Rightarrow$ $3166 \cdot 0.175 \cos 22°$ = about 510 tm
- The slight increase in the draught may lead to a slight deterioration in the cross curves.
- Wetting of the deck cargo and icing raise the centre of gravity of the vessel 1-2 cm.

The total heeling moment in the initial state is 1512 + 510 = 2022 tm. The difference can be attributed to the inaccuracies in the above estimate and the basic data. Figure 5 presents the statical righting arm for this situation.

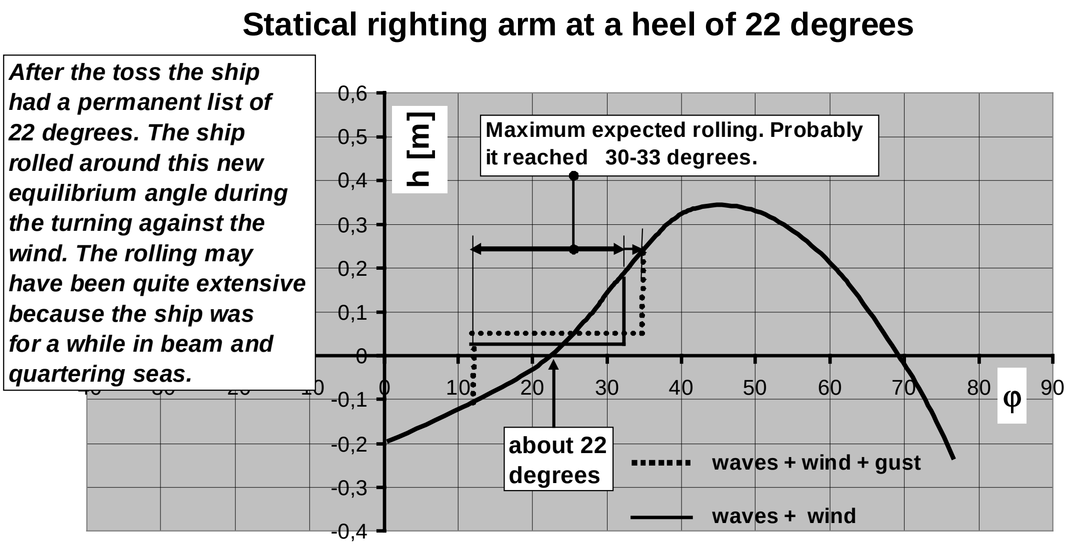


Figure 5. Statical righting arm at a heel of 22°

After the toss, the bow of the vessel was turned against the wind. During the turn, the vessel may have heeled to slightly more than 30°. When she was in head seas the rolling subsided, but remained at a few degrees because of the cross waves.

The final static state of the list

After a few hours the vessel reached a new equilibrium with a list of 26-27°. The list had increased mainly because of the leaks and could be stabilised with the pumps.

According to the curve in figure 5 a list of 26-27° requires a further 930 tm of heeling moment. The additional factors compared to the 22° list include e.g.:

- Water leakages into holds 2, 3 and 4, $6m \cdot 100t$ = 600 tm.
- Water leakage into the starboard dry tank, $8m \cdot 30t$ = 240 tm.
- Water leakage into fuel tank TT N 2, $6m \cdot 30t$ = 180 tm.
- Water leakage into diesel fuel tank N 2, $8m \cdot 25t$ = 200 tm.
- Pumping of water into the ballast tank on the other side making the compensating moment -400 tm, weight increases about 55 t.
- Further packing of the cargo, for example 10 cm on deck and 5 cm in the hold, combined effect about 70 + 80 tm = 150 tm.
- Formation of ice was not strong but slightly more ice formed on the listed side, estimate 100 tm (weight increase about 20 t).
- Wetting and further icing of the deck cargo, estimate $7m \cdot 10t$ = 70 tm.
- Shifting of deck cargo outside the gunwale created more righting volume, estimated effect 75 tm (additional displacement 10 t).
- The statical righting arm deteriorated slightly.
- Vertical centre of gravity was raised by 2-3 cm as a result of these changes.

The total increase of the heeling moment = (600 + 240 - 400 + 180 + 200 + 150 + 100 + 70 - 75)cos26° = 958 tm. The difference, 958 - 930 can be explained as in the above. The statical righting arm corresponding to this situation is presented in figure 6.

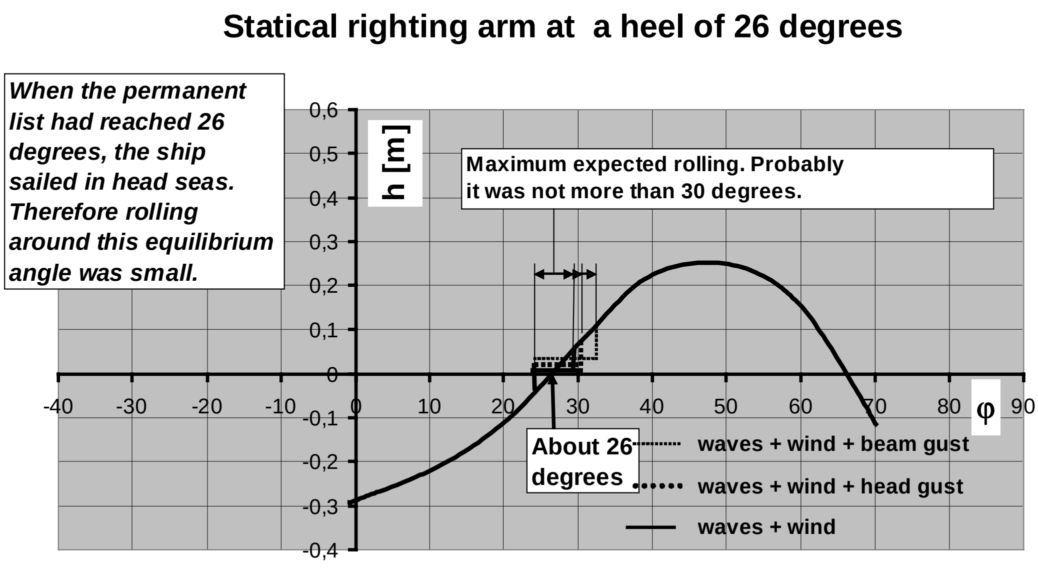


Figure 6. Statical righting arm for a heel of 26°

The list of the vessel can be explained with these approximations. The vessel was rolling to both sides of this equilibrium, finally only a few degrees since she was in head seas.

The displacement of the vessel grew by an estimated 50 t (icing in the first stage + wetting)+ 100 t (holds) + 30 t (dry tank) + 55 t (port side ballast tank) + 25 t (diesel fuel tank)+ 20 t (additional icing) + 10 t (additional wetting and icing of cargo) + 10 t (volume increase of the deck cargo), a total of 300 t, giving a displacement of 10660 tonnes. It is close to the result of the calculations performed on board, 10592 tonnes.

According to the calculations performed on board, the draught was reduced and the stability deteriorated slightly. The table 1 below presents some data.

Table 1. Some data at departure from Pietarsaari and after the accident

Displacement	Stern draught	Average draught	Bow draught	List	Centre of gravity	
10360 t	7.52 m	7.29 m	7.04 m	0°	6.76 m	Departure
10592 t	7.50 m	7.21 m	6.91 m	25.7°	6.81 m	Accident

4 Evaluation of the stability of the damaged vessel

The stability of the vessel is evaluated at a 26° list. The analysis is based on the righting arm in figure 6.

Rolling due to the wind gusts

In head seas the vessel was rolling a few degrees in the cross waves. The resistance coefficient of the wind surface when the wind is blowing from the bow is 0.2. If the wind comes directly from the bow, the resistance coefficient is zero, but a gust may enter at some small angle from the side. If a gust comes in directly from the side for some reason, the resistance coefficient of the wind surface is 1.2 giving a heeling moment:

$$0{,}5 \cdot 1{,}225 \cdot 27^2 \cdot 1044 \cdot 1{,}2 \cdot 5{,}2 = 2909 \text{ kNm} = 297 \text{ tm}$$

Air density = 1.225 kg/m^3, wind surface = 1044 m^2, resistance coefficient = 1.2 and the lever of the wind surface = 5.2 m. The righting arm corresponding to this moment is 297/10592 = about 3 cm. Figure 6 shows that the vessel lists to about 33°. Such a list is still not dangerous. Therefore, if the vessel remains against the wind, the gusts do not present a danger to her.

On the other hand, she could heel to about 40° in beam wind. At that heel, loosing the deck cargo overboard is possible.

Possible accumulation of ice

The warning for icing was still standing after the accident, so it was possible that more ice formed on the vessel. The accumulation of ice has two effects: Firstly, it raises the centre of gravity, which reduces stability. Secondly, more ice forms on the listed side, which increases the list. The effect of 200 tonnes of new ice is looked at in the following. The ice is placed at a height of 12 m from the base line and at a distance of 3 m from the centerline on the listed side. The result is the curve marked with a broken line in figure 7. After the added ice the new heeling angle would be 30° and the vessel would roll to 34-36° depending on the force of the wind plus gusts when the vessel is in head wind.

The passage of the vessel to Utö was against in head seas, so the rolling was only a few degrees. The passage from Utö to Turku was in a beam seas but the wind came from the side of the list. The effect of the wind was weakening as the wind speed was slowing down and the vessel was nearing the coast. Because of these reasons, the icing became less and the rolling subsided.

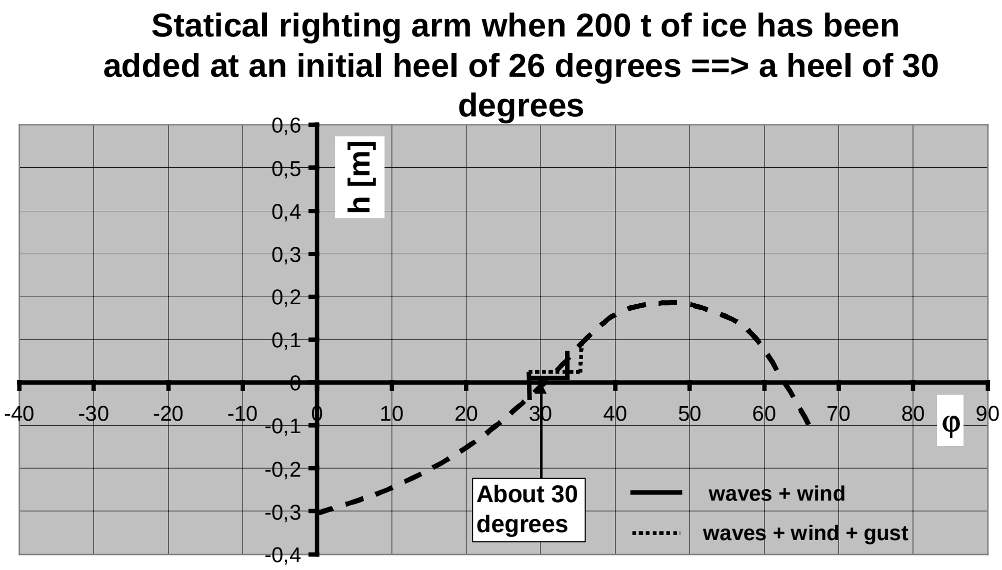


Figure 7. Approximated situation if 200 tonnes of additional ice accumulate after the accident

5 Summary

In summary it can be concluded that the load condition of the vessel at departure from Pietarsaari was such that the stability reserves were sufficient for this accident.

The deck cargo of sawn timber began to shift when the vessel heeled to 16-25° during the turn. The shifting cargo tossed the ship to 30°.

Enclosure 4. Friction review

M/S FJORD PEARL, Friction and friction coefficients

1 Friction forces

Friction force R between the cargo and the underlying surface resisting the sliding of the cargo can be calculated from the formula R = $\mu \sum N_i$ where all the supporting powers affecting the sliding surface vertically are summed and the result is multiplied by the friction coefficient. These supporting forces and the friction coefficient as well as any other possible forces resisting the sliding must therefore be known in order to determine the total force preventing the sliding.

No comprehensive mathematical theories exist on friction. A good part of the theory rests on empirical results. Based on these, it is assumed that the friction coefficient obeys the following ”laws”:

1. The coefficient does not depend on the area unless the area is so small that the body on top actually penetrates into the body below, in other words the pressure becomes too great.
2. The coefficient does not depend on the kinetic velocity in an extensive velocity range.
3. The coefficient does not depend on the pressure force; see "law" 1.

The coefficient depends on the quality of the contact surfaces and on the matter in between, if any. If the surfaces are dry or nearly dry, the friction is known as dry friction. This coefficient diminishes to a certain limit as the surfaces become smoother but begins to grow again if the surfaces are smoothed further to the limit of ”fusing” caused by adhesion. If a liquid substance collects in between the surfaces the situations nears lubrication and the friction is known as viscous friction. The friction coefficient can be determined for various situations, for example for wet, dirty or greasy surfaces. The other extreme is a glued joint, an example of which is two surfaces frozen together. In the case of the FJORD PEARL, the surfaces where dry in places, icy or frosty in other places, watery in some areas and in some instances could also be frozen together.

There are usually two different friction coefficients in use. The static friction coefficient μ_s is determined by equation R $\leq \mu_s \sum N_i$. Its limit value is the static friction coefficient. When the bodies are sliding with regard to each other, the dynamic friction coefficient takes effect, determined by equation R = $\mu_k \sum N_i$.

In principle, the static and the dynamic friction coefficients are the same but in practice the surfaces are already made smoother by the initial shift, which is why the static friction coefficient is usually bigger than the dynamic friction coefficient in tests.

Usually, the friction coefficient is determined with the following methods:

1. The surface on which the body rests is tilted. The tilting angle at which the body starts to shift and another angle at which it moves at a steady speed are measured. The tangent of the tilting angle = friction coefficient.
2. The body on the surface is pulled or pushed with and increasing force. The forces at which the body starts to move and at which it is in steady motion are measured.

The testing schemes of these methods are laid out in figure 1. Method 2 typically gives a curve presented in figure 2 as the result. The static friction is the ratio shifting force/pressure force.

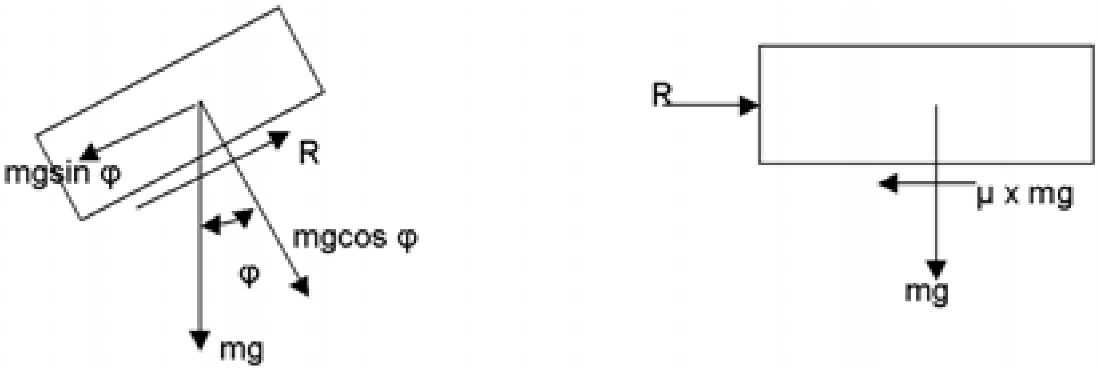


Figure 1. Determining the friction coefficient

1. Tilting: at equilibrium $\mu = \tan\varphi$, where μ is the friction coefficient between the cargo and the underlying surface and φ is the tilting angle. This method can be used to determine the stability angle or internal friction coefficient of a body consisting of parts.
2. Pulling or pushing on a horizontal surface: R = μ mg, where R is the force at which mass m starts to shift or at which it moves at a steady speed

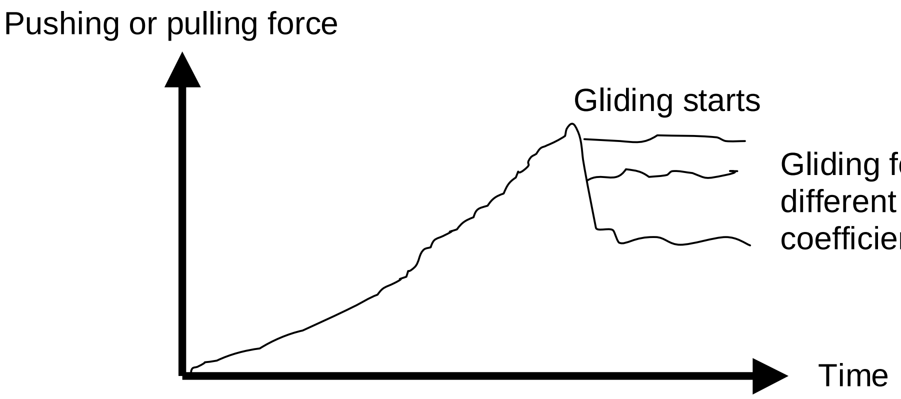


Figure 2. Typical result of a friction test

2 Value of the friction coefficient

Various different friction surface types were created as the cargo was placed on board. In addition to the surfaces of the timber packages contacting one another and to the horizontal friction surfaces between the hatch covers or the deck and the packages, also vertical friction surfaces were created in between the packages. These are known as external friction surfaces. In addition, there were horizontal and vertical friction surfaces in the timber packages themselves. The underlying surface of the cargo was level only on the cargo hatches and on the deck. The friction review in between the tiers becomes complicated due to the small differences in the dimensions of the packages and to their various positions. In this case, the movement between the tiers is not restricted only by the friction but by the various bumps and corners of the packages as well. Increasing the friction coefficient between the corresponding level surfaces can approximate the net effect. The following review begins by a review of external, level friction surfaces.

The situation was complicated further by the lifting ropes around the packages and by the protective hoods. The ropes crossed each other in some spots. There were several types of hoods. Usually the top and bottom surfaces of the hoods were different. Therefore, the friction coefficients between the top surface of the hood and the sawn timber above were different from the friction coefficients between the bottom surface of the hood and the sawn timber also protected by a hood.

Figure 3 presents the main combinations of two horizontal surfaces.

1. Two packages lengthwise stacked staggered on top of one another
2. Packages stowed lengthwise on top of packages stowed across
3. Packages stowed across on top of packages stowed lengthwise
4. Packages lengthwise on top of a cargo hatch
5. At the sides lengthwise packages stowed evenly on top of one another
6. At the sides lengthwise packages on top of ground timber

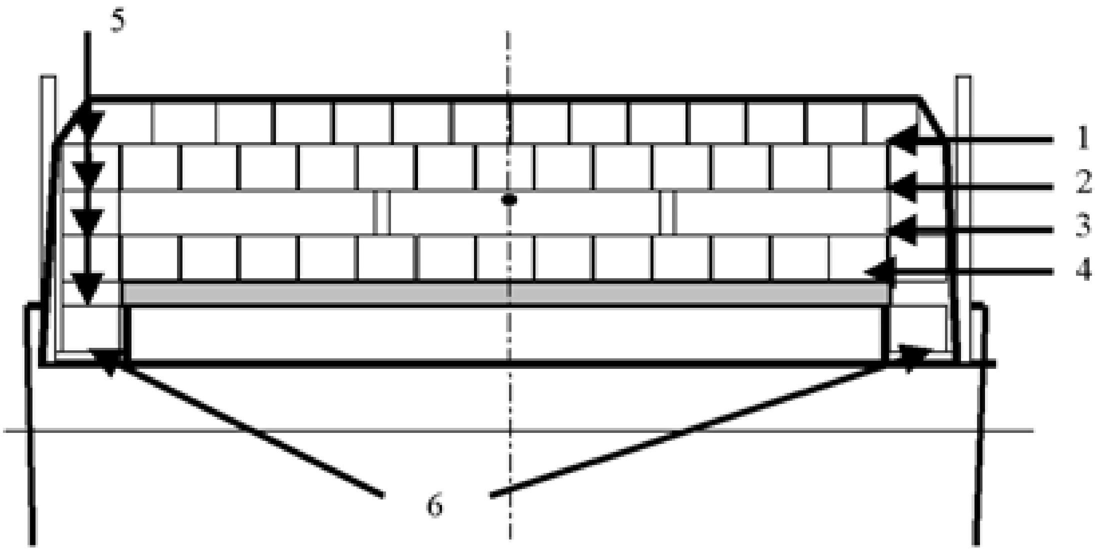


Figure 3. Various friction surfaces of the deck cargo of the FJORD PEARL

The stability of the stack of sawn timber packages

In the cargo securing manual the determination of the securing force is based on the static stability angle of the stack of sawn timber packages. A stack of sawn timber packages remains stable on a surface tilted to this angle. It is determined experimentally, without any lashings. The angle depends on the amount of the tiers, on the friction coefficient between the packages, on the way of the stacking (all packages longitudinally or every other tier across), on the dimensions and the friction coefficient of the planks, on the tension in the steel strings around the packages, etc. The cargo securing manual gives the static stability angle χ for a stack of three tiers of packaged timber (first and third tiers longitudinally, second across) equal to 17°. If this angle is translated directly to the friction coefficient one gets μ = 0.306. It is not known which static angle or friction coefficient was used in the actual cargo securing calculations. The data on the cargo did not include friction coefficients.

The friction coefficients

The investigation team carried out the securing calculations based on friction coefficients. Information was gathered from various sources for estimating the friction coefficients. No data was found for cases where ropes are placed between surfaces. A test series was commissioned from the Technical Research Centre of Finland (VTT) in order to complement the gathered data.

The ICHCA (International Cargo Handling CO-Ordination Association) guidelines give the values presented in table 1 for the friction coefficients. It is not clear from the table if the given values are static or dynamic friction coefficients. The brackets give the coefficients for use in trucks. The surface combinations are without ropes.

Table 1. Friction coefficients of the ICHCA guidelines

Material pair	**Dry, clean**	**Damp, unclean**	**Snowy, icy**
Wood-wood	0.30 (0.3-0.5)	0.30 (0.3-0.4)	0.20 (0.2-0.3)
Wood-metal	0.30 (0.3-0.5)	0.30 (0.3-0.4)	0.10 (0.1-0.2)
Wood-friction plywood	0.40	0.40	0.30

Reference 12, RINA 95 (Royal Institution of Naval Architects) gives the following friction coefficients, no ropes:

Table 2. Friction coefficients of Reference 12

Material pair	**Dry (dyn./stat.)**	**Damp (dyn./stat.)**
Rubber-steel	0.5/0.7	0.3/0.6
Wood-steel	0.4/0.6	0.25/0.4

The friction properties of the hoods of sawn timber have been studied with regard to work safety at the Finnish Institute of Occupational Health for the Ministry of Social Affairs and Health, on the step simulator, reference 11. The measurements were made for a hood covered with frost. The surface temperature of the hood was -8 ± 2°. The summary of the results is presented in table 3 below. The friction coefficients vary considerably for different hood types. The coefficients are kinetic friction coefficients.

Table 3. Results for frosted hoods, Finnish Institute of Occupational Health

Hood	**A**	**B**	**C**	**D**	**E**	**F**	**G**	**H**	**I**	**J**	**K**	**L**	**M**
μ	0.45	0.18	0.24	0.38	0.27	0.47	0.31	0.24	0.19	0.43	0.74	0.48	0.18

In addition the Finnish Institute of Occupational Health carried out measurements by replacing the shoe in the step simulator by a piece of 2" x 4" plank. The friction coefficients for it varied between 0.17-0.36 whereas the order was roughly the same. The test situation corresponded to a surface pressure of a 2500 kg package on the hood.

The most common hood types of these on the vessel were probably C, A and B.

As a final result of the measurements of the Finnish Institute of Occupational Health the requirements for the friction coefficients for the hoods were grouped as follows:

- Category 1, extreme grip, minimum 0.3
- Category 2, good grip, 0.20-0.29
- Category 3, uncertain grip, 0.15-0.19
- Category 4, slippery, 0.05-0.14
- Category 5, extremely slippery, below 0.05

Reference /Knott/ reports about friction measurements on top of hatch covers by tilting them. The results were:

- Wet wood on wet hatch, μ = 0.51
- Wet wood on dry hatch, μ = 0.645
- Average, including dry wood on wet hatch, μ = 0.584

The European Committee on Standardisation is drafting a cargo-securing standard, the draft of which is referenced as /Draft prEN 12195-1/. The standard draft gives the following static friction coefficients on sawn timber when the underlying surface is:

- Grained laminate or plywood, μ = 0.5
- Ribbed aluminium, μ = 0.4
- Painted rough steel, μ = 0.5

The figures are valid for dry, clean surfaces free of frost, snow and ice and they are based on minimum values of numerous measurements. In wet conditions the number of securings must be doubled. The dynamic friction coefficient is 0,7 x the static coefficient if no other value is given.

The Russian rules /18/ for cargo securing give the following friction coefficients:

- Wood-steel, μ = 0.3-0.6
- Wood-wood, μ = 0.45-0.65

The test series at the Technical Research Centre of Finland gave the average results presented in table 4, reference 7.

Table 4. Results of the VTT test series

Material pair	Dry	Damp	Thawing
Wood-Metal	0.53		
Wood-Metal, icy	0.32	0.15	0.10
Wood-Rope-Metal	0.35		
Wood-Dunnage-Metal	0.48		
Wood-Rope-Metal, icy	0.35	0.10	0.06
Wood-Rope-White hood, icy	0.40		
Wood-Rope-Clear Hood, icy	0.21		
Wood-Rope-Brown wood, icy	0.44		

Material combinations wood-rope-metal and wood-metal are the most important ones on board the vessel and they corresponded to the situation on the hatch covers. The combination wood-rope-hood was prevalent between the other tiers.

Some of the tests were conducted on an iced steel plate cooled to -22°C. The measuring was repeated as the plate thawed gradually at room temperature. Figure 4 shows the effect of the thawing. The surfaces were initially icy, the surface temperature being -2-4°C. In the end, the surfaces had completely thawed out. The friction coefficient diminishes radically as the thawing progresses. The coefficient begins to grow again when the ice has melted completely.

Consequently, the friction coefficient of a cold timber cargo may decrease drastically when the ship arrives in warmer areas.

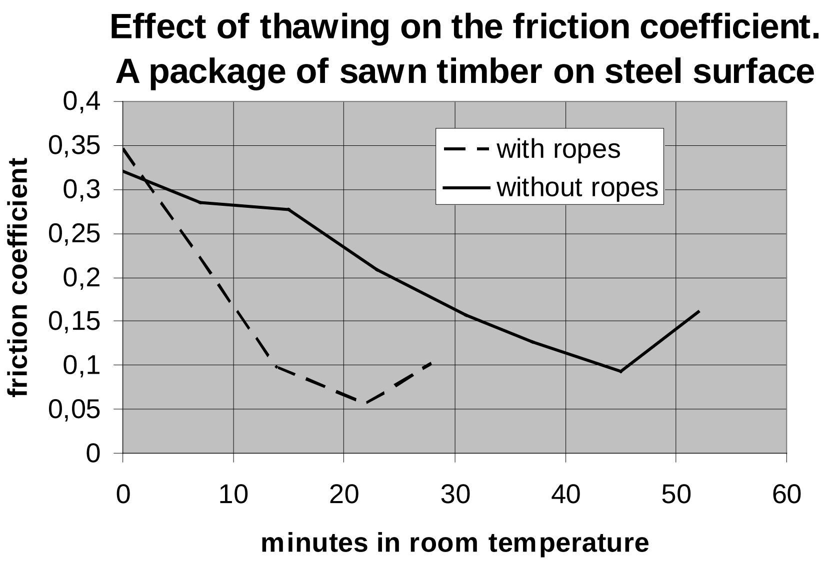


Figure 4. Effect of thawing on the friction coefficient

3 Weather conditions and friction coefficient

In the wind and the waves, the deck cargo of the vessel wetted partially although it was covered with tarpaulins. The cargo itself was cold, down to -20°C at the loading. During the voyage, the outermost sections of the deck cargo warmed to the air temperature, -4°C. The cargo was coldest in the holds. It is difficult to estimate the effect of the levelling of the temperature on the top surfaces of the holds and on the friction coefficient. It is probable that frosting occurred, which reduced the friction. The water temperature was about +1°C. There is no data of the humidity of the air. It is probable that the seawater that met the cargo froze at least in some places. Thus, it is possible that there was both water and ice at the same time between the packages. The tiers did not come into close contact with each other in all parts of the deck cargo because of the ropes in between. There were gaps that possibly let in water that froze. A combination of friction, lubrication and gluing ensued and may be looked at by using the average friction coefficient.

The friction surfaces consisted of various zones:

1. At the ropes, the material pair upwards was rope/wood and downward rope/hood or rope/steel.
2. There was no contact next to the ropes or ice had formed in between the surfaces.
3. Because of the bending of the timber packages material pairs wood/hood or wood/steel were created in large areas while the contact pressure varied.
4. The dimensions of the timber packages varied resulting in ridges, edges, grooves and dents in the contact surfaces of the packages.
5. Additional material pairs rope-rope and inside surface of hood-wood or outside surface of hood-wood were created. Since the structure of the hood consisted of layers, internal friction surfaces were created inside the hoods.
6. Snowy, icy and frosty areas had been created already during the loading. In addition, crushed ice was created during the voyage when the chunks of ice cracked. During the voyage, water and humidity increased the icing.
7. Frost accumulated on the surfaces when the temperature rose (-20°C ja -14°C $\Rightarrow$ -4°C).

4 Critical friction coefficient/listing angle combinations

Figure 5 presents the calculated critical friction coefficient/listing angle combinations. They are based on the equations in enclosure 2 supplemented by the dynamic factors as in enclosure 5. The picture shows additionally the tipping angle when there is no tension in wires.

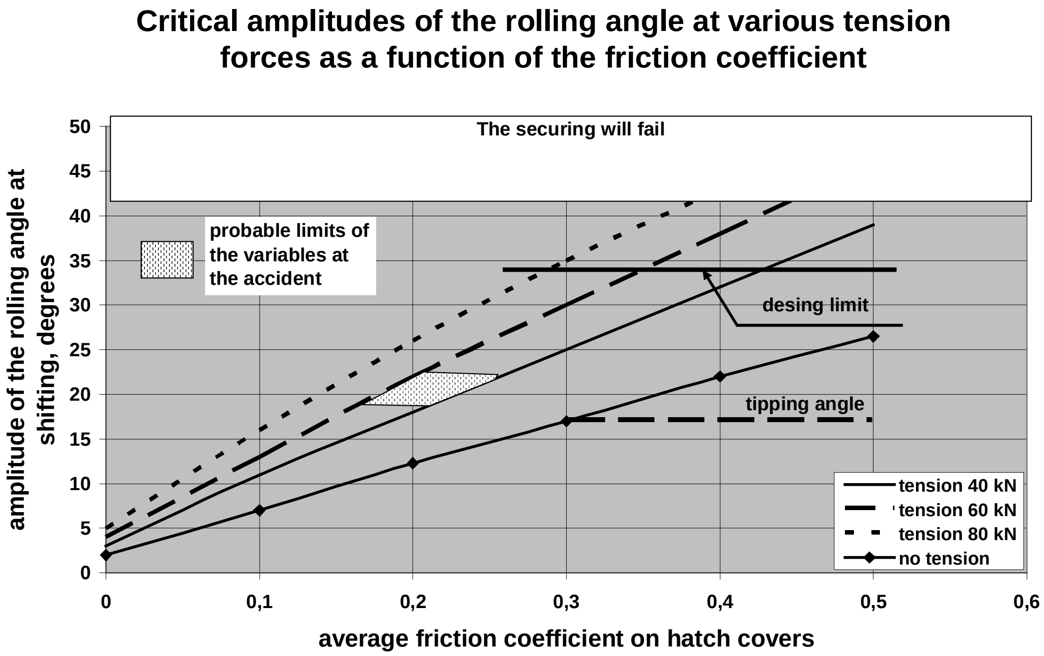


Figure 5. Critical friction coefficient/rolling angle combinations

The deck cargo remains stable below the curves. There were 52 securing wires including two thirds of the diagonal wires. The tension is the vertical force created in the wire when the tightening is done on top of the cargo. The vertical force in the wire at both sides is smaller due to the wire friction against the corners of the cargo. The differences level out as the cargo packs. The calculations assume that the friction coefficient for the packages furthest to the side is the friction coefficient on the hatch covers +0.15. Thus, if the friction coefficient on the hatch covers is zero, a small rolling angle is yet allowed. When the wire has no tension, it is not slack, but begins to resist the shift if the heel angle exceeds the stability angle of the stack.

The securing calculations suppose that the bilge keels are lost. The corresponding maximum expected heel angle was 34°. Actually, the ship had bilge keels at the accident. The expected heel angle reduces to about 32°.

A probable combination listing angle-friction coefficient-tension at which the cargo shift may have occurred, based on the stability calculations, is marked in figure 5. The limit where the tension in the wire becomes too great snapping it is also marked in the scheme. This limit is approximate and depends on the condition of the wires and the other securing equipment. It is probable that due to local variations of thawing, the average friction coefficient has been more than 0.15. It can also be assumed that it has not been more than 0.25.

It can be seen that the tension of the securing wires has a strong effect on the maximum allowable rolling amplitude. Figure 6 shows the effect of tension in more detail as calculated by the investigator team. For example, if the friction coefficient is 0.2 the cargo will shift at an angle of 12° with no tension and at an angle of 26° when the tension wire is very taut.

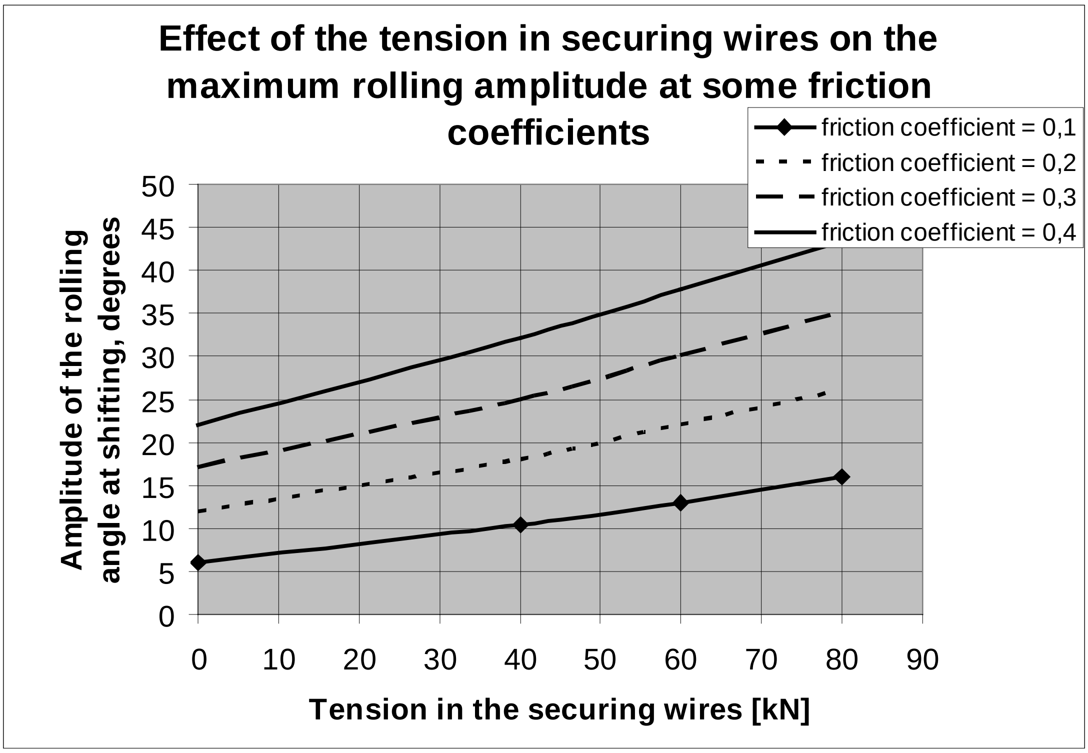


Figure 6. Effect of tension in the securing wires on the critical listing angle at various friction coefficients on the hatch covers

The securing wires were tightened at about 12 hour intervals. It can be assumed that the tension at the tightening rose near the uppermost limit curve in figure 5. Probably, the 40 kN curve was approached before the next tightening. The rougher the sea, the quicker the wires slacken.

Consequently, the real tension should be known, at least when tightening the wires. The ship should have necessary equipment to measure the tension.

Figure 7 shows a principal possible scenario of the development of the forces during the voyage of FJORD PEARL. Actual curves contain peaks. The abrupt changes in the holding forces show the tightening moments.

These results do not represent the actual design values since they do not strive to include the safety margins. The results are only referential due to the unclear basic data.

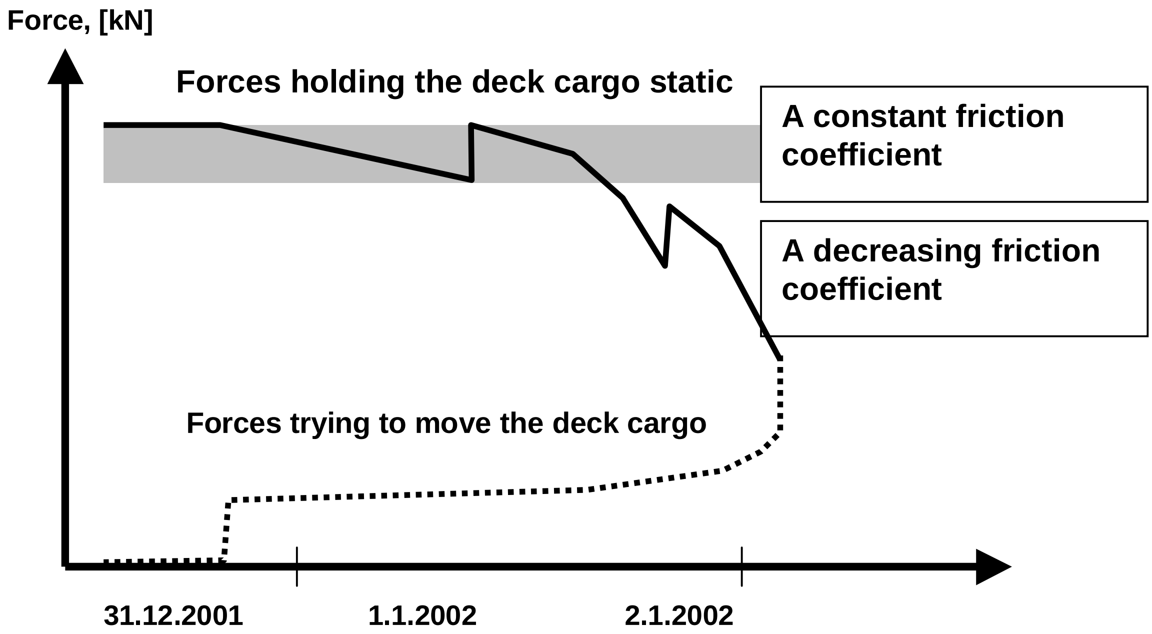


Figure 7. A possible scenario of the development of forces during the voyage of FJORD PEARL

Enclosure 5. Forces affecting the deck cargo

M/S FJORD PEARL, forces affecting the deck cargo

In addition to the static forces caused by a constant list, the cargo of the vessel is affected by forces created by the motions of the vessel in rough seas.

The waves create six motion components. Translational components are: longitudinal surge, 1, lateral sway, 2 and vertical heave, 3. Rotational components are: roll about the longitudinal axis, 4, pitch about the transversal axis, 5 and yaw about the vertical axis, 6.

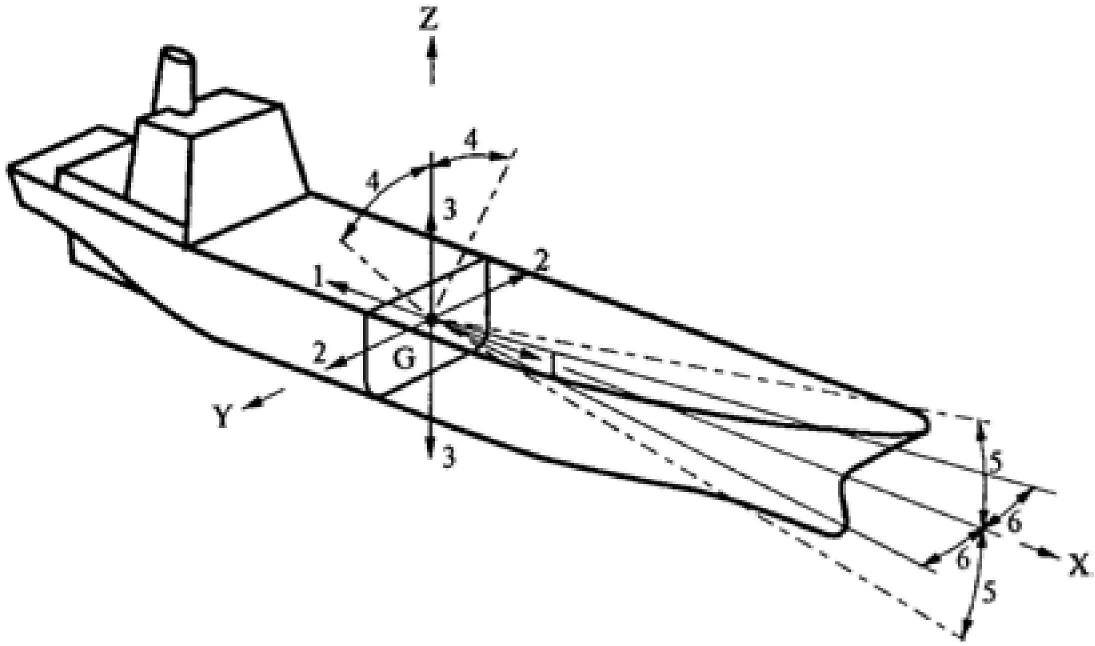


Figure 1. Motions of the vessel in rough seas

In the following, the effects of the most significant motions, rolling and heaving, are calculated whereas the combined effects of pitching, swaying and yawing on rolling are only described. Changes in the list and accelerations are created because of these motions resulting in forces affecting the cargo. The effect of the translational motions is the same throughout the ship, whereas the effect of the rotational motions increases when moving further away from the axis of rotation. The cargo will start to shift when these forces exceed the friction and tension forces holding the cargo static. The longitudinal components of these forces are clearly smaller than the transversal components, so the former ones are left outside the study in the following.

Forces are evaluated in a simplistic form, without the use of extensive software for the calculation of ship motions. This was considered adequate for the case in hand, since the motions of the vessel during the accident are known in sufficient detail for estimation of the forces in the accident situation.

The cargo-securing manual recommends that the deck cargo should go over board if the vessel heels more than 40°. Consequently, the deck cargo should be secured adequately but not too tightly.

Forces affecting the deck cargo

The following forces (accelerations) affect the deck cargo in rough seas:

- Acceleration due to gravitation and its components during the rolling parallel to the deck and perpendicular to it
- The most significant effects caused by the waves are
  - The tangential and radial acceleration due to rolling and their components parallel to the deck and perpendicular to it. The stability properties of the vessel have a great effect on her rolling
  - Vertical acceleration due to pitching and heaving
  - Lateral acceleration due to yawing and swaying
- Friction forces
- Forces of the securing equipment
- Wind forces
- Wave forces
- Buoyancy affecting the submerged deck cargo
- Accelerations due to manoeuvring
- In addition, the inertial force due to the stopping of the shifted deck cargo

These forces can be divided in the following way with regard to the cargo shift:

1. Forces holding the cargo static

- μ mgcosφ, the friction force parallel to the surface under the cargo due to its mass, where
  - μ = Friction coefficient between the cargo and the hatch cover or between the tiers of the cargo
  - m = Mass of the part of cargo under review
  - g = Gravitational acceleration
  - φ = Heeling angle
- Force F_k affecting the securing wire and the resulting friction force μF_k
  - In this case F_k is nearly perpendicular against the deck, so the cosine-correction is ignored. In addition, the strap effect at the bends of the wire must be considered since the wires are stretched across the cargo
- Supporting forces of possible supports

2. Forces striving to shift the cargo

- mgsinφ, is the static force caused by the list parallel to the underlying surface
- Dynamic, lateral tangential force due to the rolling
- Dynamic, centrifugal force due to the turning*
- Force created by the wind against the side of the cargo
- Wave pressure against the side of the cargo

3. Heeling forces and moments

- The inertial force and its moment due to the stopping of the shifted cargo
- Moment caused by the shifted centre of gravity of the cargo
- Moment caused by the wind pressure*
- Moment caused by the wave pressure*
- Moment of the water leaking into the tanks and moments of the pumped water

4. Additional forces affecting the situation

- Friction coefficients between the various surfaces of the deck cargo and between the wire and the cargo; including the effects of snow, ice, freezing temperatures and other materials between the surfaces
- Vertical accelerations caused mainly by the rolling and the heaving (increasing or decreasing the effect of gravity)
- Buoyancy affecting the cargo under water
- Slamming*
- Shape of the hull of the vessel, appendages such as bilge keels*

The most significant forces are handled in more detail later. The forces marked by an asterisk are explained shortly in the following:

a) Centrifugal force caused by the turning $F = mv^2/r$, where v is the speed and r is the turning radius. The speed of the vessel was 4 m/s. The turning radius in the prevailing circumstances is assumed clearly larger than that of a tight turn; it is estimated at 10 times ship length, in other words approximately 1200 m. The resulting heeling force is about 140 kN. The height of the centre of gravity of the vessel is 6.76 m. The counteracting underwater force is estimated to have effect at a height of 4 metres. The resulting heeling moment is now $2.76 \cdot 140$ = 386 kNm, which corresponds to a heel of less than one degree in calm water according to the stability curves of the vessel. The heel is to port.

   The corresponding force acting the cargo is about 17 kN, which was counteracting the shift. This force is small compared to the other significant forces.

   These forces can be left unaccounted because they are small, but also because the turning of the vessel in the waves did not proceed along an arc. In addition, it is uncertain whether these forces were effective when the cargo shifted.

b) The wind area of the vessel was 1044 m^2 in the stability calculations and the centre of gravity of this area lie at 5.2 m above the water level. The wind speed was 27 m/s in the gusts. The heeling force of the wind may be obtained by the formula $½CA\rho v^2$. C is resistance coefficient that depends on the position of the vessel in relation to the wind and is 1.2 for a side wind, shown in figure 2; A is area; ρ is air density. The result is $0.5 \cdot 1.2 \cdot 1044 \cdot 1.225 \cdot 27^2$ = 559 kN and the heeling moment is 2908 kNm. The corresponding heel is about 5°. A steady wind speed of 18 m/s gives a heeling moment of 1292 kNm and a heel of about 2°. The wind has an effect in the opposite direction from the steering in the accident conditions. The toss happened when the vessel was turning from beam wind to leading wind.

c) During the turn, the main direction of the waves and wind was the same. The waves reached the deck cargo at varying heights because of the rolling. According to the reports of the ship's officers, the vessel listed first 15° to port during the turn, which put the deck cargo closest to the gunwale in the water. The effect of this is difficult to estimate but it is possible that water got in between the packages furthest to the side. Therefore, the friction coefficient might have diminished locally.

   In the design conditions, the rolling angle of the vessel is 24° according to the cargo-securing manual. The effect of the bilge keels is not included in these calculations. In reality, the bilge keels reduce the rolling by about 4°, giving an angle of 20°. The reported roll to 15° was against the wind. The wind maintained a list of about 2-5°. Therefore, the vessel heeled effectively 17-20° because of the waves.

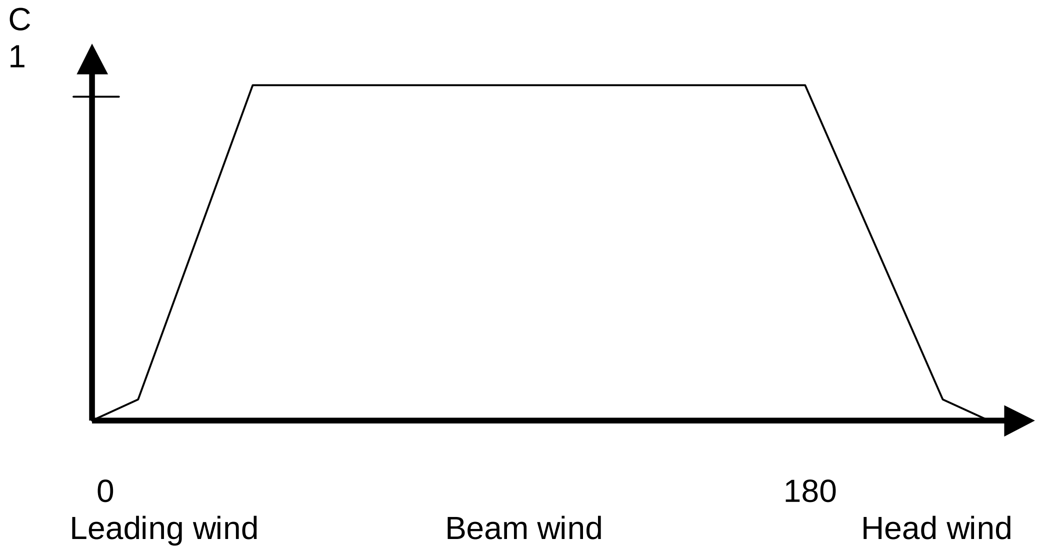


Figure 2. Relation of the wind resistance coefficient C on the position of the vessel with regard to the wind

The wind area of the deck cargo was about 200 m^2, and the shifting force caused by the wind was $0.5 \cdot 200 \cdot 1.225 \cdot 1.2(18-27)^2 \cos\varphi$ = (48-107)cosφ kN.

The accident conditions were milder than the design conditions. The wind speed was 18-27 m/s, whereas the design conditions correspond to a wind speed of about 43 m/s. The significant wave height was approximately 4 m, whereas the theoretical wave height in the design conditions was 11 m. The probability for exceeding this wave height is 3%. The corresponding significant wave height is 7-7.5 m. It is obvious, that the roll to 15° was caused by a disadvantageous position of the vessel with regard to the waves during the turn.

The circular frequency ω of the prevailing wave is the square root of ($2\pi g$ / wave length) = 0.72 1/s and the period is thus 8.7 s. The roll period was about 23 s according to the cargo-securing manual, and corresponding circular frequency about 0.27 1/s. Thus, the circular frequency of the waves was clearly larger than the circular frequency of the rolling of the vessel. The waves no longer had energy at the specific circular frequency of the vessel. Therefore, no resonance was created during the turn.

On the other hand, the waves met by the vessel varied statistically. The turning of the vessel lasted a few minutes and the vessel met 7-8 waves per minute, so it is probable that there were waves more disadvantageous than the average as well.

d) Slamming was not reported but heavy pitching was noted instead. The pitching was caused by the fact that the significant wavelength was close to the ship length. There were cross-waves in the area so that the ship encountered head waves before the turn, too. Thus, the lashings may have slackened more quickly.

e) The shape of the hull is normal. The ice class of the vessel is УЛ, which means that the bilge keels must not be included in the design calculations of stability. In reality, the bilge keel dampened the rolling in the accident situation. The effect of this can be estimated by the method in the cargo-securing manual. The effect of this reduces the rolling angle of 24° by 4-5°. Correspondingly, the maximum dynamic heeling angle reduces from 34 to about 32°.

Significant forces

Gravity, static

The following force components have an effect on the deck cargo with mass m when the vessel is listed:

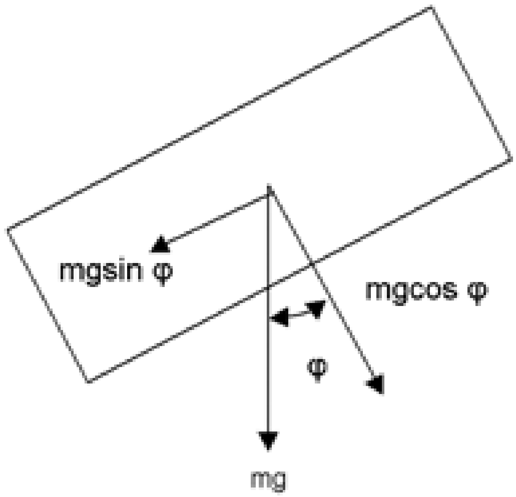


Figure 3. Static list and force components

The listing angle = φ. The table below presents the coefficients for multiplying the mass (kg) for obtaining force (N). The sine component strives to shift the cargo and the cosine component multiplied by the friction coefficient strives to hold the cargo steady.

Table 1. Static force components as coefficients

angle	gsin φ	gcos φ
5	0.85	9.77
10	1.70	9.66
15	2.54	9.48
20	3.36	9.22
25	4.15	8.89
30	4.91	8.50
35	5.63	8.04
40	6.31	7.51

Dynamic forces due to the waves

Forces due to rolling

Rolling takes place about the rolling axis. The approximate location of the axis is half way between the vertical centre of gravity of the vessel and the centre of gravity of the displacement. The rolling axis is often placed at the centre of gravity of the vessel or on the waterline in the centerline. The location of the centre of gravity of the vessel is known from the stability calculations. The investigation team does not know the centre of gravity of the displacement, but the estimated height at the timber load line of 7.3 m is 4.4 m. Thus, the rolling axis is at (4.4 + 6.8)/2 = 5.6 m from the base line.

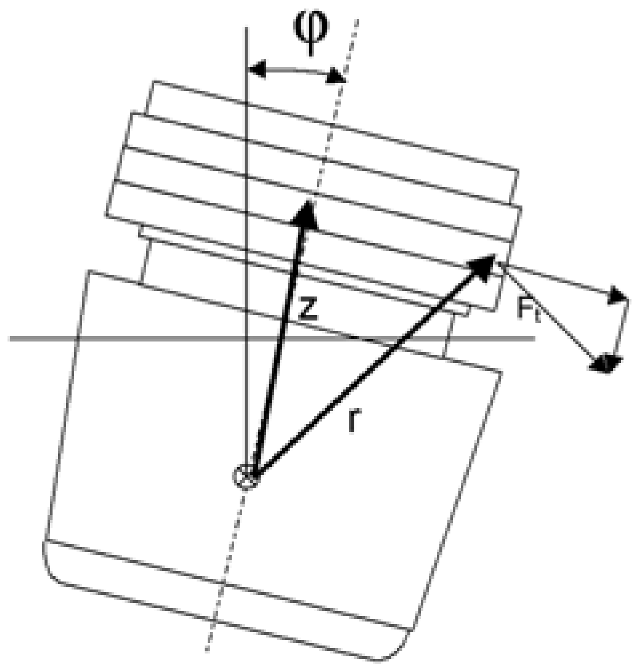


Figure 4. *Rolling of a vessel with deck cargo*

The equation of rolling is $\varphi = \phi_A \sin \omega$ t, where

φ = Heeling angle
ϕ_A = Maximum rolling angle, rolling amplitude
ω = Circular frequency = 2π/T
T = Rolling period

Rolling creates a tangential force in the direction of the tangent of the path of the cargo, and a centrifugal force in the direction of the rolling radius, a radial force:

Tangential force:

$$F_t = \frac{mr4\pi^2\phi_A}{T^2}$$

where

m = Mass of the body in question
r = Distance of the centre of gravity of the body in question from the rolling axis

The tangential force reaches its maximum when the vessel is in the extreme position, in other words when $\varphi = \phi_A$ and when the section of cargo is furthest from the rolling axis. The tangential force has two components: the component in the direction of the deck and the perpendicular component directly against the deck. See figure 4.

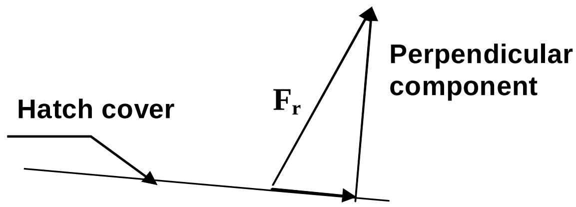
The component parallel to the deck is obtained when r in the above formula is replaced by its vertical projection z, which in this case is constant in each package tier. This component strives to shift the cargo.

The component perpendicular to the deck depends on the distance, parallel to the deck, from the centreline to the place of the cargo in question and is largest at the furthermost edge of the cargo, being zero amidships. So, it does not depend on the vertical distance from the roll axis. This component increases the weight of the cargo against the deck.

Radial (centrifugal) force:

$$F_r = mr\left(\frac{d\phi}{dt}\right)^2$$

where $\frac{d\phi}{dt}$ = angular velocity = $\phi_A(2\pi\cos\omega t)/T$



The centrifugal force is zero, when the vessel is in the extreme position and reaches its maximum at the middle position. At other angles, it has a component parallel to the deck and a perpendicular component directly against the deck. The minus sign signifies that the force directly against the deck has an effect in the opposite direction of gravitation. The situation is the worst with regard to a cargo shift when the vessel is in the extreme position.

The following table presents components of these forces. The figures mark the coefficient by which the mass must be multiplied to obtain the force. Comparing table 1 to tables 2 and 3 sees the relative magnitude of the dynamic and static forces. Table 2 presents the coefficients for the various tiers of cargo at a rolling amplitude of 30° in the extreme position and at a 0° angle. Table 3 presents the coefficients for the first tier at rolling amplitudes of 10°, 20°, 30° and 40° in the extreme position and at a 0° angle.

Table 2. Force components as coefficients for various tiers of cargo at rolling amplitude of 30°

		All angles	Extreme position Angle =30°		Middle position Angle = 0°		
Tier	z [m]	Tangential force, component parallel to deck	Tangential force, perpendicular component	Radial force	Tangential force, perpendicular component	Radial force, component parallel to deck	Radial force perpendicular component
1	5.5	0.21	0.30	0	0	0	-0.11
2	6.6	0.26	0.30	0	0	0	-0.14
3	7.7	0.30	0.30	0	0	0	-0.16
4	8.8	0.34	0.30	0	0	0	-0.18

Table 3. Force components as coefficients for first tier of cargo at various rolling amplitudes

	All angles	Extreme position = Roll angle		Middle position Angle = 0°		
Roll angle	Tangential force, component parallel to deck	Tangential force, perpendicular component	Radial force	Tangential force, perpendicular component	Radial force, component parallel to deck	Radial force perpendicular component
10	0.07	0.10	0	0	0	-0.01
20	0.14	0.20	0	0	0	-0.05
30	0.21	0.30	0	0	0	-0.11
40	0.29	0.41	0	0	0	-0.20

Heave motion causes a vertical force

$$F_h = \frac{m4\pi^2 h}{T^2}$$

where

h = wave amplitude
T = wave period

According to the data from the Finnish Institute of Marine Research, let h = 1.8 m and T = 9 s, giving a vertical force approximately 0.9 times mass. This corresponds to beam waves of pure sinusoidal shape. The real situation is different from this and is estimated that 0.6 times mass is suitable. The perpendicular component of this force against the deck alternately increases and reduces the weight of the cargo. The force component parallel to the deck increases or reduces the force shifting the cargo, for its turn. It can have a simultaneous effect with the forces due to the rolling. In this case, the reduction of the weight is disadvantageous, since the friction force also diminishes correspondingly.

Combination of the forces

Comparing table 1 to tables 2 and 3 can evaluate the combined total effect of these forces. The component of the tangential force parallel to the deck increases the shifting force, in the first tier on the hatch covers by 4-5% at angles of 10-40°. The component of the heave acceleration parallel to the deck reduces the shifting force by ($\sin\varphi$)6%, leading to a total increasing of 0%, 3.5%, 3%, 2% and 1.5% at heeling angles of 0°, 10°, 20°, 30° and 40°.

The perpendicular component of the tangential force increases the weight of the cargo against the deck by 5,5%, 3,5%, 2% and 1% at heeling angles of 40°, 30°, 20° and 10°. Heaving lightens the weight by ($\cos\varphi$)6%, leading to a total lightening of 0%, 2%, 4%, 5% and 6% at heeling angles of 40°, 30°, 20°, 10° and 0°.

Consequently, the investigation team has used the results in the table 4 in calculating the limiting rolling amplitude-friction coefficient-wire tension combinations in enclosure 4.

Table 4. Increase/decrease of the dynamic forces on the static forces

Rolling amplitude	**Perpendicular change [%]**	**Change parallel to deck [%]**
0	-6	0
10	-5	+3.5
20	-3.5	+3
30	-1.5	+2
40	+1.0	+1.5

The total effect of the dynamic forces is slight and can be even considered to be within the accuracy of the calculations. This effect is often left out entirely. The effect is considered in the cargo-securing manual by decreasing the static stability angle of the stack of the sawn timber packages. For FJORD PEARL the angle decreased from 17° to 16.5°.

Enclosure 6. Finnish Maritime Authority, statement

Merenkulkulaitos
Keskushallinto

15.9.2003 Dnro 1311/335/2003

Onnettomuustutkintakeskus
Sörnäisten rantatie 33 C
00580 HELSINKI

SAAPUNUT
16. 09. 2003
317/5M

M/S FJORD PEARL,
LASTIN SIIRTYMINEN JA VAARATILANNE 2.1.2002

Olemme tutustuneet asiantuntijoidemme kanssa tutkintaraporttiin ja lausumme seuraavaa:

Nostoköydet ehkä litistyivät lastin alla. Keinokuituköysi on liukasta erityisesti metallia vasten. Välipuita ilmeisesti ei käytetty.

Kiinnitys tapahtui pelkästään vetämällä vaijerit lastin ylitse. Mitään pakettityyppistä kiinnitystä ei tapahtunut.

Tukitolppia olisi tullut hyödyntää kiinnityksessä. Niistä olisi saatu välikiinnikkeiden kanssa tanakka kokonaisuus. Välikiinnikkeitä ei ilmeisesti käytetty lainkaan.

Lastia ei myöskään kiinnityksen jälkeen kiilattu ylhäältä millään tavalla.

Puutavarapakettien korkeus vaihteli. Korkeusvaihtelu ei kuitenkaan ollut kovin suuri. Välipuita ei ilmeisesti taaskaan käytetty.

Puutavarapaketit muovitetaan sateen ym. varalta normaalisti päältä, päistä ja sivuilta. Alapuolelta suojaaminen ei ole tavallista. Tällöin tulee muovi vasten puuta ja välipuita tulisi käyttää.

Lastia ei menetetty. Tämä osoittaa, että kiinnikkeiden lujuus riitti, mutta ne eivät olleet kireällä eivätkä kulkeneet lasti ympäri.

Mielestämme IMO:n Code of Safe Practice for Ships Carrying Timber Deck Cargoes noudatettuna antaisi tähänkin tapaukseen ohjeet. Merimiestaitoakin voitaisiin käyttää hyödyksi.

<u>Suositukset</u>:

1. Kitkatiedot auttavat aina aluksen päällikköä. Kannatamme.

2. Lastinkiinnityskäsikirja näyttäisi tässä tapauksessa kaipaavan tarkennuksia. Myös laskentaesimerkki on hyödyksi.

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Porkkalankatu 5
00180 Helsinki

Postiosoite
PL 171
00181 Helsinki

Puhelin
0204 481

Faksi
0204 48 4355

NORDEA 166030-107626
OKO 500001-20377634
SAMPO 800015-38014

3. Kansilastit saadaan pysymään paikoillaan erilaisilla lastaustavoilla.
Oikeanlainen kiinnitys on kaiken perusta.
Lippuhallinnon on syytä harkita parannuksia käsikirjaan.

3. Tällainen tieto voitaisiin nähdäksemme toimittaa IMO:n välityksellä jäsenhallinnoille.

Toimistopäällikkö Pekka Korhonen

Merenkuluntarkastaja Marko Rahikainen

Enclosure 7. Central Marine Research & Design Institute (CNIIMF), statement

ЗАО «Центральный научно-исследовательский и проектно-конструкторский институт морского флота» (ЦНИИМФ)

JSC Central Marine Research & Design Institute (CNIIMF)

Лаборатория крепления грузов **Cargo Securing Laboratory**

FAXIMILE MASSAGE

SAAPUNUT 29.09.2003 356/5m

16.09.2003 г

Our. Ref. No. OBC-1/31-03

TO:	**Finnish Accident Investigation Board**
ATT:	**Pertti Silvonen, accident investigator maritime accidents**
Fax:	**+358 400 979 113**
	+358 916 067 811

No. of pages, including first 5

Ref. No. 231/5 M on 2 July 2003

Dear Sirs!

M/S FJORD PEARL, SHIFTING OF CARGO ON 2 JANUARY 2002

We thank you for the continuation of our cooperation in the matter of investigation of conditions of safe carriage of timber cargoes.

We with great satisfaction perceived completeness, objectivity and thoroughness of the investigation of circumstances, connected with shifting of the deck saw timber cargo on m/s FJORD PEARL on 2 January 2002, reflected in the presented Investigation Report C 1/2002 M of the Finnish Accident Investigation Board.

Special satisfaction was evoked by the investigator's attention to influence of friction effect upon deck packaged sawn timber cargo's shiftability.

For us as elaborators of the ship's Cargo Securing Manual, the most significant was the establishment of the fact, that the cargo had been stowed and secured in accordance with its requirements.

We also consider important to note, that in spite of the fact of shifting of the whole cargo for about 1.5 m to starboard, none of the timber lashings broke, the cargo hung by the lashings, making a dangerous least, bounding the ship's ability to maneuver and enter the port of refuge. In a similar situation excessive lashing's strength led m/s Kodima into shallow water with loss of practically the whole cargo and substantial damages of the ship but all the wires, securing the deck cargo were intact.

Then the question arises: "Why did, nevertheless, shifting of the cargo take place?"

In the course of preparing the present comments our own investigation on the friction coefficients of a large number of possible combinations of friction pairs, formed inside a multi-tier (from 2 up to 6 tiers) stack between packages of sawn timber, covered by different types of protective hood only on five sides, except for the bottom side, and stowed in a stack together with the thrown upon them during loading polypropylene lifting ropes, was carried out.

191015, Санкт-Петербург, ул. Кавалергардская, 6.
телефон: (812) 275-8965
тел./факс: (812) 327-2638

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ISO 9001 Quality system certified by Russian Register

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ЗАО «Центральный научно-исследовательский и проектно-конструкторский институт морского флота» (ЦНИИМФ)

JSC Central Marine Research & Design Institute (CNIIMF)

Лаборатория крепления грузов

Cargo Securing Laboratory

FAXIMILE MASSAGE

The results of our investigations, individual fragments of which are presented on the attached photos, showed a wide range of values of the friction angles of different contacting pairs inside a multi-tier stack.

The lowest angles of slope of sliding surfaces at the moment of initial shifting amounted 12° – 14°(photo 1). which corresponds with friction coefficients of about 0.21 - 0.25.

Prevailing angles of slope amounted 20° – 24°, and maximum ran up to 39.4° (photo 2).

Steadily low values were shown by the friction angles of the polypropylene lifting ropes over almost all of the kinds of the used surfaces (photo 3).

The friction tests performed at the Technical Research Centre of Finland on the effect of warming on the friction coefficients were of a great interest for us.

However, it is necessary to admit, that a carrier has no possibility to determine any of the values specified in the Investigation Report and above in this paper.

The Shipper shall provide the Master with such information prior to loading taking into account the season and conditions of an intended carriage.

Accumulation of such information will allow in future to elaborate valid calculation methods of steel uprights' strength, application of which with number of tiers more than 3, probably would solve the problem of taking into account the need of reduction of the friction coefficients as a result of an undeclared by the shipper change of conditions of producing of sawn timber packages, presented for carriage on a ship.

Such work could be carried out by joint efforts of Finland and Russian investigators on demand of Finland and Russian shippers under the supervision of both Finland and Russian Maritime Administrations with subsequent presentation of the results to the IMO's attention.

Look forward for continuation of our cooperation.

Yours sincerely,

Evgeniy Karpovich
the Head of Cargo Securing Laboratory, CNIIMF

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Photo 1


Photo 2

Photo 3

Enclosure 8. Malta Maritime Authority, Merchant Shipping Directorate, statement

The prefix 21 has been added to all the Directorate Telephone and Fax lines

Malta Maritime Authority
Merchant Shipping Directorate
Maritime House, Lascaris Wharf
Valletta VLT01 MALTA

FAX TRANSMISSION

N.B. If you are not the named recipient of this Fax, please notify us, preferably by telephone or E-mail, forthwith, and do not read, copy or disclose this Fax to any person.

Date: 22 August 2003

To: **MR. PERTTI SIIVONEN**

Organisation: **ONNETTOMUUSTUTKINTAKESKUS**
SORNAISTEN RANTATIE 33 C
FIN-00580 HEKSINKI
FINLAND

Fax: 00358918257811
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From: Anthony Zerafa
Technical Dept.

Phone: +356 21250360
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Telex: 1362 REGSHP MW

Number of pages including this one: two

Subject: ***MV FJORD PEARL IMO #7524354*** ***COMMENTS ON DRAFT REPORT C 1/2002 M.***

Regarding the above-mentioned subject;

We have read and agree with the draft report.

There are only minor comments from our side to some details.

Included with this fax message, please find enclosed a copy of *page 2* of the draft report, with the corrections for your perusal.

These are;

Home port Valletta, *(not La Valletta, Malta.)*
Displacement max 10,000 mt, *(not 10720)*
DWT, max 6070 mt, *(not 6780)*

Thanking you for your cooperation.

Regards

Anthony Zerafa

Anthony Zerafa