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1990-02-05

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REPORT
regarding assistance to The Electrical Research Institute of Mexico, Cuernavaca, Mexico, related to the design and manufacture of hydraulic mini-turbines and made under UNIDO Special Service Agreement No. CLT 89/470

By Evald Holmén *) [illegible]

The main part of the service was undertaken from January 15th to 26th 1990, and made during a visit to Cuernavaca, Mexico. Some problems were left open during the visit and were later communicated to the Institute.

Notes about considerations and recommendations made during the visit are enclosed as <u>Appendix 1</u>.

Later communications are enclosed as <u>Appendix 2</u>.

The mini-turbine project at the Institute is handled by a working group of two full-time and 3-4 part-time working persons. The coordinator of the group is Flavio Ferran. The group is together with for instance groups for solar and wind energy working under Non Conventional Energy Resources Department headed by Ana Maria Martinez. The department in turn belongs to Energy Resources Division under Dr Pablo Mulas del Pozo. Guillermo Fernandez de la Garza is director for all The Electrical Research Institute of Mexico.

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The working group has been engaged with the design and manufacture of a hydraulic mini-turbine unit for about two years. The status of the work now was that the greater part of the design and the necessary calculations were made. Some of the main parts of the unit as generator, speed-increaser and governor were purchased and delivered to the Institute. Some other parts as shaft, flywheel and frame structures were manufactured in a workshop belonging to the Institute.

The design, the calculations and the manufactured parts as well as the plans for the future were, as can be seen in Appendix 1, scrutinized down to the last detail. It was necessary to go that deep, as the turbine technology is partly empirical, where experience from previous designs plays a dominant part, and partly a domain where theories from many fields are put into practice.

Taking into consideration that the group started the engineering work from a zero position of specific knowledge in the field of water turbines, the development is considerable. A big part of the design work has followed a manual in turbine design made up by Olade, giving a robust and for the requirement of today too conservative design. The requirement specification and the goal for the project has changed since the start, which will be explained later on.

Going out from the requirement of today, many of the details could have been given a cheaper design. If a generator with a lower speed had been chosen, there had been no need for the speed-increaser. On the other hand, using the requirement from the start of the project, that lower speed should have required a 9 times as heavy flywheel (3 tons instead of 350 kg).

At the start of the project, two years ago, the main plan was to electrify an irrigation distric situated around 100 km from the nearest point of the main electricity grid. The intention was to use 16 mini-turbines for building an isolated grid system containing about 2 MW. The power should be used for lighting a village and for pumping water in the irrigation system. Only 10% of the planned irrigation system was used, partly due to lack of energy for pumping. On that basis the design work was started using the technical manuals from Olade for guidance. The manuals from Olade are mainly prepared for making turbines suitable for frequency regulation on isolated grid systems.

After a certain time the working group felt that they met with a resistance from some authorities and especially from CFE - The State Electricity Board. A 90 km long transmission line was built from a substation of the main grid to the village in question.

The project then had to be limited and restricted to the design and building of one mini-turbine, which should be connected to the transmission line and the main grid. There was no need inside the new requirement for the governor, already purchased from Brazil, capable of frequency regulation of an isolated grid, nor for the big flywheel making the governing of the frequency of the grid possible.

The synchronous generator planned was exchanged for a simple induction generator suitable only for hanging on a big grid system. The rather conservative turbine design and the chosen lower specific speed necessary for good operation of an isolated grid was, however, kept also after the limiting of the project requirement, together with the governor and the flywheel. That in order to gain knowledge from this type of units, if in the future a new irrigation district can be found, where a group of turbines can be installed for working isolated from the main grid.

There was also another good thing kept - the idea of making model tests for optimizing the final design of the prototype runner. This will also make it possible to design and test runners with higher specific speed in the same test rig. Such runners could then be used when designing considerably cheaper turbine units for connection to the main grid. The existing design is too exclusive for connection to a big grid system.

Personal Comments

My main impression from the visit is that the group has made a good job. The circumstances have not always been the best. Especially the change of the goal for the project has disturbed the development process. There are decisions made and steps taken early in the process that cannot easily be changed to fit new goals.

The support to Flavio Ferran from the management and direction of the Institute is good.

A secondary but as important outcome from this mini-turbine development process is the education of the personnel involved. The working group will, when the mini-turbine is installed and in operation, have gained a considerable knowledge about water turbines, in many parts, I presume, a knowledge outstanding in the open market in Mexico. That, because the turbine technology is partly, as has been said before, an empirical science and thus only reachable by trial and error processes. The model tests made will contribute substantially to this type of knowledge.

The knowledge gained can then be used for big turbines as well as small. I have my knowledge from big turbines, 10 to 450 MW units, but I have no problems using it for mini-turbines. The opposite process is almost as easy. The same model tests results can from a technical point of view be used for small as well as big turbines.

I have agreed to assist the group by telefax communications when special problems occur until the unit is commissioned and put into operation. The group has agreed to send the results of the model tests to me. I am a collector of technical knowledge in the field of water turbines, and these results will help me in making an evaluation of the manuals from Olade, if required.

The best time for additional assistance to the Institute is at the time of commissioning of the unit. I think such a visit should be useful, if the expenses can be covered.

Stockholm, 1990-02-05

Evald Holmén

IND/014
HLE/GON

Flavio Luis Evald Raul

Appendix 1

RECOMMENDATIONS FROM ENG. EVALD HOLMEN (EXPERT FROM SWECO COMPANY/SWEDEN STOCKHOLM, SPONSORED BY UNIDO), FORMULATED DURING HIS TECHNICAL-VISIT TO MEXICO (15TH TO 26TH JANUARY OF 1990) RELATED TO THE DESIGN AND MANUFACTURE OF A HYDRAULIC MINI-TURBINE THAT IS BEING UNDERTAKEN BY THE ELECTRICAL RESEARCH INSTITUTE OF MEXICO.

CONSIDERATIONS, CALCULATIONS AND DESIGNS MADE BOTH MANUALLY AND VIA COMPUTER (AUTOCAD) WERE SCRUTINIZED TOGETHER WITH:

FLAVIO FERRAN,	CIVIL ENG. COORDINATOR
RAUL GONZALEZ,	MEC. ENG.
LUIS PUENTE,	MEC. ENG.
MARCO BORJA,	ELECTRONIC ENG.
RAUL BORJA,	ELECTRIC ENG.

THE FOLLOWING NOTES WERE MADE:

<u>1990-01-15</u>.

1. FOR THE INTAKE STRUCTURE AND THE TRASH-RACK IT IS RECOMMENDED TO HAVE A WATER VELOCITY OF 1 M/S OR LESS. THIS CAN BE ACHIVED BY LOWERING THE - BOTTOM OF THE INTAKE BY ∿ 0.7 M ACCORDING TO ENCL.1.

2. FOR THE TRASH-RACK IT IS RECOMMENDED TO HAVE THE UPPER END FREE.SEE ENCL. 1. WHEN CLEANING THE RACK IT IS THEN EASY TO REMOVE THE WASTE.

3. FOR THE STEEL PENSTOCK IT IS RECOMMENDED TO HAVE IT IN A TRENCH AND - - EMBEDDED IN SAND. THE NEED FOR SUPPORT WILL THEN BE LESS AS WELL AS -- THE TEMPERATURE EXPANSION AND THE PROBLEM WITH THE EXPANSION JOINT.

4. THE EXPANSION JOINT CAN BE DESIGNED AS A SIMPLE O-RING JOINT WITH TWO - O-RINGS GLIDING AGAINST A STAINLESS STEEL SURFACE THE TIGHTNESS OF THE SEAL CAN THEN BE TESTED BY APPLYING COMPRESSED AIR BETWEEN THE TWO - - O-RINGS.

1990-01-16.

5. AFTER A CLOSE STUDY OF THE LAND CONTOUR IT WAS FOUND IMPOSSIBLE TO PLACE THE PENSTOCK IN A COVERED TRENCH. IN OPEN AIR THE TEMPERATURE SPAN FOR THE PENSTOCK WILL BE 50 - 60°C AND THE EXPANSION 20 - 30 mm, WHICH CAN BE DIFFICULT TO TAKE IN ONE EXPANSION-BOX. ESPECIALLY THE GLIDING SUPPORTS WILL REQUIRE MORE ATTENTION WHEN DESIGNED.

6. HOWEVER, IT WAS RECOMMENDED AFTER DUE CONSIDERATIONS, TO USE ONLY ONE - EXPANSION BOX PLACED NEAR THE UPSTREAM END OF THE PENSTOCK. THE NORMAL 2-0-RING BOX COULD BE USED. FOR THE GLIDING SUPPORT IT WAS RECOMMENDED TO CONTACT DESIGNERS FROM OIL-PIPING INDUSTRY FOR IDEAS. CONTACT WILL ALSO BE TAKEN WITH A SWECO SPECIALIST AFTER RETURN TO SWEDEN.

7. THE WATER-WAY TROUGH THE TURBINE WAS SCRUTINIZED. IT WAS RECOMMENDED TO USE A RADIUS OF $\sim$ 180 mm BETWEEN THE OUTER GUIDE-VANE-RING AND THE RUNNER CHAMBER.

8. THE DRAFT-TUBE WAS RE-DESIGNED IN SOME PARTS. IT WAS RECOMMENDED TO USE A SMALLER DIVERSION ANGLE IN THE BEND AREA IN ORDER TO LIMIT THE RISK -- OF BOUNDARY LAYER SEPARATION.

1990-01-17.

9. THE OVER ALL DIMMENSIONS AND THE WORKING POINT FOR THE RUNNER WERE SCRUTINIZED FROM THE FOLLOWING DATA:

H_n = 7 m, Q = 2 M^3/S
D = .8 m, n = 450 rpm
d_{HUB} = .36 m.

IN UNIT DATA IT GIVES A WORKING POINT AT:

n_{11} = 130 AND Q_{11} = 1.18

COMPARED WITH STATE OF THE ART TODAY THE CHOICE OF WORKING POINT IS SAFE ON THE CONSERVATIVE SIDE.

10. THE DETAILS OF THE RUNNER BLADE SHAPE CALCULATION ACCORDING TO OLADE'S - METHOD WERE SCRUTINIZED. THE MAIN PARAMETERS DEFINING THE SHAPE OF THE BLADE, AS THE DISTRIBUTION ALONG THE RADIUS OF: BLADE-LENGTH (L), ARC HIGHT (F), S-SHAPE, THICKNESS (T) AND SLOP-ANGLE (B) WERE COMPARED WITH A COMPUTER DESIGNED BLADE FOR THE SAME WORKING POINT. SEE ENCL.2. THE COMPUTER PROGRAM, USED DESIGNS A BLADE SHAPE FROM STATISTICAL DATA - - TAKEN FROM A NUMBER OF RUNNERS TESTED IN MODEL TEST-RIGGS.

11. FROM THIS COMPARISON IT CAN BE JUDGED THAT THE RUNNER, DESIGNED ACCORD-- ING TO OLADE'S METHOD, WILL HAVE AN OPTIMAL WORKING POINT AT OR NEAR - - THE EXPECTED. AN EFFICIENCY AT THE OPTIMUM POINT OF 87 - 90% CAN ALSO - BE EXPECTED.

12. THE COMPUTER DESIGN IS SHOWN IN ENCL.3 AND AN EXPLANATION FOR SYMBOLS - USED IN ENCL.4.

13. A COMPUTER PROGRAM TRANSFORMING THE DESIGN PARAMETERS TO CYLINDRICAL - - COORDINATES FOR THE BLADE SHAPE WAS ALSO DEMOSTRATED - ENCL.5.

1990-01-18.

14. THE DESIGN OF GUIDE-VANE-MECHANISM TOGETHER WITH THE FORCE AND STRENGTH CALCULATIONS MADE WERE SCRUTINIZED. THE CALCULATIONS HAD BEEN MADE - - ACCORDING TO OLADE'S METHOD. THE PROFILE USED FOR THE VANE SHAPE IS -- SUITABLE. THE OVERLAPPING OF 25% IS ON THE LARGEST LIMIT. THE POSITION OF THE SHAFT CENTRE IN THE PROFILE GIVES A SUITABLE DISTRIBUTION OF THE MANEUVER FORCES.

15. CHECK-CALCULATIONS OF THE FORCES AND STRENGTH-FIGURES FOR THE VANES, - - SHOWED GOOD CORRESPONDENCE WITH CALCULATIONS USING OLADE'S METHOD. THE STRENGTH IN THE VANE-SHAFT IS $\sim$ 2,300 KG/CM^2. IT WAS RECOMMENDED TO USE

A MATERIAL SIMILAR TO TEFLON FOR THE SHAFT BEARINGS AND U-RING OR O-RING FOR SEALING. AS THE BEARING IS WATER-LUBRICATED AND THE STRENGTH IN THE SHAFT IS HIGH, IT IS RECOMMENDED TO USE STAINLESS STEEL IN THE SHAFTS.

16. THE INSTITUTE HAS ORDERED A GOVERNOR-OIL PRESSURE-UNIT FROM BRASIL. THE CAPACITY OF THE PRESSURE UNIT 75 L-VOLUME AND 275 KGM-TORQUE WAS - - FOUND TO BE SUFFICIENTE. HOWEVER, MORE KNOWLEDGE ABOUT THE SERVOMOTOR IS REQUIRED BEFORE MAKING THE FINAL DESIGN OF THE OPPERATING RING AND THE LINK MECHANISM. THE QUESTION, IF THE OPERATING RING SHOULD BE - - PLACED UPSTREAM OR DOWNSTREAM OF THE GUIDE-VANES, WAS LEFT OPEN.

17. THE CALCULATION OF THE FLYWHEEL REQUIREMENT WAS MADE ACCORDING TO A CRITERION FROM OLADE, AND BASED UPON HAVING A SPEEDRISE OF MAXIMUM 30%. A RECALCULATION, OF THE ACCELERATION TIME FOR THE SHAFT SYSTEM GAVE T_a = 11.8 SEC. IT CAN BE REGARDED AS A VERY SAFE FIGURE AND IT MAKES POSSIBLE TO REGULATE THE TURBINE FREQUENCY IN AN ISOLATED NET.

18. THE STRENGTH CALCULATIONS OF THE FLYWHEEL GAVE FOR ALL NORMAL OPERATION OF THE UNIT SAFE FIGURES. FOR THE WORSE CONDITION, THE RUNAWAY SPEED, THE STRENGHT OF THE MATERIALS IS AT THE YIELD-POINT. TAKING INTO CONSIDERATION THE LOW PROBABILITY FOR SUCH A CONDITION, IT CAN BE REGARDED AS AN OPTIMAL DESIGN.

19. IT IS PLANED TO HAVE A DISK BRAKE OF STANDARD TYPE ON THE TURBINE SHAFT AS A SAFETY DEVISE. THE DISK IS AIR VENTILATED. CALCULATIONS MADE - - ACCORDING TO CATALOGUE INFORMATION SHOWED THAT THE BRAKE COULD STOP THE UNIT IN LESS THAN 5 SEC. THERE WAS NO POSSIBILITY TO CHECK THE CATALOGUE DATA. THE ARRANGEMENT REQUIRES A SAFE OIL SUPPLY AT A PRESSURE - OF 70 BAR.

20. RETURNING TO THE SHAFT SYSTEM IT WAS RECOMMENDED TO MAKE ARRANGEMENTS - FOR TAKING CARE OF A FORCE FROM THE RUNNER DIRECTED UPSTREAM AND OCCURRING AT TRANSIENT CONDITION AND OF THE SAME MAGNITUDE AS THE DOWNSTREAM FORCE.

21. IT WAS RECOMMENDED TO USE 10 STAY VANES FOR SUPPORTING THE UPSTREAM BULB CONTAINING THE SHAFT BEARING. IT WAS ALSO RECOMMENDED TO, IF POSSIBLE, USE A CONICAL SHAPE OF THE BULB AT THE POSITION OF THE STAY VANES IN - - ORDER TO COMPENSATE FOR THE AREA OCCUPIED BY THE VANES.

1990-01-19.

22. THE STRENGTH CALCULATION FOR THE TURBINE SHAFT WAS SCRUTINIZED. THE - - DIAMETER OF THE SHAFT IS 3 1/4 INCH AND THE SHAFT IS ALREADY MANUFACTURED. A MAXIMUM STRENGTH INCLUDING STRENGTH CONCENTRATIONS, OF 1,850 - KG/CM^2 WAS FOUND. THIS FIGURE IS VERY HIGH - THE NORMAL STRENGHT LEVEL IN THE BRANCH IS $<$ 800 KG/CM^2. THE MAIN REASON FOR THE LOW LEVEL IS -- RISK FOR CORROSION-FATIGUE AND CRITICAL SPEED.

IN ORDER TO COPE WITH THE HIGH STRENGTH, A ALLOYED STEEL WAS CHOSEN FOR THE SHAFT. THE FATIGUE LIMIT FOR THE MATERIAL WAS, ACCORDING TO SPECIFICATIONS, $\sim$ 3900 KG/CM^2 THUS GIVING A SAFETY FACTOR OF 2.1.

23. THE FIRST TWO, CRITICAL SPEEDS OF THE TURBINE SHAFT SYSTEM WERE CALCULATED TO $\sim$ 470 AND 8400 RPM. THE NORMAL OPERATING SPEED FOR THE SHAFT IS 450 RPM. THIS MEANS THAT THE FIRST CRITICAL SPEED IS IN THE SPEED RISE RANGE 100 - 130% SPEED.

THE SHAFT SYSTEM HAS TO BE REDESIGNED. TWO WAYS WERE RECOMMENDED: EITHER TO MAKE A NEW SHAFT WITH A BIGGER DIAMETER OR TO MOVE THE BEARING IN THE BULB AS NEAR THE RUNNER AS POSSIBLE. THE EXISTING SHAFT CAN STILL BE USED IF A FIRST CRITICAL OF > 1,350 RPM IS CALCULATED AFTER USING -- THE SECOND RECOMMENDATION.

NORMAL PRACTICE IS TO HAVE THE FIRST CRITICAL SPEED $\sim$30% HIGHER THAN - THE RUNAWAY SPEED (ASSUMED TO BE $\sim$ 1,050 RPM).

24. THE SPEED-INCREASER IS A NORMAL ONE STEP INCREASER ON INDUSTRIAL TYPE. THE DATA WAS 242 HP AT 450/1,850 RPM. RECALCULATED TO KW IT IS 181 KW.

THE HIGHEST OUTPUT FROM THE TURBINE IS ASSUMED TO BE ∿ 112 KW. THAT GIVES A SAFETY FACTOR OF 1.6 WHICH IS ON THE LOWER LIMIT FOR TURBINE USE.

THERE WAS NO INFORMATION ON THE BEARING CALCULATION. A LIFETIME OF 100,000 HRS IS NORMALLY REQUIRED.

THERE WAS NO AUTOMATIC DEVICE FOR PROTECTION OF THE OIL-LEVEL AND TEMPERATURE RISE WHICH IS NECESSARY.

25. THE AMBIENT TEMPERATURE FOR THE GENERATOR AND THE SPEED-INCREASER WAS DISCUSSED. THE LOSSES FOR THE GENERATOR CAN BE ASSUMED TO BE 5 KW AND FOR THE SPEED-INCREASER TO BE 2 KW. WITH A TEMPERATURE FOR THE AIR SURROUNDING THE POWERHOUSE OF 30°C IT CAN BE DIFFICULT TO SATISFY THE REQUIREMENTS OF MAXIMUM AMBIENT TEMPERATURE OF 40°C FOR THE GENERATOR. IT WAS RECOMMENDED TO CHECK IF WATER-COOLING IN THE BUILDING IS NECESSARY.

1990-01-22.

26. THE DRAFT TUBE DESIGN WAS DISCUSSED. FOR MINIMIZING THE LOSSES IN THE BEND THE INNER CURVATURE SHOULD BE SHAPED AS A CIRCULAR ARC. THE BEND COULD BE MADE UP FROM 6 SECTIONS.

THE DRAFT TUBE AS WELL AS OTHER PIPINGS SHOULD BE SANDBLASTED AND PAINTED WITH EPOXY-OR ASFALT-PAINT. AS THE WATER IS COLDER THAN THE AIR THERE WILL BE CONDENSED WATER ON THE OUTSIDE OF THE STRUCTURE FACILITATING THE RUST-PROCESS.

THE MAIN DIMMENSIONS OF THE DRAFT TUBE COULD BE MADE AS ONE OF THE TWO ALTERNATIVES ENCL.6.

27. THE INTENTION OF THE INSTITUTE IS TO USE A SOUND-AND VIBRATION DAMPING MATERIAL BETWEEN THE STEEL-FRAME SUPPORTING THE GENERATOR AND THE CONCRETE. THE BRAND OF THE MATERIAL IS "VIBRA-CHECK". FOR THE NORMAL SPEED

OF THE UNIT 1,850 RPM = 30.8 Hz THE DAMPING OF VIBRATION AMPLITUDES IS, ACCORDING TO CATALOGUE DATA, 30%. THE MANUFACTURER ALSO RECOMMENDED NOT USE ANCHORBOLTS.

AFTER A DUE CONSIDERATION OF THE FACTS INVOLVED IT WAS RECOMMENDED TO USE THE "VIBRA-CHECK" MATERIAL, BUT TO PUT IN ALIGNMENT-PLATES IN BETWEEN -- THE "VIBRA-CHECK" AND THE FOOT-PLANES OF THE FRAME. FOR THE QUESTION - - WHETHER TO USE OR NOT USE ANCHOR-BOLTS, IT WAS RECOMMENDED TO CALCULATE - THE SHORT-CIRCUIT TORQUE FROM THE GENERATOR AND THE REACTION TORQUE FROM THE INTERTIA IN FLYWHEEL AND GENERATOR ROTOR IN ORDER TO SEE IF THERE IS A TILTING TORQUE ON THE FRAME SYSTEM. IF, INSIDE A LARGE MARGIN OF SECURITY, THERE OS A RISK OF TILTING, THEN ANCHOR-BOLTS HAVE TO BE USED FOR AT LEAST SOME OF THE FEET OF THE FRAME.

THE SUPPORT OF THE FRAME COULD BE DESIGNED ACCORDING TO ENCL. 6.

28. RETURNING TO THE FLYWHEEL, IT WAS FOUND OUT THAT THE SHRINKAGE, OF THE ALREADY MANUFACTURED SHAFT SHRINKED WHEEL, WAS 0.002". IT WAS CALCULATED THAT THE SWELLING OF THE HOLE FOR THE SHAFT WILL BE ∿ 0.0036" AT RUNAWAY CONDITION. IT MEANS THAT THE FLYWHEEL WILL BE FREE TO CHANGE POSITION ON THE SHAFT AT RUNAWAY CONDITION.

AS THE PROBABILITY FOR RUNAWAY CONDITION IS VERY LOW WHEN, AS INTENDED, 2-3 SAFETY SYSTEM TO PREVENT RUNAWAY IS ISED, IT WAS RECOMMENDED NOT TO REDESIGN THE FLYWHEEL. FOR THE SAFETY OF PERSONNEL IT WAS RECOMMENDED TO SHRINK ON A FLANGE HINDERING THE WHEEL FROM MOVING IN THE AXIAL DIRECTION, AND USING A RATHER LARGE U-SECTION PLACED ∿ 1.5" OUTSIDE THE PERIPHERY OF THE WHEEL. THIS U-SECTION SHOULD BE WELL ANCHORED IN THE STEEL-FRAME. SEE ENCL.7.

29. AS A MEANS OF MAKING THE GOVERNOR, THE BRAKE AND POSSIBLY THE INTAKE GATE TO REACT SAFELY AT OVERSPEED, AN IDEA OF USING A MECHANICAL PENDULUM FOR TRIGGING THE SIGNAL WAS INTRODUCED. THE PENDULUM COULD BE USED, IN PRINCIPLE, ACCORDING TO ENCL.7. WHEN AT OVERSPEED ONE OF THE TWO PENDULUMS -

GOES OUT, UNDER THE INFLUENCE OF THE CENTRIFUGAL FORCE, IT SHOULD, AT A CALCULATED AND CHECKED SPEED, TOUCH A TRIGGER TO OPEN HYDRAULIC VALVES, CAUSING THE GOVERNOR TO CLOSE THE GUIDE-VANES AND MAKING THE BRAKE TO - BRAKE THE SPEED. WITH SUCH A DEVICE THE UNIT WILL PROTECT ITSELF, AS - THE LAST RESORT, IN CASE OF COMPLETE BRAKE DOWN OF THE ELECTRIC SAFETY SYSTEM. THIS SAFETY DEVICE IS COMMONLY USED IN NORTHERN EUROPE BOTH - FOR SMALL AND LARGE WATERTURBINES.

1990-01-23.

30. THE DATA-HANDLING SYSTEM INTENDED FOR USE AT THE SITE WAS DEMONSTRATED - AND DISCUSSED. THE FOLLOWING SENSORS WERE PLANNED.

1.- TWO PRESSURE SENSORS. IT WAS RECOMMENDED TO USE ONE IN THE PENSTOCK IMMEDIATELY UPSTREAM FROM THE TURBINE AND ONE AT THE ENTRANCE OF THE DRAFT TUBE BEND AT THE INNER RADIUS.

2.- A FLOW-DISCHARGE TRANSDUCER TO BE PLACED IN THE PENSTOCK. THE TRANSDUCER WAS OF THE SAME TYPE AS THAT USED FOR BOOT-SPEED MASUREMENT.

3.- A POSITION SENSOR FOR THE GUIDE-VANE MOVEMENT.

4.- TEMPERATURE SENSORS FOR THE BEARINGS, THE SPEED-INCREASER AND THE - GENERATOR.

5.- ELECTRICAL SENSOR FOR THE OUTPUT AND THE VOLTAGE.

IT WAS RECOMMENDED TO ADD A VIBRATION SENSOR, PLACED WHERE IT IS FOUND MOST SIGNIFICANT DURING MEASUREMENT AT COMMISSIONING, AND A TEMPERATURE - SENSOR FOR THE AIR IN THE POWERHOUSE.

IT WAS ALSO RECOMMENDED TO INVESTIGATE IF THE SAME SYSTEM COULD BE USED - FOR MEASURING PRESSURE, SPEED AND GUIDE-VANE MOVEMENT UNDER TRANSIENT - CONDITION AND LOAD-REJECTIONS.

31. IT WAS RECOMMENDED TO INQUIRE IF THERE IS A MANUFACTURER OR A REPRESENTATIVE FOR A FOREIGN MANUFACTURER OF WATER LUBRICATED BEARINGS FOR FAST - RUNNING SHAFTS IN MEXICO. A COUPLE OF PROBLEMS IN DESIGN, ASSEMBLING AND OPERATING THE TURBINE SHOULD BE SIMPLIFIED IF A WATER-LUBRICATED BEARING COULD BE USED IN THE BULB. MOST OF THE MINITURBINES FOR LOW HEAD IN THE WORLD USE WATER-LUBRICATED BEARINGS IN THAT POSITION.

WHEN USING A WATER-LUBRICATED BEARING A SHAFT-SLEEVE OF STAINLESS STEEL SHOULD BE USED. THE MATERIAL SHOULD BE A MARTENCITIC STEEL OF TYPE 13% Cr, 1% Ni AND TEMPERED TO ∿ 300 HB. STAINLESS STEEL OF THE TYPE 18% Cr, -- 8% Ni IS NOT SUITABLE.

32. THE FOLLOWING PROCEDURE IS RECOMMENDED WHEN STARTING UP A UNIT:

BEFORE THE COMMISSIONING OF THE UNIT.

1.- CHECK AND MEASURE ALL CLEARANCES FOR MEASURING THE CLEARANCE. AT THE RUNNER TIP THE SHAFT HAS TO BE TURNED TO AT LEAST 4 POSITIONS.

2.- CHECK ALL SEALINGS. WHERE POSSIBLY WITH COMPRESSED AIR.

3.- CLEAN AND INSPECT ALL THE WATER-WAY.

4.- BLOCK THE GUIDE-VANES IN CLOSED POSITION.

START OF COMMISSIONING.

5.- FILL SLOWLY THE UPSTREAM WATER-WAY FROM THE INTAKE GATE CHECK FOR - - LEAKAGE.

6.- OPEN THE GUIDE-VANES MANUALLY AND SLOWLY TO AN OPENING WHERE THE - - SHAFT JUST STARTS TO ROTATE AT A LOW SPEED. CHECK THE ROTATING SYSTEM FOR VIBRATIONS, RUN OUT AND ANORMAL BEHAVIOUR. STOP THE UNIT FOLLOWING THE EMERGENCY STOP-SEQUENCE.

7.- START THE UNIT ACCORDING TO POINT 6 AND INCREASE THE SPEED IN STEPS 25, 50, 75 AND 100% OF NORMAL OPERATING SPEED. FOR EACH STEP THE - SPEED SHOULD BE KEPT CONSTANT UNTIL THE BEARING TEMPERATURES HAVE - REACHED A CONSTANT VALUE. THE ROTATING SYSTEM SHOULD ALL THE TIME BE CHECKED FOR VIBRATIONS AND OTHER ANORMAL BEHAVIOUR. WHEN THE BEARING TEMPERATURES HAVE CONSTANT VALUES AT 100% SPEED STOP THE -- UNIT BY THE NORMAL STOP-SEQUENCE.

8.- START THE UNIT MANUALLY AND INCREASE THE SPEED TO ∿ 130% OF NORMAL SPEED WHERE THE MECHANICAL PENDULUM SHOULD STOP (TRIP) THE UNIT. IF THE UNIT IS NOT TRIPPED AT THE EXPECTED SPEED THE PENDULUM HAS TO BE ADJUSTED AND THE TEST HAS TO BE REPEATED.

9.- THE PROTECTION PARAMETERS SUCH AS TEMPERATURE, OIL-LEVEL AND OIL-PRESSUPRE SHOULD BE ADJUSTED FOR ALARM AND TRIP, AND THE TRIP FROM - THE PROTECTION SYSTEM TESTED.

10.- THE SPEED FOR CONNECTING THE GENERATOR TO THE GRID IS ADJUSTED AND TESTED.

11.- THE GENERATOR IS CONNECTED TO THE GRID AND THE LOAD SLOWLY INCREASED TO 25% OF NOMINAL LOAD. THE UNIT SHOULD BE CHECKED FOR VIBRATIONS - AND OTHER ANORMAL BEHAVIOUR. THE LOAD SHOULD THEREAFTER BE TRIPPED (REJECTED) VIA THE LOAD-BRAKER.

12.- POINT 11 IS THEN REPEATED FOR 50, 75, 100% LOAD AND FOR EVENTUAL - - OVERLOAD.

DURING ALL THE LOAD-REJECTIONS THE FOLLOWING TRANSIENT PARAMETERS - SHOULD BE RECORDED.

a).- PRESSURE AND PRESSURE RISE IN THE PENSTOCK AT TURBINE INLET.

b).- PRESSURE AND PRESSURE FLUCTUATIONS AT THE INLET OF THE DRAFT - TUBE.

c).- SPEED AND SPEED RAISE.

d).- THE MOVEMENT OF THE GUIDE-VANES.

13.- THE CLOSING OF THE TURBINE FOR A MECHANICAL FAULT IS TESTED FROM, FOR INSTANCE, A BEARING TEMPERATURE SENSOR.

THE GUIDE-VANES SHOULD THEN BE CLOSED WITH THE GENERATOR CONNECTED TO THE GRID. THE LOAD BRAKER SHOULD BE TRIPPED FROM THE INVERSE POWER SENSOR.

14.- THE UNIT SHOULD BE TEST RUN AT NOMINAL LOAD FOR ONE WEEK. THE TEMPERATURE OF THE GENERATOR, THE SPEED-INCREASER AND THE BEARINGS SHOULD BE CHECKED TOGETHER WITH OTHER PARAMETERS FOUND RELEVANT FROM THE PREVIOUS TESTS.

1990-01-24.

33. IN CONNECTION TO A DISCUSSION ABOUT CAVITATION DAMAGE AND CAVITATION SAFETY, A FORMULA FOR A ROUGH ESTIMATION OF THE SAFETY AGAINST CAVITATION DAMAGE WAS DEMONSTRATED. SEE ENCL.8.

34. RETURNING TO THE DRAFT TUBE DESIGN, A DIAGRAM SHOWING CURVES FOR BOUNDARY LAYER SEPARATION WAS DEMONSTRATED. SEE ENCL.9. THE USE OF THE DIAGRAM WHEN DESIGNING DRAFT TUBES WAS ALSO DEMONSTRATED.

35. THE INSTITUTE HAS THE INTENTION TO MAKE AND TEST A MODEL TURBINE WITH A RUNNER DIAMETER OF 300 mm. THE FACILITIES FOR MAKING THE TESTS WERE INSPECTED. THEY ARE SITUATED OUTSIDE THE INSTITUTE IN A LABORATORY AREA WHERE NORMALLY WATER-WAYS ARE MODELLED AND TESTED. THERE IS AN UPSTREAM AND A DOWNSTREAM BASIN GIVING THE POSSIBILITY FOR A 3 TO 4 M TEST HEAD, AND PUMP CAPACITY FOR A 0.2 M^3/S DISCHARGE. THE DISCHARGE COULD BE MEASURED BY A WEIR. THE FACILITIES WERE FOUND SUITABLE, EXCEPT THAT THE

UPSTREAM AND DONSTREAM LEVELS MAY BE DIFFICULT TO KEEP CONSTANT DURING - THE TESTS. THE TEST FACILITIES CORRESPOND WELL WITH THE MODEL RUNNER -- DIAMETER OF 300 mm AND THE ACTUAL WORKING POINT.

36. THE OUTPUT BRAKE ARRANGEMENT TOGETHER WITH OTHER ARRANGEMENTS FOR MEASURING HEAD, DISCHARGE, SPEED AND TORQUE WERE DISCUSSED. MOST OF THE EQUIPMENT - WAS ALREADY DESIGNED AND MOST OF THE MODEL TURBINE MANUFACTURED. AN OVER ALL SURVEY OF THE ARRANGEMENT POINTED OUT THAT THE TESTS WILL BE TIME CONSUMING AND THE ACCURACY ABOUT ± 1.5%. THAT IS FULLY COMPARABLE WITH - -- MODERN FIELD TESTS, BUT LESS ACCURATE THAN A MODERN MODEL TEST RIGG HAVING ACCURACY FIGURES OF ± 0.3%.

37. FOR THE TEST PROCEDURE IT WAS RECOMMENDED TO USE UNIT-VALUES n_{11} AND Q_{11} FOR CHOOSING THE TEST POINTS AND PRESENTING THE RESULTS. SUCH A TEST - - PROCEDURE IS DESCRIBED FOR A SWEDISH TEST RIGG IN ENCL.10. IT WAS RECOMMENDED TO USE A UNIT SPEED RANGE n_{11} = 120 - 150 INCREMENTED PARTH IN 5 STEPS AND A RANGE OF RUNNER BLADE ANGLES OF ± 4° FROM THE DESIGN POINT. FOR GUIDE-VANE ANGLES, 10 STEPS HAVE TO BE TESTED FOR EACH UNIT SPEED AND EACH GUIDE-VANE ANGLE, GIVING ALLTOGETHER ∿ 250 TEST POINTS. A GOOD - - ESTIMATE IS 30 HOURS OF OPERATIONS IN THE TEST RIGG FOR TESTING ONE MODEL RUNNER.

38. FOR THE SELECTION OF THE MATERIAL FOR THE RUNNER, IT WAS RECOMMENDED TO - USE AN ALBRONZE ACCORDING TO ENCL.11. AS A SECOND CHOOSE A STAINLESS -- STEEL. A ∿ 18% Cr AND ∿ 8% Ni STEEL MAY BE USED FOR THIS LOW HEAD. A COMMONLY USED STAINLESS STEEL TODAY FOR BIG TURBINES AND HIGH HEAD IS ASTM A 296, CA-6MN. SEE ENCL.12.

A LOW CARBON STEEL COULD BE USED BUT THEN THE RUNNER HAS TO BE PAINTED AND THERE MAY BE RUST AND CAVITATION PROBLEMS IN THE CLEARANCE OF .6 - .8 mm - AT THE PERIPHERY.

39. IT WAS RECOMMENDED TO USE BRONZE OR STAINLESS STEEL FOR THE RUNNER CHAMBER AROUND THE PERIPHERY OF THE RUNNER. THE REASON IS THAT THIS SURFACE

CAN NOT BE PAINTED.

40. THE MANUFACTURING PROCEDURE FOR THE RUNNER WAS DISCUSSED. SOME COMMONLY USED PROCEDURES WERE DESCRIBED. FOR CHECKING THE SHAPE THE SIMPLEST WAY IS TO USE A RIGG OF TEMPLATES FOR THE SUCTION SIDE OF THE BLADE. IF THE BLADES SHOULD BE WELDED TO THE HUB SUCH A RIGG HAS TO BE USED BOTH FOR FITTING THE BLADE TO THE HUB AND FOR POSITIONING THE BLADE ON THE HUB FOR WELDING.

IT WAS RECOMMENDED TO USE A SURFACE TOLERANCE OF $\overset{0.1}{1\%}$ OF THE RUNNER DIAMETER - FOR THE ACTUAL CASE ± 0.8 mm. THE SURFACES HAVE TO BE POLISHED. IT IS RECOMMENDED TO USE AN OVERTHICKNESS OF 1. TO 1.5 mm ON EACH SIDE ON THE BLADE FOR GRINDING AND POLISHING THE SURFACE.

<u>1990-01-25</u>.

41. THE CONNECTION OF THE MINIPOWER UNIT TO THE HIGH VOLTAGE LINE WAS DISCUSSED. QUESTIONS ABOUT THE PERSONNEL SAFETY WHEN MAINTAINING THE LINE AND THE MINITURBINE BY MISTAKES IS STARTED AND CONNECTED TO THE LINE CAME UP. THESE QUESTIONS WILL BE CARRIED TO SWECO EXPERTS AND ANSWERED LATER BY TELEFAX COMMUNICATIONS TOGETHER WITH EXISTING KNOWLEDGES ABOUT TO CONNECT A SMALL UNIT TO A BIG HIGH VOLTAGE LINE.

42. THE BRASILIAN GOVERNOR WAS DELIVERED TO THE INSTITUTE YESTERDAY. AFTER INSPECTION THE FOLLOWING POINTS WERE NOTED AND SHOULD BE CHECKED.

1.- LOW PRESSURE -- STOP THE UNIT.

2.- LOW OIL LEVEL IN THE GOVERNOR - STOP THE UNIT.

3.- LOW LEVEL IN THE PRESSURE VESSEL - STOP THE UNIT.

4.- HIGH TEMPERATURE OF THE OIL - STOP THE UNIT.

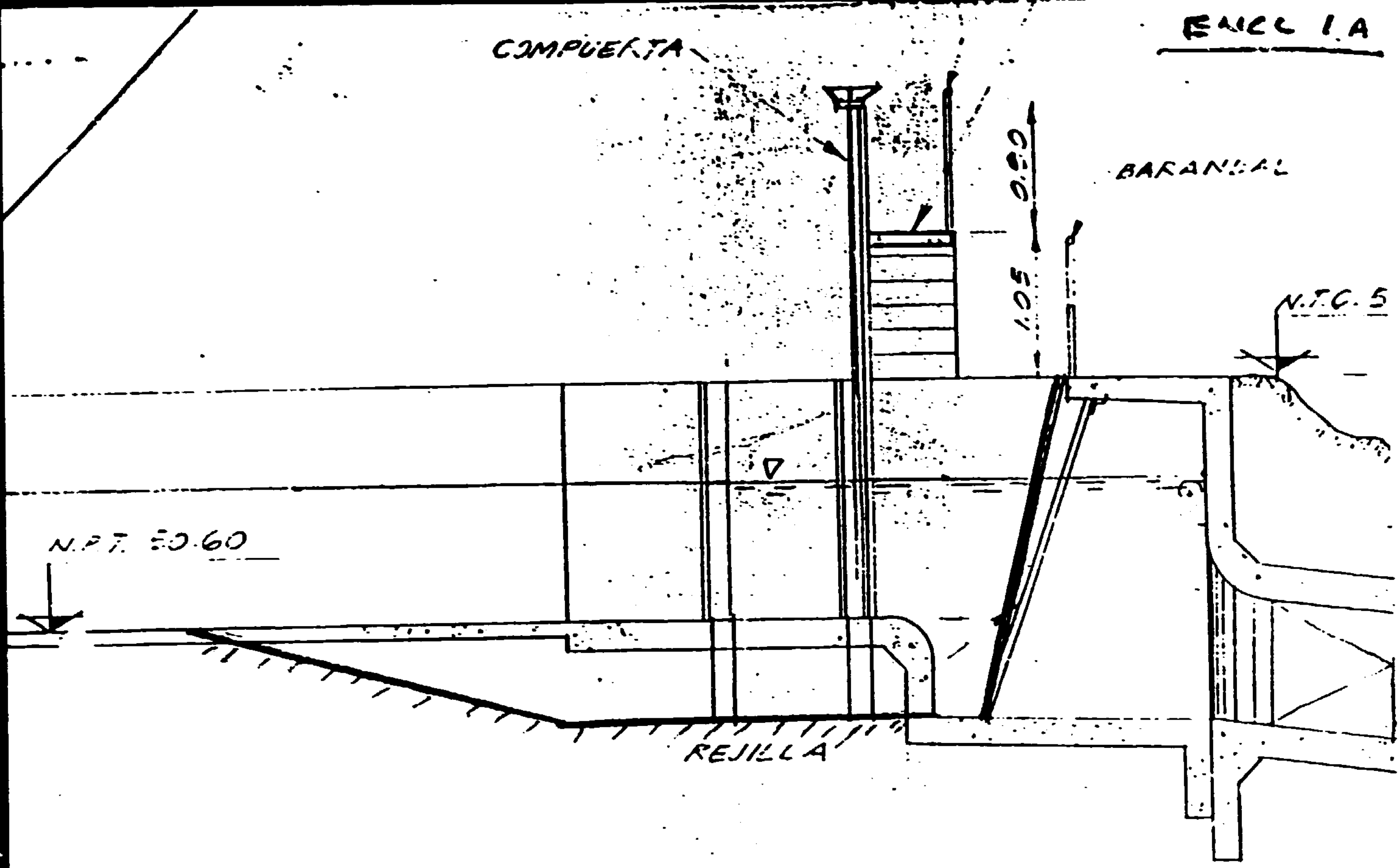
HIGH OIL LEVEL IN THE PRESSURE VESSEL MEANS THAT COMPRESSED AIR IS REQUIRED, THIS IS PROTECTED BY THE LOW OIL LEVEL IN THE GOVERNOR AND SHOULD BE BALANCED. SEE ENCL.13.

CUERNAVACA, MEXICO. 1990-01-26.

EVALD HOLMEN

EH/FFR/rce*

ENCL 1.A



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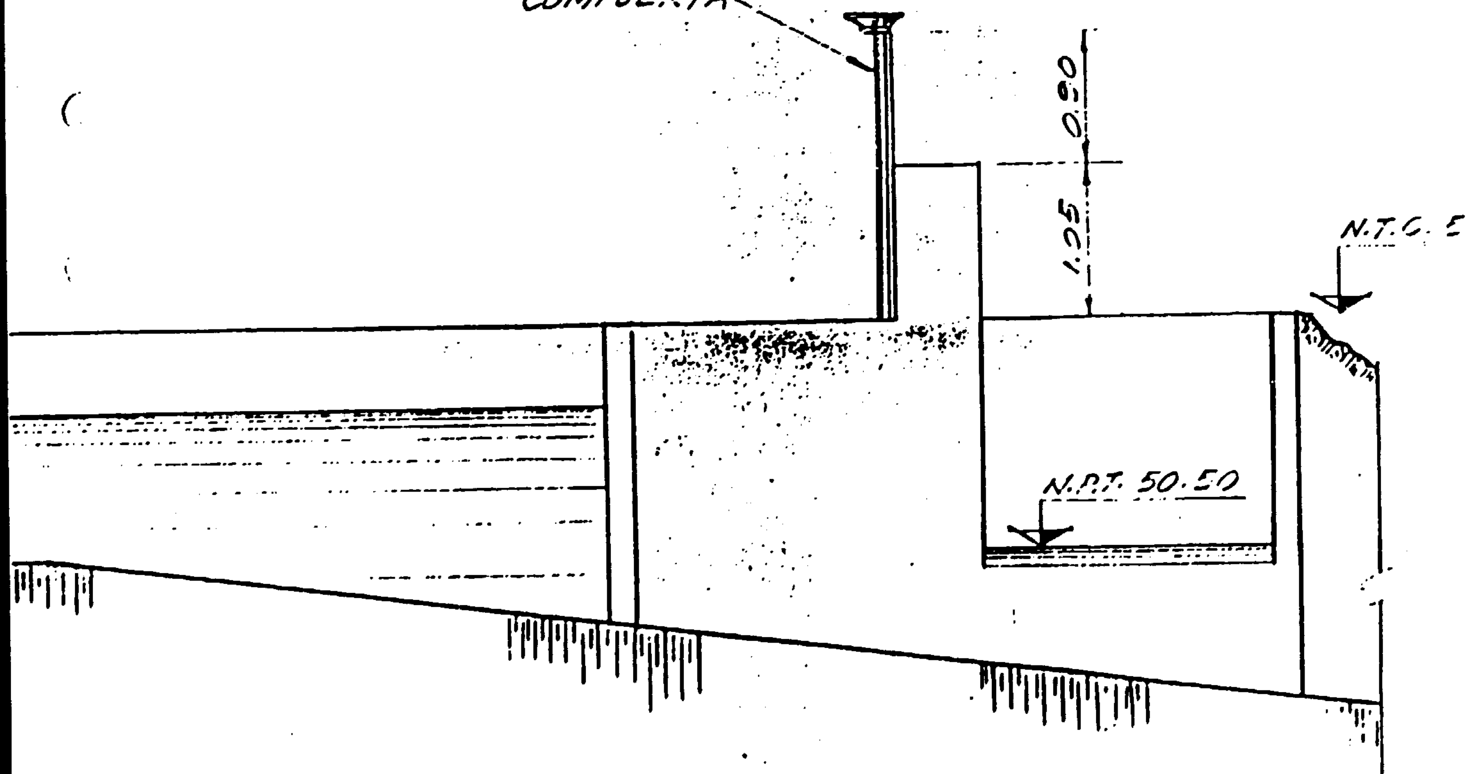
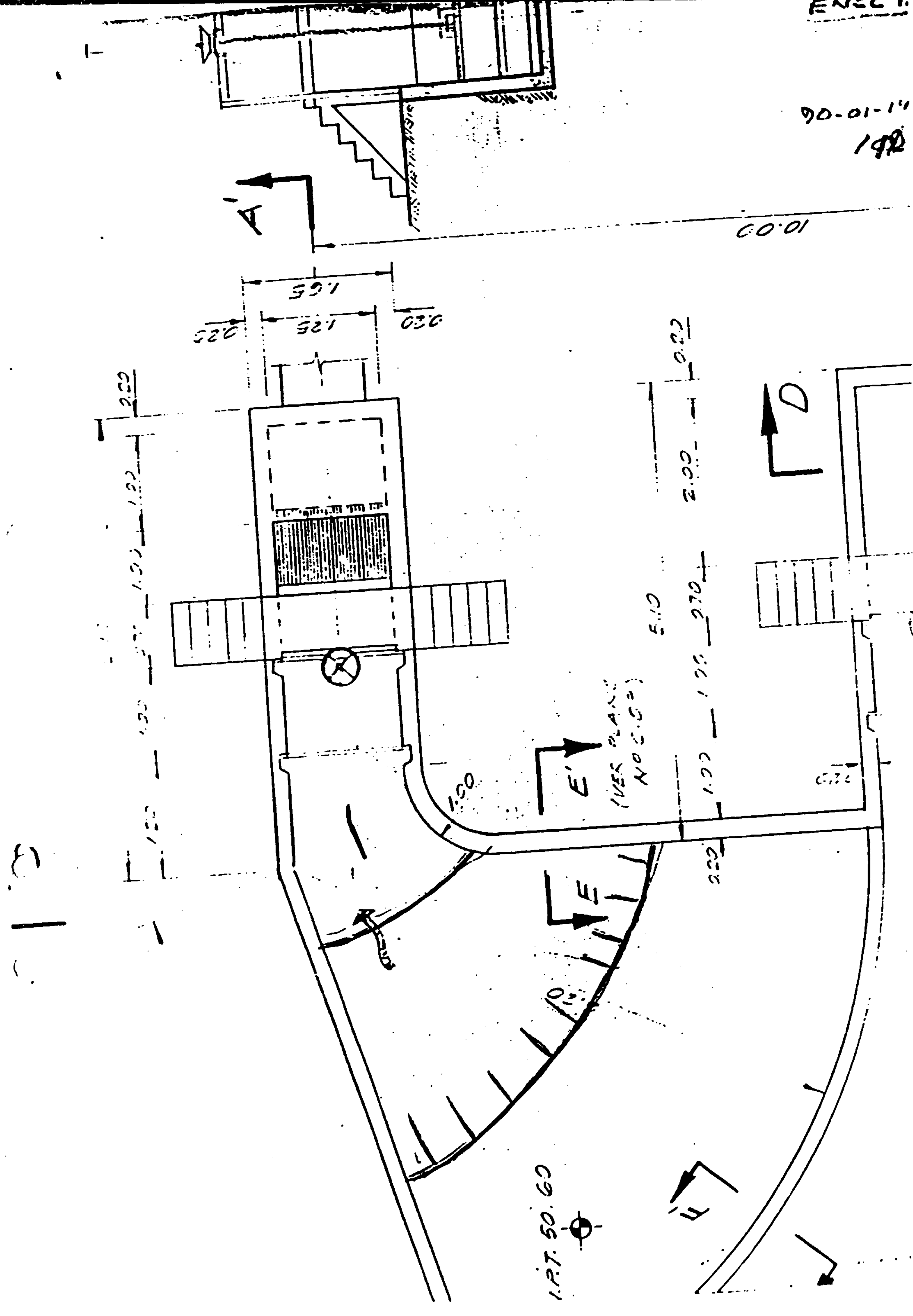
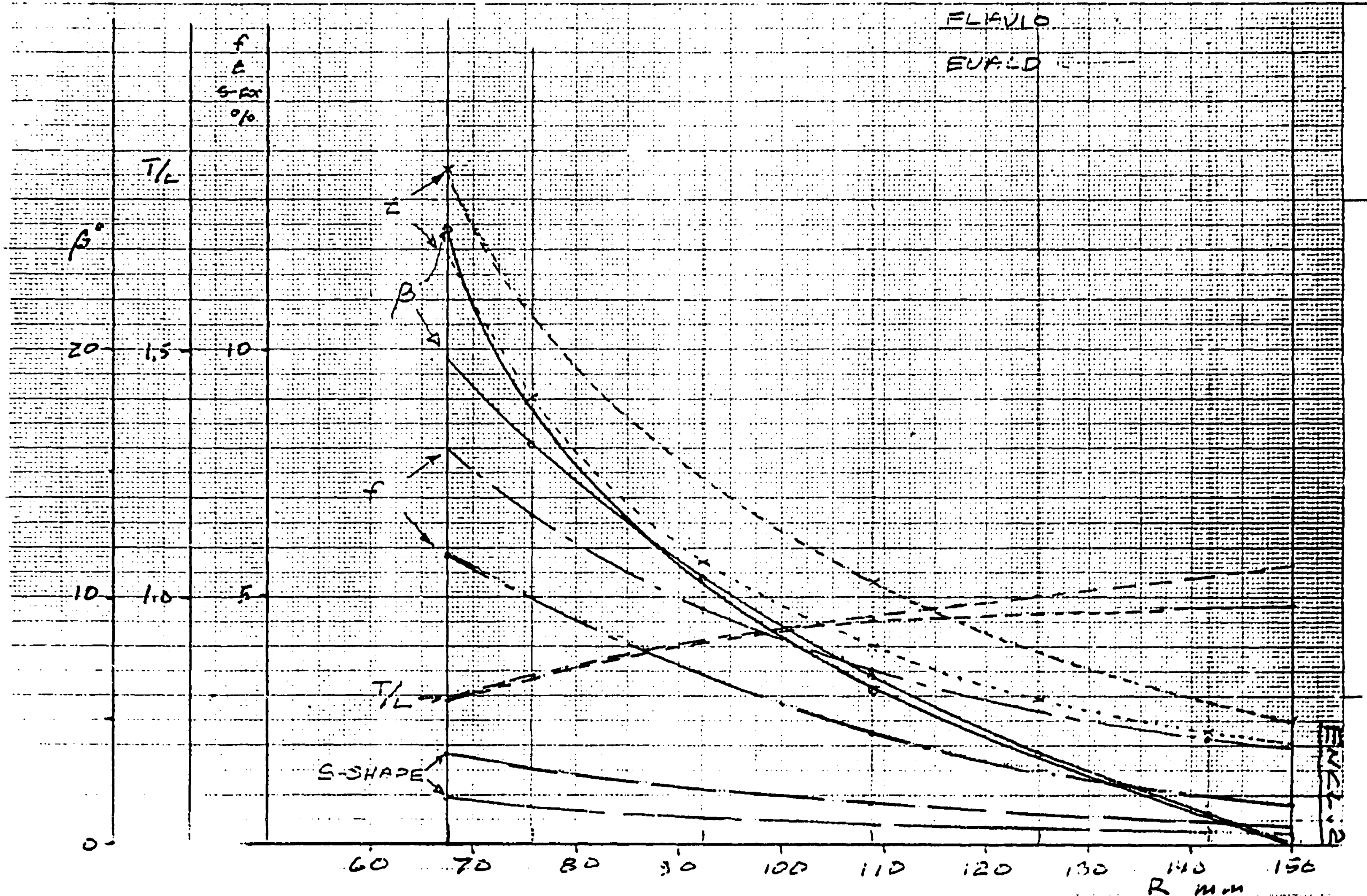


ENCL

90-01-14

192




FLAVLO
EVALD
ENCL.2
f
ε
s-ex
%
β°
T/L
Z
β
f
T/L
S-SHAPE
20
1.5
10
10
1.0
5
0
60
70
80
90
100
110
120
130
140
150
R mm

ENCL 2

CALCULATION OF RUNNER BLADE [illegible]

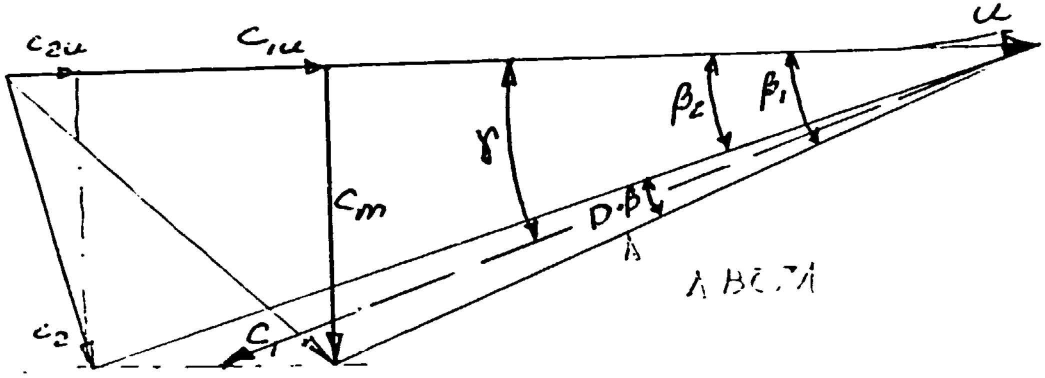
ENCL. 3

```
Runner identif.  Flavio - Evald
Diameter          D=   .8      m
Net head          H=   7       m
No of blades     z1=   5
Discharge         Q=   2.00    m3/s
Speed             n= 449.8    rpm
Unit speed      n11= 136.0
Unit disch      Q11=    1.180
Median vel       cm=    4.98  m/s
Hub ratio        ny=    0.45
Angular vel       Ω=   47.10  rad/s
Inlet angle    deg=    40.3


Calculated values for 5 cylindersections


Radius       m        0.3780      0.3340      0.2900      0.2460      0.2020
u          m/s       17.80       15.73       13.66       11.59        9.51

Beta1      deg       20.16       23.43       27.86       34.12     [illegible]
c1u        m/s        3.47        3.93        4.53        5.33        6.50
whirl-in  m2/s        1.312       1.312       1.312       1.312       1.312

Beta2      deg       16.30       17.90       19.66       21.44     [illegible]
c2u        m/s        0.76        0.30       -0.29       -1.10     [illegible]
whirl-ou  m2/s        0.287       0.101      -0.085      -0.271     [illegible]

D-Beta     deg        3.86        5.53        8.21       12.68     [illegible]
Gamma      deg       18.23       20.67       23.76       27.78       33.17
T/L                   1.041       1.001       0.961       0.912     [illegible]

My                    0.322       0.359       0.407       0.466       0.531
Teta       deg        9.61       12.33       16.12       21.77     [illegible]

Profile data:

Length       m        0.4565      0.4195      0.3794      0.3388    [illegible]
Bend.radi.   m        2.726       1.954       1.353       0.927     [illegible]
Atta.ang   deg        0.773       1.106       1.641       2.576     [illegible]

Arc height   %        2.10        2.69        3.52        4.76      [illegible]
    do.      m        0.0096      0.0113      0.0134      0.0161    [illegible]
S-shape      %        0.22        0.29        0.38        0.51        0.71
    do.      m        0.0010      0.0012      0.0014      0.0017    [illegible]

Thickness    %        2.28        2.92        3.99        5.67        9.01
    do.      m        0.0104      0.0122      0.0151      0.0192    [illegible]
Outl.thick   %        0.35        0.38        0.42        0.47      [illegible]
    do.      m        0.0016      0.0016      0.0016      0.0016    [illegible]

Slop.ang.deg          1.22        3.65        6.75       10.77      [illegible]

Turning-x    m        0.205       0.189       0.171       0.152     [illegible]
Turning-y    m       -0.0079     -0.0052     -0.0016      0.0075    [illegible]


Sigma      =    0.50          Draft head =   6.30 m

Volume of the blade       0.95 dm3
```

THE PROGRAM PART THE STREAMROOM BETWEEN PHERIFERI AND HUB INTO 5 STREAMTUBES AND THE "RADIUS" IS THE MIDDLE OF EACH.

u = TANGENTIAL VELOCITY OF THE BLADE

VELOCITY-TRANGLE:



WHIRL-IN = C_{1u} * RADIUS

WHIRL-OT = C_{2u} * RADIUS

$T = \frac{2\pi * \text{RADIUS}}{Z1}$; Z1 = NOMBER OF BLADES

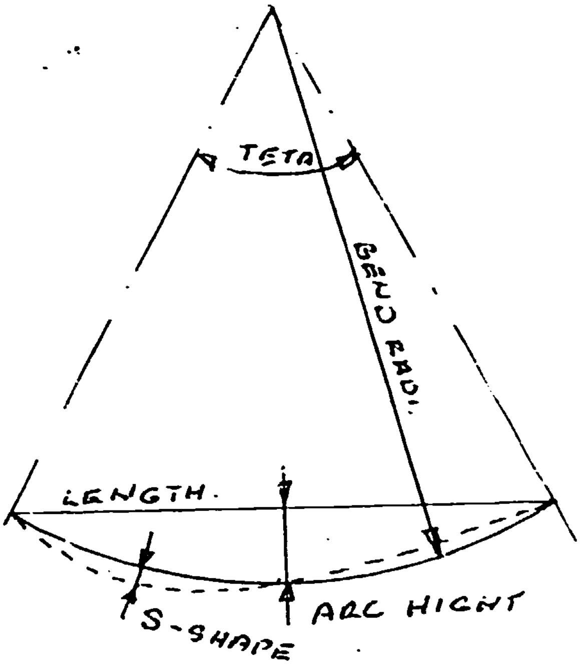
L = CHORD LENGTH AT THE RADIUS.

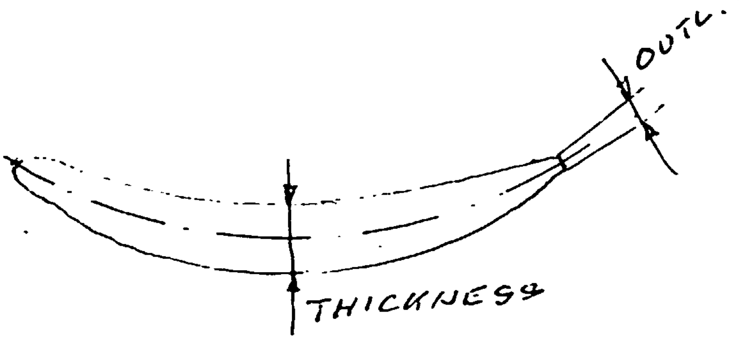
$T/L = f(n_{11}; \gamma)$ FROM STATISTICS

$My = f(T/L; \gamma)$ FROM STATISTICS

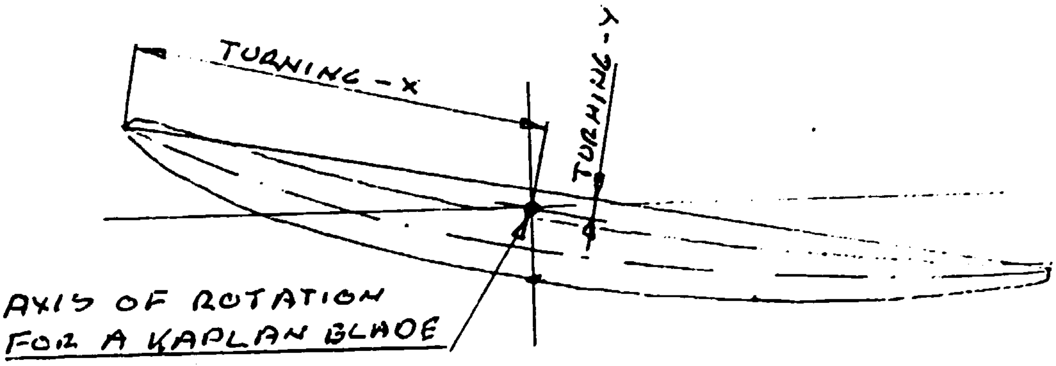
TETA $= f(My; \text{D-BETA})$

TETA = (ΔBETA — ANG. OF ATACK)/My





SLOP. ANG. CORRESPONDS TO GAMMA(γ) BUT THE BLADE IS TURNED SO THAT THE CHORD IS IN AN AXIAL PLAN AT THE PHERIFERI.



CALCULATION OF CYLINDER PROFILE

ENCL.5

Runner identif. Flavio - Evald
Runner diam. = 800 mm
Cylinder radius= 378 mm
Profile length = 456.5 mm
Arc height = 2.1 %
Thickness max = 2.28 %
Slope angle = 1.22 deg
Turning-x = 205 mm
Step angle = 5 deg
Inlet angle = 6.00 deg

S-shape = .22 %
Outl. thickness= .35 %

Turning-y = -7.9 mm

Outlet angle = -3.50 deg

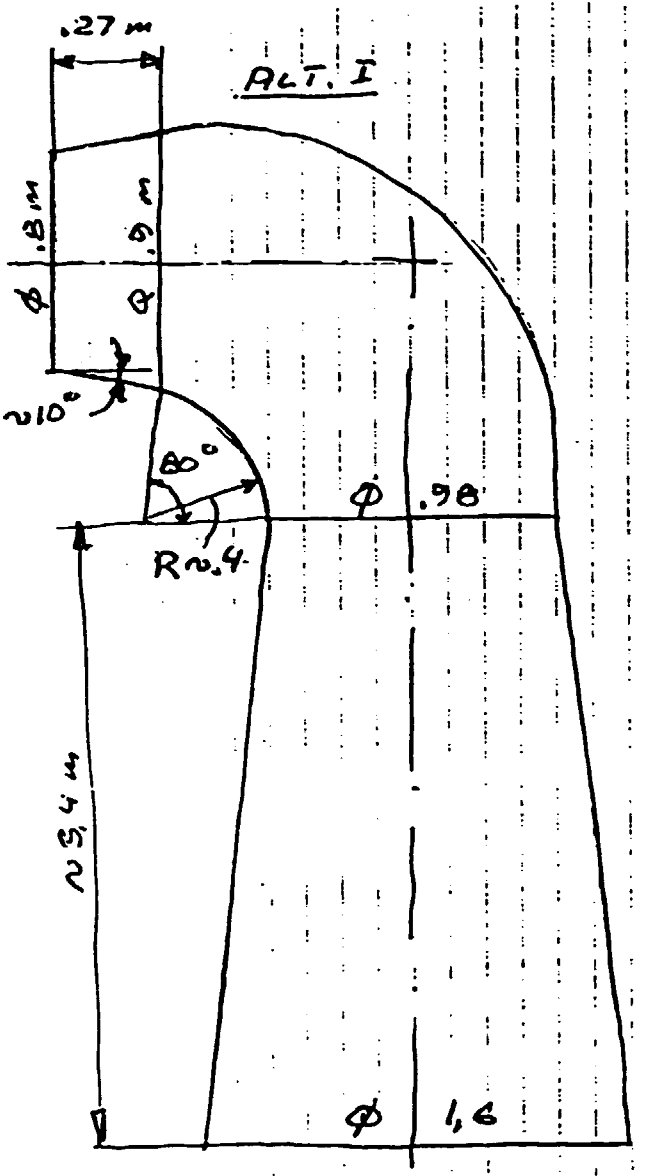
Cylindersection of the profile at the slope angle:

Step in mm = 32.99

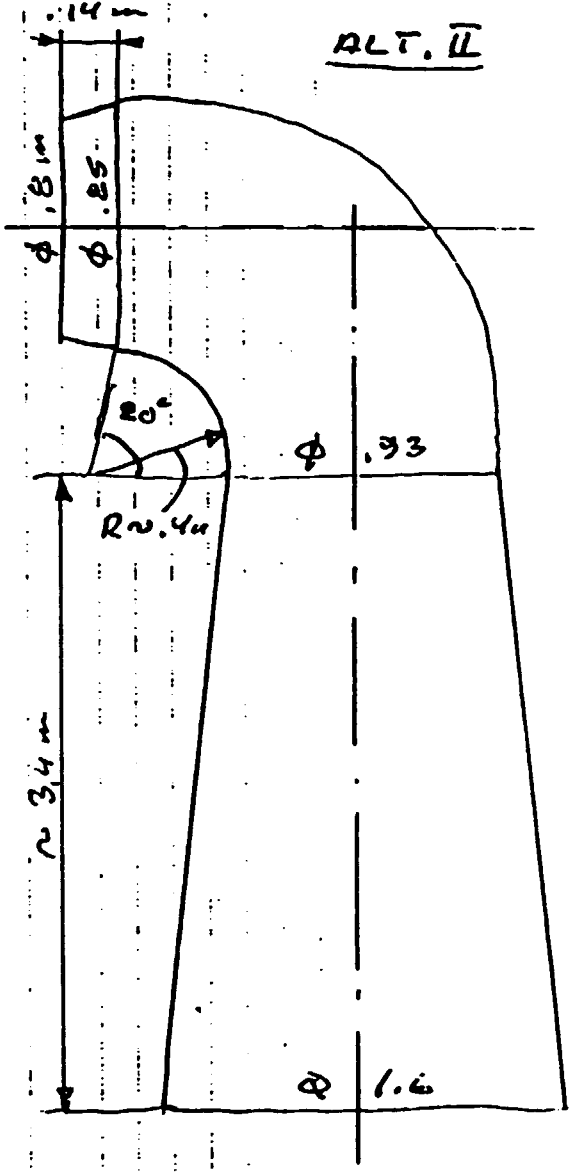
Step deg.	Arc-L mm	z-suct. mm	z-pres. mm	Thickn. mm	Cord-L mm	Cord R mm
-31.1	-205.1	4.7			-195.2	323.7
-31.1	-205.1		2.3	2.4	-195.2	323.7
-30.0	-197.9	6.0	2.9	3.0	-189.0	376.8
-25.0	-164.9	11.3	5.5	5.7	-159.8	377.2
-20.0	-131.9	15.5	7.8	7.6	-129.3	377.5
-15.0	-99.0	18.7	9.7	8.9	-97.8	377.7
-10.0	-66.0	20.9	11.2	9.7	-65.6	377.9
-5.0	-33.0	22.3	12.1	10.2	-32.9	378.0
0.0	0.0	23.0	12.6	10.4	0.0	378.0
5.0	33.0	23.0	12.8	10.2	32.9	378.0
10.0	66.0	22.6	12.8	9.7	65.6	377.9
15.0	99.0	21.6	12.9	8.7	97.8	377.7
20.0	131.9	20.4	13.0	7.4	129.3	377.5
25.0	164.9	18.9	13.1	5.9	159.8	377.2
30.0	197.9	17.3	13.0	4.3	189.0	376.8
35.0	230.9	15.4	12.8	2.6	216.8	376.4
38.1	251.3	14.1			233.2	297.5
38.1	251.3		12.5	1.6	233.2	297.5

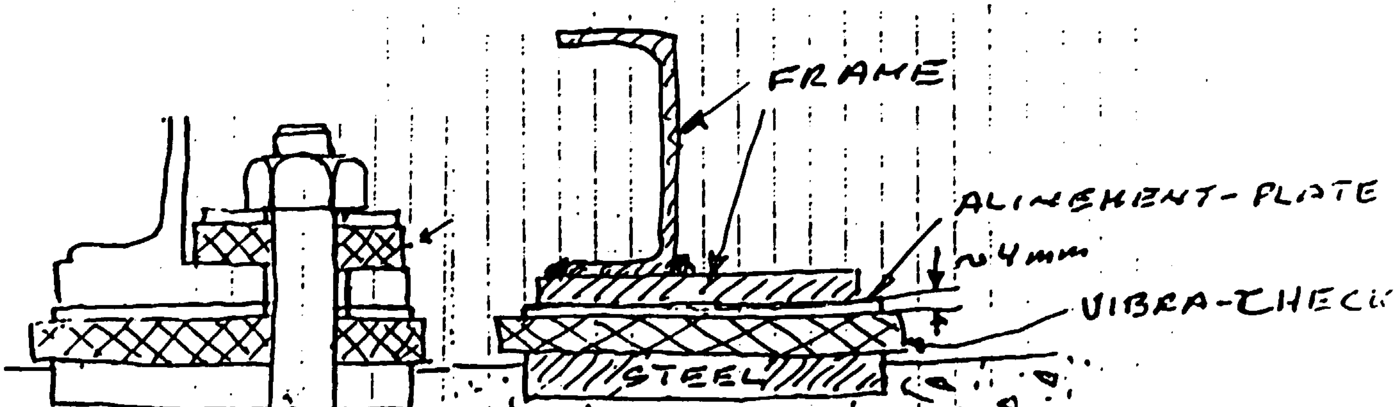
ENCL 6.

ALT. I

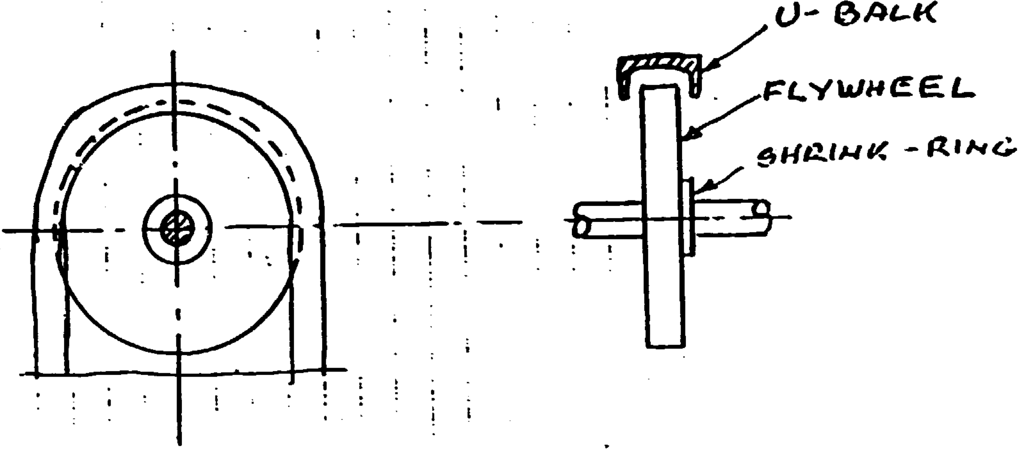


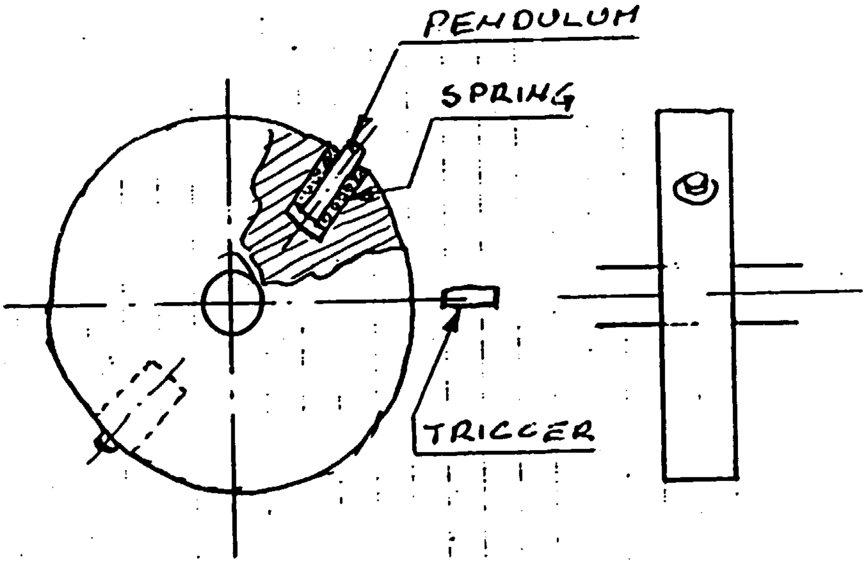
ALT. II





ENCL. 7

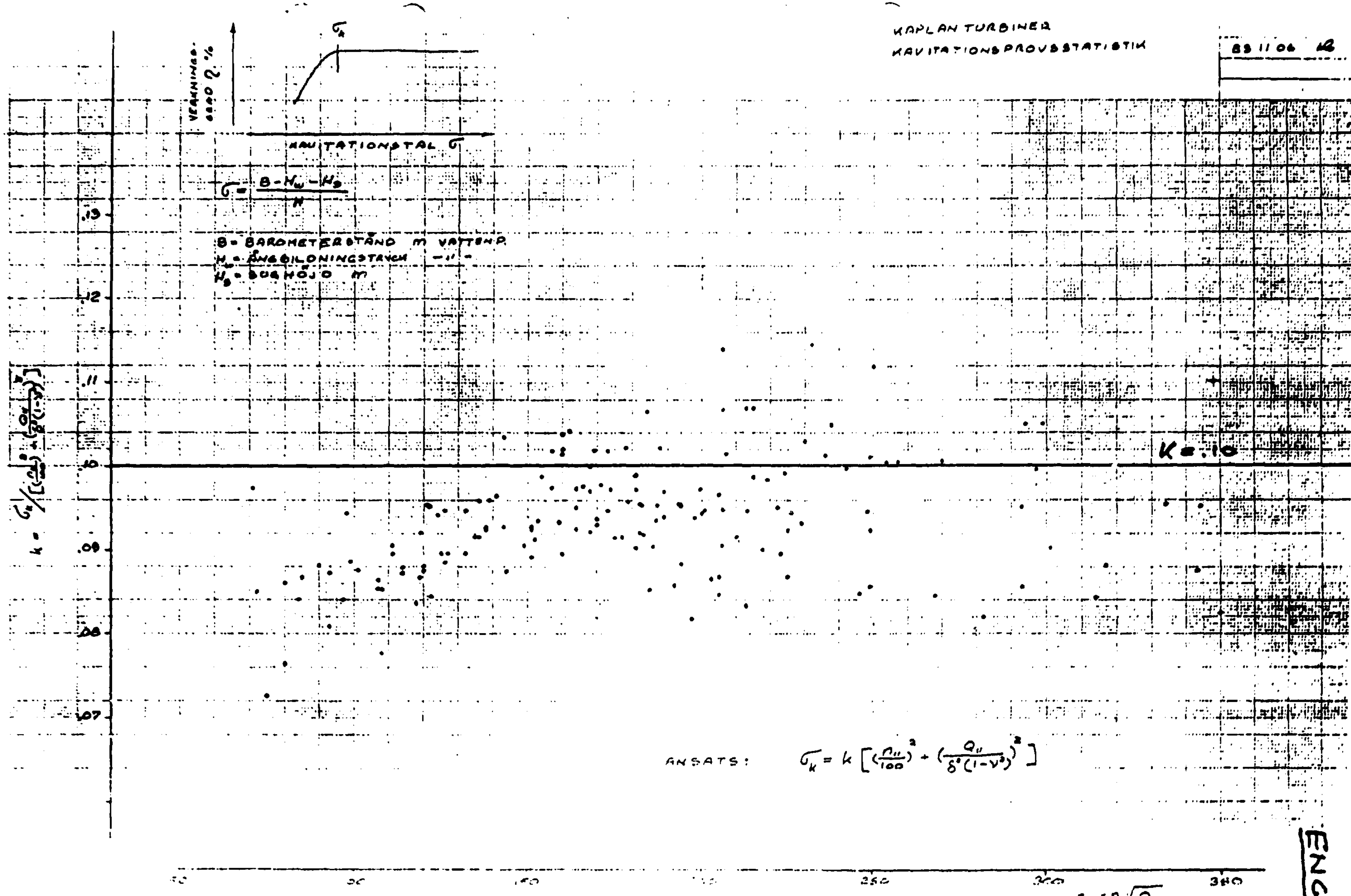




ENCL. 8

KAPLAN TURBINER
KAVITATIONSPROVSSTATISTIK

83 11 06 12



ANSATS: $\sigma_k = k \left[\left(\frac{n_{11}}{100}\right)^2 + \left(\frac{Q_{11}}{\delta^2(1-\nu^2)}\right)^2 \right]$

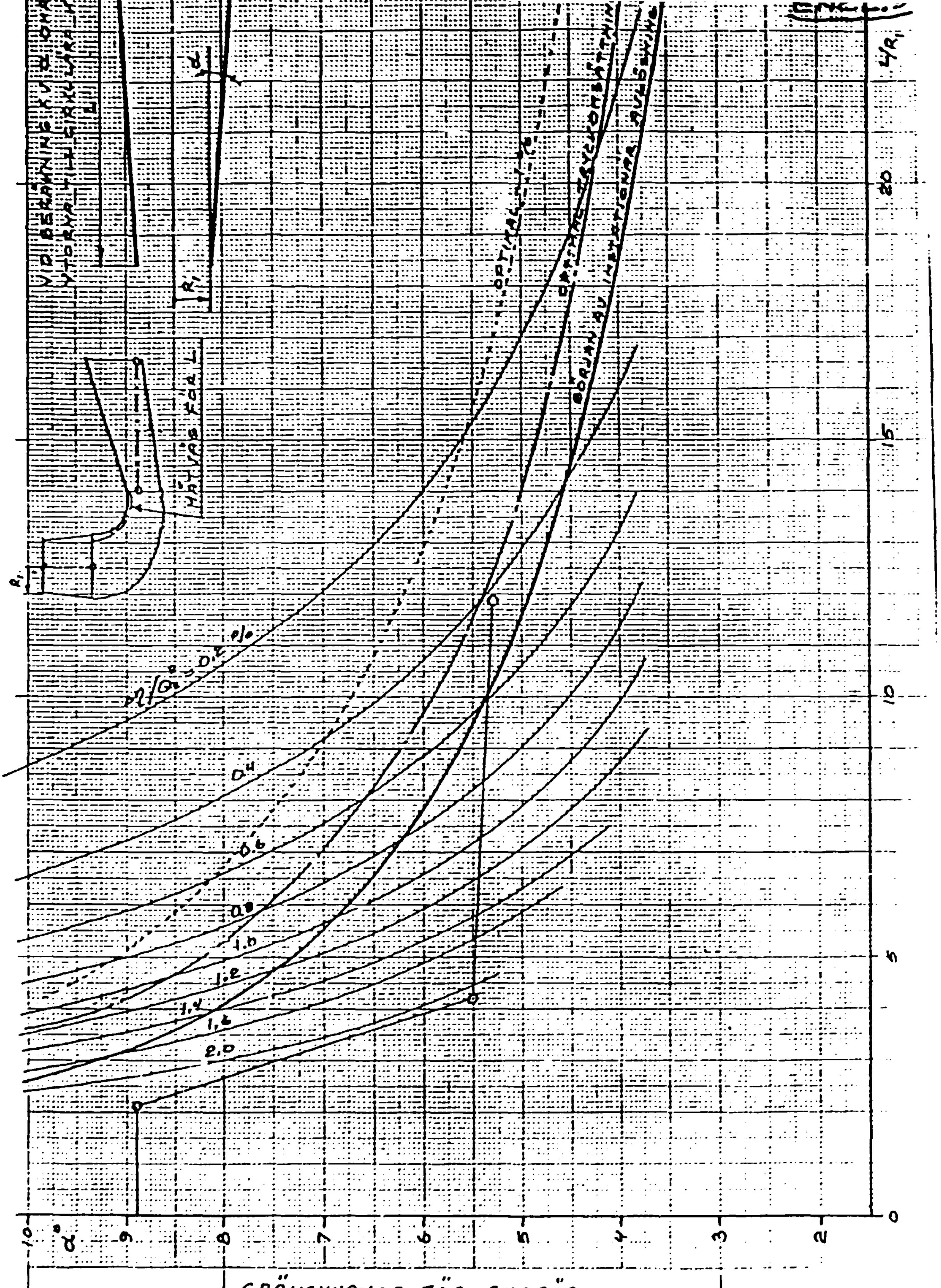
D = LÖPHJULSDIA. m
H = FALLHÖJD m
n = VARVTAL RPM

Q = VOLYMSTRÖM m^3/s
δ = STRYPNING = D_{STRYP}/D
ν = NAVFÖRHÅLLANDE = D_{NAV}/D

$n_{11} = \frac{n D}{\sqrt{H}}$

$Q_{11} = \frac{Q}{D^2\sqrt{H}}$

$n_S = 3.65\, n_q$ VID $\eta \approx 92$ %
KW/ m, RPM



GRÄNSKURVOR FÖR SUGRÖR

NOHAB
TROLLHÄTTAN
SWEDEN

ENCL. 10

TEST STAND FOR KAPLAN TURBINE MODELS

1.1 Description of the test stand

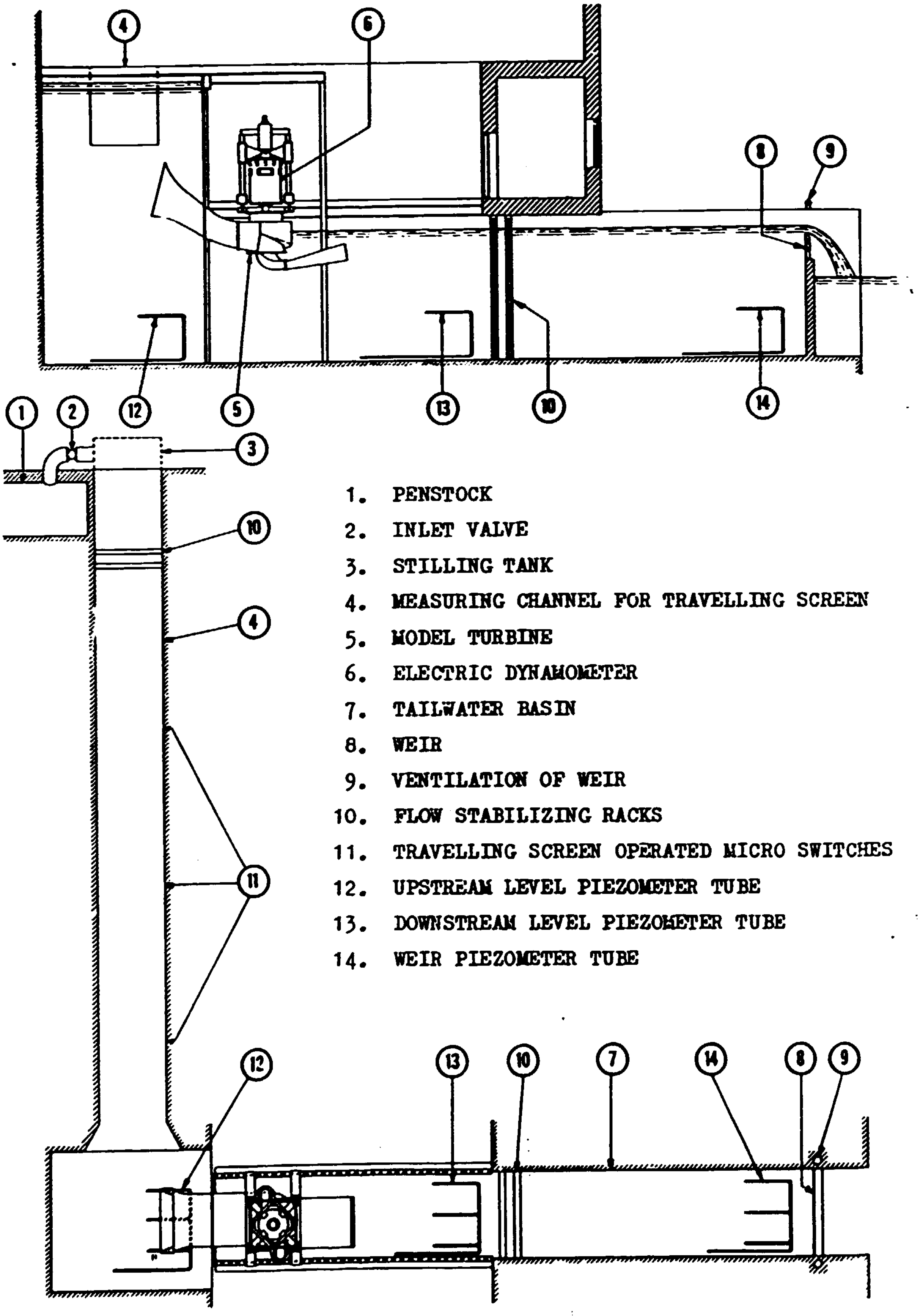
The stand is designed for testing models having a runner diameter of 500 - 600 mm, tested under cavitation free conditions ($\sigma \approx 2.5$). It is provided with long head and tail race channels having smooth parallel walls and horizontal bottom. The water is taken from the Göta River through a penstock under a 25 m head into a stilling tank, where practically all the energy is destroyed. From this chamber the water passes through a series of stabilizing racks so that the flow enters the head race channel with practically uniform velocity throughout the entire cross-section. The water flow is regulated by means of remote operated sluice valves.

After passing through the turbine, the irregularities in the flow are eliminated by means of a series of racks in the beginning of the tail race. The tail race terminates in a weir free from lateral contraction.

In this test stand various turbine characteristics are determined, such as output, discharge, runaway speed, differential pressure, axial thrust and wicket gate torque.

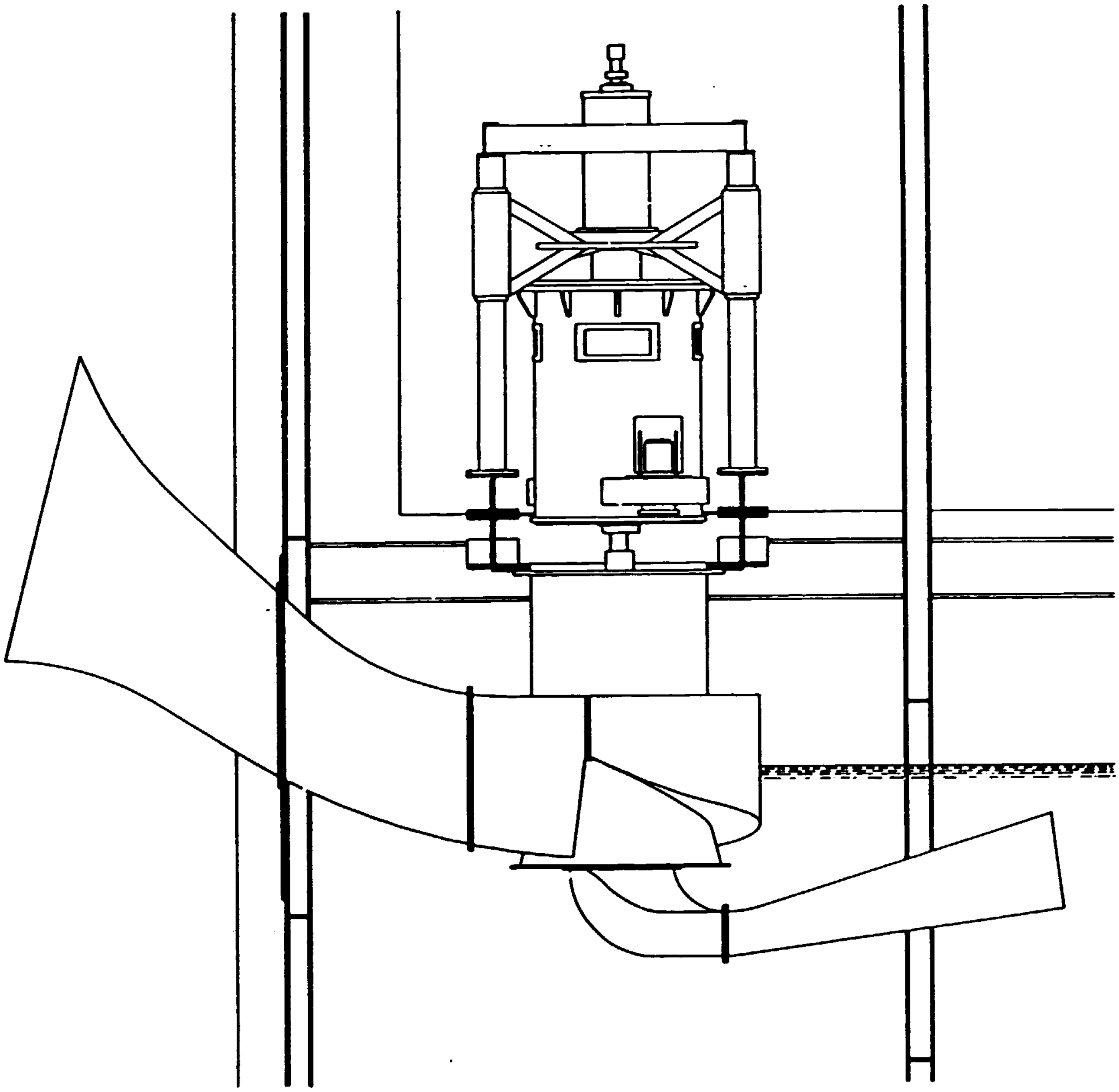
Underlaget för denna bilaga har vänligen tillhandahållits från NOHAB, Trollhättan.

1.1.11 General arrangement of the turbine test stand.

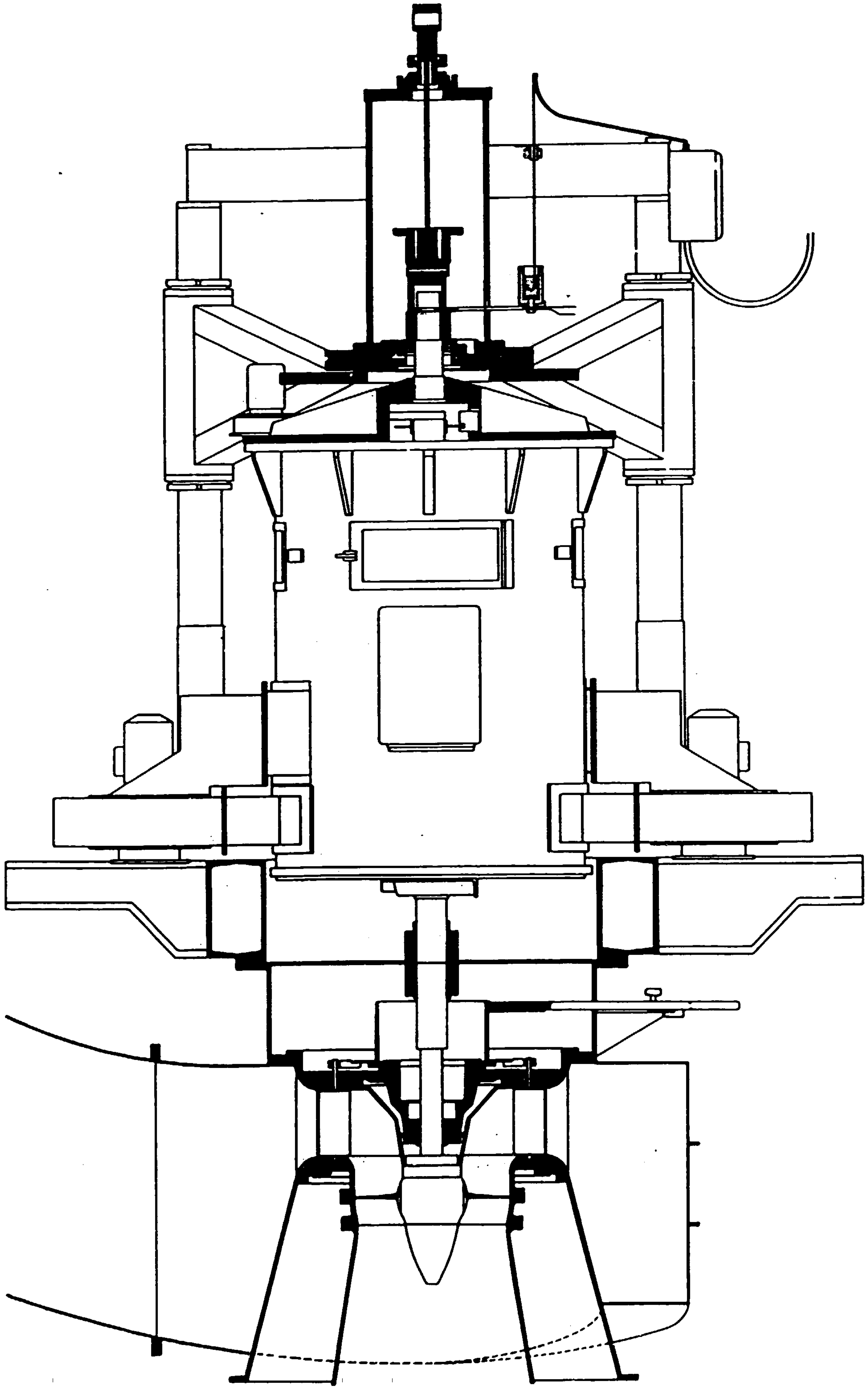


1. PENSTOCK
2. INLET VALVE
3. STILLING TANK
4. MEASURING CHANNEL FOR TRAVELLING SCREEN
5. MODEL TURBINE
6. ELECTRIC DYNAMOMETER
7. TAILWATER BASIN
8. WEIR
9. VENTILATION OF WEIR
10. FLOW STABILIZING RACKS
11. TRAVELLING SCREEN OPERATED MICRO SWITCHES
12. UPSTREAM LEVEL PIEZOMETER TUBE
13. DOWNSTREAM LEVEL PIEZOMETER TUBE
14. WEIR PIEZOMETER TUBE

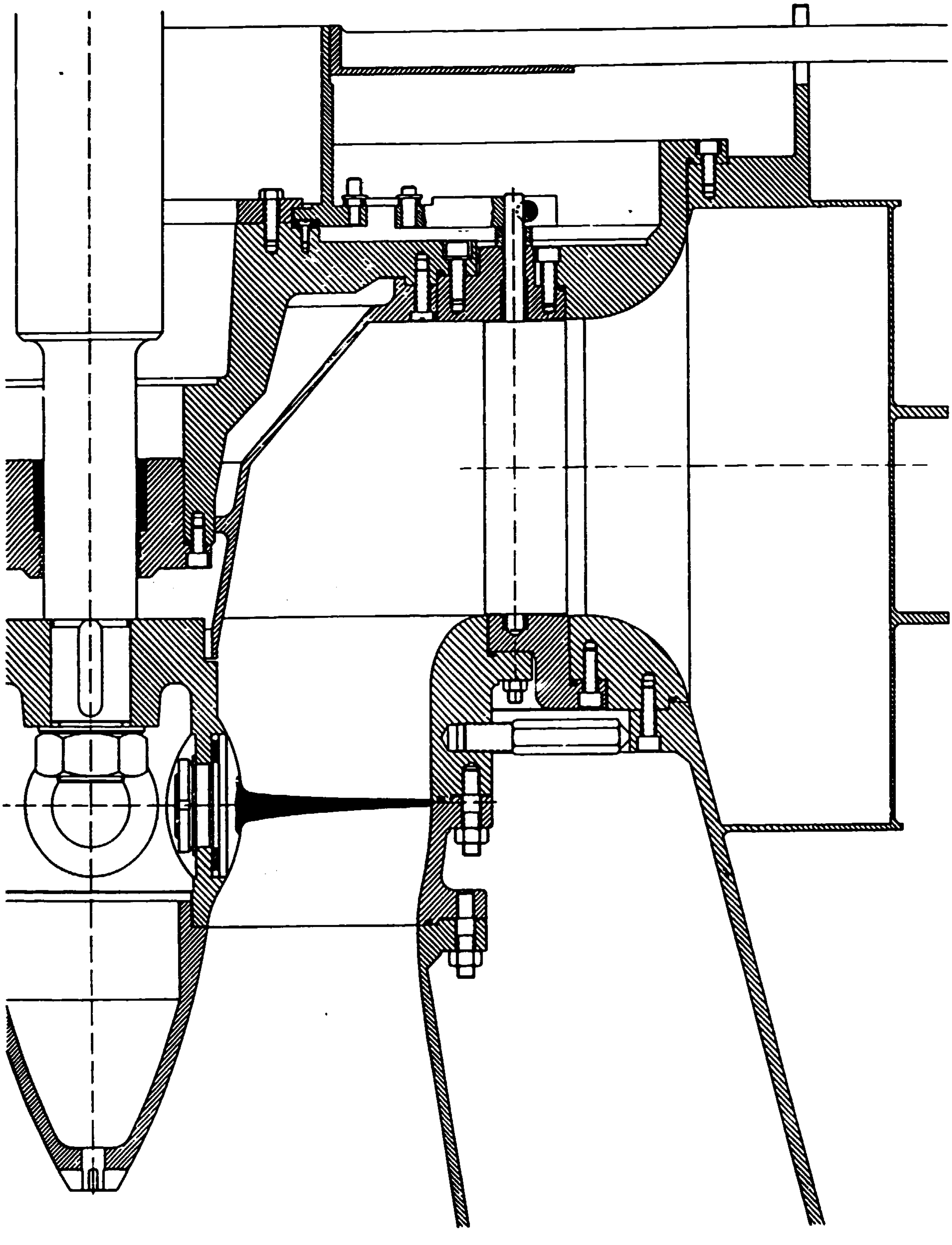
1.1.12 Model turbine cross section



1.1.13 Electric dynamometer arrangement



1.1.14 Runner and wicket gate arrangement



500 mm model kaplan runner of medium running type.

The model runner in question has been used in tests for Gardikfors Power Station in the north of Sweden owned by the Swedish State Power Board.

The prototype turbine has the following data:

Head = 39 m
Speed = 150 rpm
Output ≈ 60.6 MW
Runner dia = 5.0 m

1.2 Efficiency and output

In order to determine the efficiency of the model turbine it is necessary to measure the following main data in the test stand:

Discharge = Q m^3/sec.
Net head = H_n metres
Speed = n rpm
Output = P kW

Based on these data the turbine efficiency, i.e. the relation between the net output (braked output) and the available natural power is calculated as follows:

$$\eta = \frac{P}{P_a}$$

where

η = turbine efficiency
P = braked output in kW
P_a = natural power $\gamma \cdot Q \cdot H_n \cdot g$ in kW

$$\eta = \frac{P}{\gamma \cdot Q \cdot H_n \cdot g} \cdot 100 \text{ in } \%$$

where

Q = discharge in m^3/sec.
H_n = net head in metres
γ = specific density of water kg/m^3
g = gravity kg m/s^2

1.2.1 Discharge

The discharge is measured by means of a weir as well as a travelling screen.

1.2.11 Weir

The weir is arranged in strict accordance with IEC Recommendation, Publication 41 and "The Swiss Rules for Hydraulic Turbines", SEI Publication No. 178 e. The head (h) above the crest of the weir is measured in a float well connected to the channel by means of three piezometer tubes. The position of the float is determined using a scale and an accurate level indicator.

Zero adjustment and checking of the level indicator are carried out in accordance with figure 1. The valve A in the pipe running between the float well and the piezometer tubes is closed. An easily mounted, pointed measuring rod with container, the design of which will be found in figure 2, is arranged above the weir crest. The valve B is opened and the float well filled until the water surface almost reaches the upper edge of the pointed measuring rod. The distance Δ between the weir crest and the lower end of the pointed measuring rod is measured after the position of the rod has been adjusted until the point of the rod exactly touches the water surface. By adding the distance Δ to the length of the rod L, a distance h from the water surface to the weir crest is calculated. The position of the level indicator is than adjusted so that the calculated distance h is read on the vernier scale.

By placing the measuring rod with container on different elevations in relation to the weir crest it is possible to check the entire measuring range.

The discharge Q is calculated according to the following general formula:

$$Q = \frac{2}{3}\mu \cdot b \cdot h \cdot \sqrt{2gh}$$

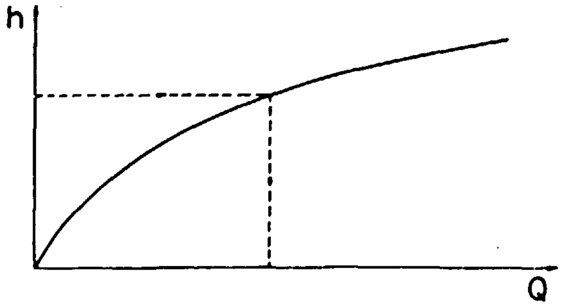
where the coefficient μ is calculated in accordance with "The swiss society of Engineers and Architects" (SIA).

$$\mu = 0.615 \cdot \left(1 + \frac{1}{h + 1.6}\right) \left[1 + 0.5 \left(\frac{h}{h + s}\right)^2\right]$$

The following data are valid for the weir in question:

$b = 2799$ mm
$s = 3910$ mm

In order to facilitate the evaluation of the test results, the equation of the weir has been plotted in a graph having the weir head h mm and the discharge Q m^3/s as coordinates. See figure below.



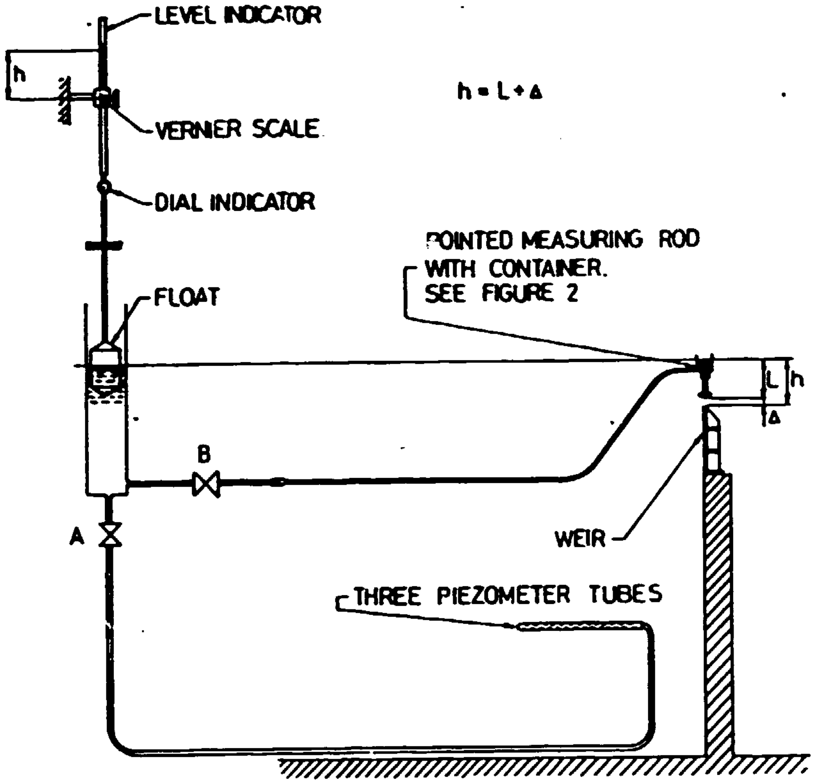

LEVEL INDICATOR
h
VERNIER SCALE
DIAL INDICATOR
h = L + Δ
POINTED MEASURING ROD
WITH CONTAINER.
SEE FIGURE 2
FLOAT
L
h
Δ
B
A
WEIR
THREE PIEZOMETER TUBES

Fig.1.

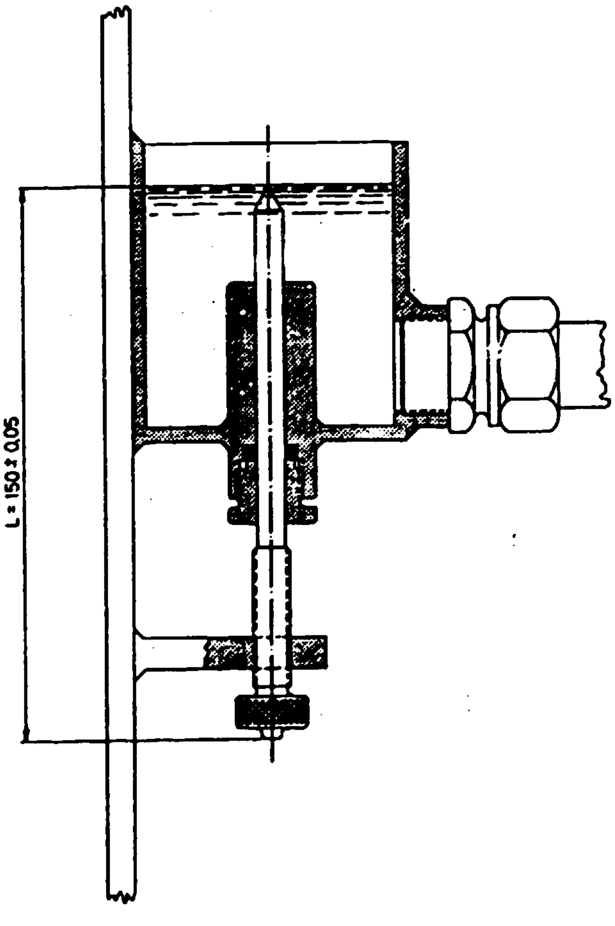

L = 150 ± 0.05

Fig.2.

1.2.12 Travelling screen

The sectorshaped screen is pivoted on a carriage moving on rails. The screen fits in the measuring channel, with a small clearances at the sides and the bottom so as to permit the screen to move with the water flow. Thus, the speed of the carriage will be very near the same as the mean velocity of the water flow. During a test the movement of the screen is timed by micro switches arranged on the edge of the channel and actuated by an arm on the carriage. The first micro switch is placed in teh beginning of the channel and far enough from the end of it to allow the screen to adapt itself to the velocity of the water flow before the timing is started. Stop switches are placed at 5 and 10 metres respectively from the first switch. As the measuring length either 5.000 or 10.000 m being used.

The discharge is determined by dividing the water volume of the measuring length by the measuring time (t). The measuring volume is the product of measuring length (L), depth of water (h), channel width (B) and a calibration coefficient (k_c), derived from a discharge calibration in a volumetric tank. Figure 3.

$$Q_s = \frac{L \cdot h \cdot B}{t} \quad m^3/s$$

where L, h and B are in metres and t in sec.
Mean value of B is 2.206 m.

$$Q = \frac{Q_s}{k_c} \quad m^3/s$$

The depth of water in the channel is measured at 3 double measuring points, equipped with piezometer taps, at the beginning of the screen measuring length and after 5 and 10 metres respectively. The mean value of the piezometer readings is used as the depth of water when calculating the discharge. At its starting position the screen is folded up on the carriage and is kept in this position by a chain. When the chain drum is released, the chain is loosened and the screen slides into the water starting its movement softly and accelerating to the mean velocity of the water without setting up ripples or waves.

After a discharge measurement has been carried out the screen is automatically brought back to its starting position by means of a motor driven winder and the chain. At the end of the channel a switch for starting the winder motor is actuated by the carriage

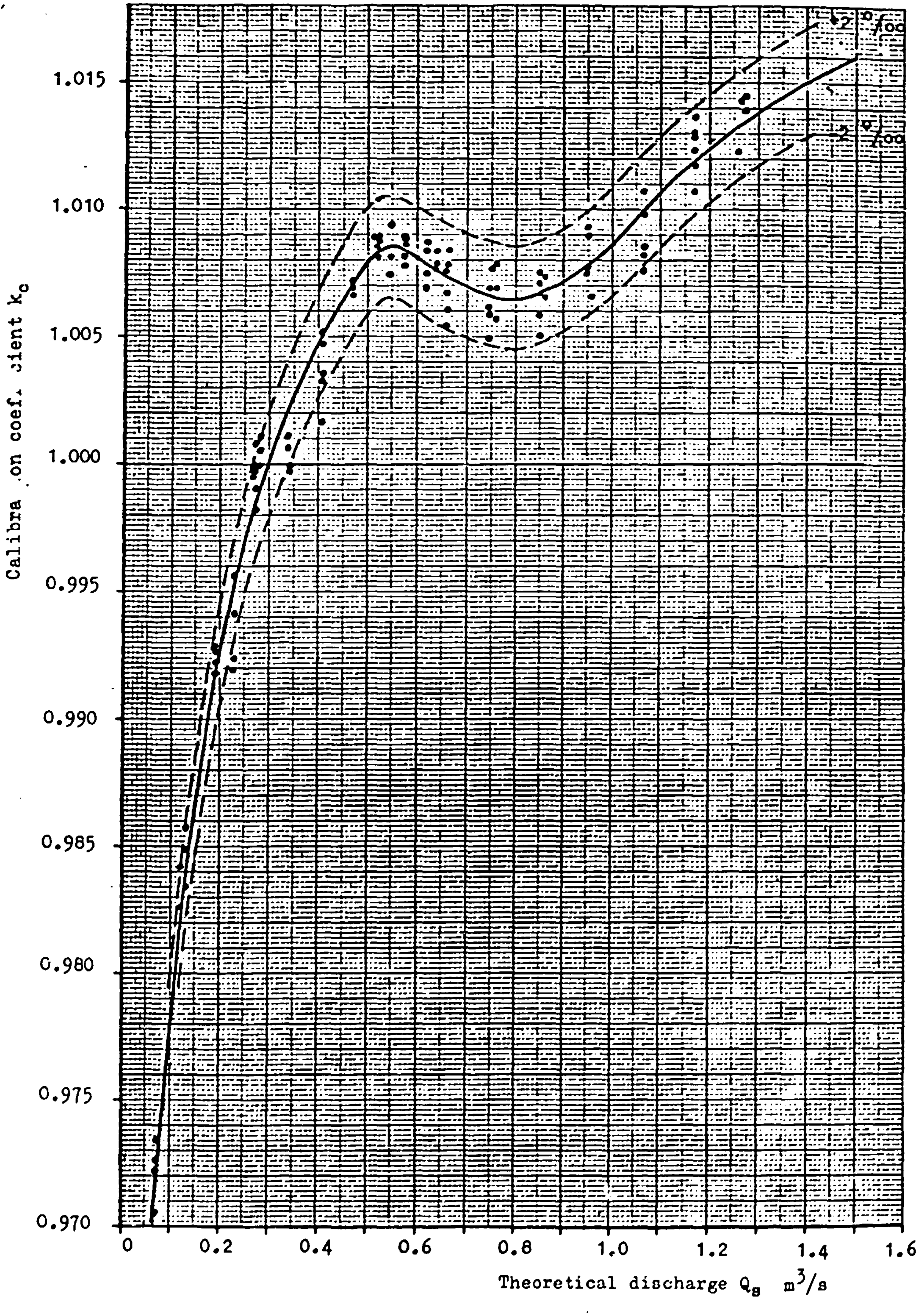


Figure 3. Travelling screen calibration

and the chain is wound up on the drum. Due to the resistance of the screen in the water and to the fastening point of the chain the screen is raised around its axis on the carriage. When the screen has reached its final position on the carriage, the carriages is pulled back to its starting position. Just before the starting position is reached the motor of the winder is stopped by another electrical switch. In the starting position the carriage is suspended by the chain with the screen folded up ready for a new travel (see photographs).

1.2.2. Head

The head is measured with the aid of a float on the tail water level. A graduated scale is vertically suspended on a flexible steel tape which runs over sheaves and is attached in one end to the float and in the other to a counter weight thus following the movement of the float.

Water level glasses from the turbine flume as well as from the piezometer taps at the spiral case inlet are placed immediately alongside the suspended scale. The position of the scale is so arranged that the distance (H) from the water level in the glasses to the tail water level can be read directly on the scale. Thus if the scale is read at the water level in the glass from the flume, the gross head (H_b) is obtained. This is valid only on the assumption that the velocity heads in the flume and the tail race are so small compared to the head that they can be neglected. The net head, defined as the head difference between the energy level in spiral case inlet (section E-E) and the energy level in the draft tube outlet (section F-F), is measured and calculated as follows. The draft tube outlet piezometer taps are connected to the tail water float well. The tail water scale is read at the level in a water glass connected to the case inlet piezometer taps. To this reading is added the difference in velocity head between the inlet and outlet section: $v_o^2/2g - v_4^2/2g$. The velocities v_o and v_4 refer to the mean velocity in the question. The net head is thus $H_{net} = H_{stat} + (v_o^2/2g - v_4^2/2g)$. This is also the net head definition in the IEC-publications No. 41 and No. 193.

Two pointed measuring rods with containers arranged accordance with figure 2 are used for checking the scale adjustment.

1.2.3. Speed

The speed (n) is electronically measured. To the shaft is attached a disc having 60 radial slots. When the slotted periphery of the disc rotates cutting a light beam which actuates a photo cell thus giving rise to 60 pulses for each revolution of the shaft. The pulses are registered by a decade counter. The time during which the pulses are counted is fixed to 10 seconds with an accuracy of ± 1/10.000 sec. by means of a 10 kc crystal oscillator.

It is thus possible to read the rpm directly on the decade counter with an accuracy close to ± 1/10 rpm.

1.2.4 Output

1.2.41 Brake force measured by means of a scale

The output is determined with an electric dynamometer. The force (F) in the brake arm is transmitted to a scale by means of a lever. The output P is determined as follows:
Prior to the test the bearing friction force ΔF is measured within the foreseen test range. This force is for each test point added to the scale reading F. Scale readings are electronically taken four times per second. The number of readings (X) which are taken during each test as well as the sum (S) of the readings are registered by separate decade counters. The means scale load during the test is thus:

$$F = \frac{\text{sum of readings}}{\text{number of readings}} = \frac{S}{X}$$

thus

$$F_{tot} = F + \Delta F$$

$$P = M \,.\, \omega \text{ kW}$$

$$M = F_{tot} \,.\, a \,.\, \frac{g}{g_0} = \text{brake torque, kpm}$$

$$\omega = \frac{\pi \,.\, n}{30} \quad \text{angular velocity, 1/s}$$

$$a = 0.974 \text{ m} \quad \text{brake arm length, m}$$

Figure 4 shows schematically how the checking is carried out. The valves B and D are opened whereby the water rises in the measuring rod containers. Thereafter the valves A and C are closed and the pointed measuring rods adjusted so that their points exactly touch the water surface. The distance between the lower ends of the measuring rods will then exactly correspond to the head indicated on the suspended scale provided the length L of the measuring rods is the same for both.

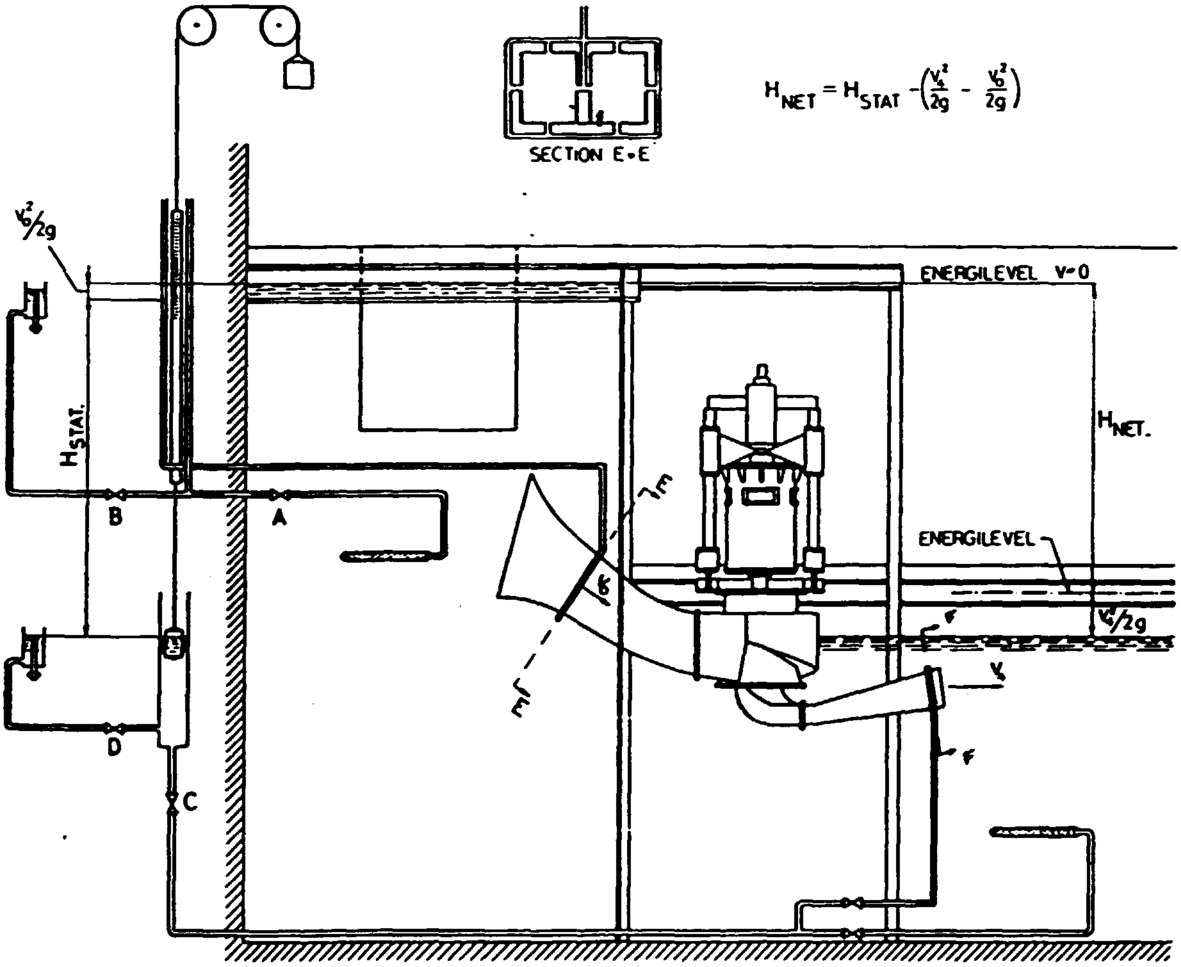


Figure 4

as $\omega = \frac{\pi \cdot n}{30}$ where n = rpm follows

$$P = \frac{F_{tot} \cdot a \cdot \pi \cdot n \cdot g}{30 \cdot g_o} \text{ or } P = k_P \cdot F_{tot} \cdot n$$

where

$$k_P = \text{brake coefficient} = \frac{a \cdot \pi \cdot 9.817}{30 \cdot 9.807}$$

with

$$a = 0.974 \text{ follows } k_P = 0.001000 \text{ and finally}$$

$$P = 0.001000 \cdot F_{tot} \cdot n \quad kW$$

1.2.5 Efficiency test method and evaluation of the results

The values of speed, output and discharge obtained at the tests are converted into the corresponding values ("unit values") for a turbine fully homologous with the tested unit but having a runner diameter of one meter and operating under a net head of one meter.

All diagrams are made up on "unit values".

The test efficiency obtained is plotted in the diagram unchanged. When stepping up the efficiency from the model to the prototype, the calculations are based on the actual model runner diameter and the actual head under which the test was carried out.

The "unit values" are designated:

Unit speed = n_{11} rpm

Unit output = P_{11} kW

Unit discharge = Q_{11} m^3/sec.

Conversion is carried out as follows: (Model runner dia D = 0.5 m)

$$n_{11} = \frac{n \cdot D}{\sqrt{H_n}} = \frac{n \cdot 0.5}{\sqrt{H_n}}$$

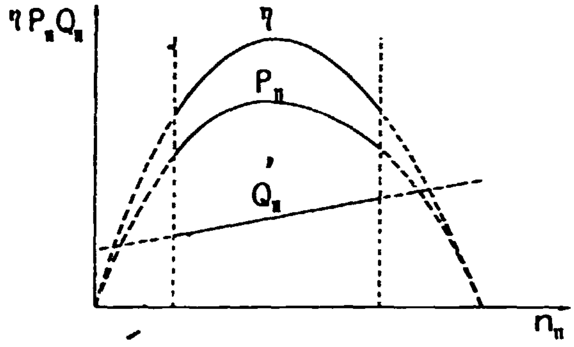
$$P_{11} = \frac{P}{D^2 \cdot H_n \sqrt{H_n}} = \frac{P}{0.25 \cdot H_n \sqrt{H_n}}$$

$$Q_{11} = \frac{Q}{D^2 \cdot \sqrt{H_n}} = \frac{Q}{0.25 \cdot \sqrt{H_n}}$$

The tests are carried out as follows:

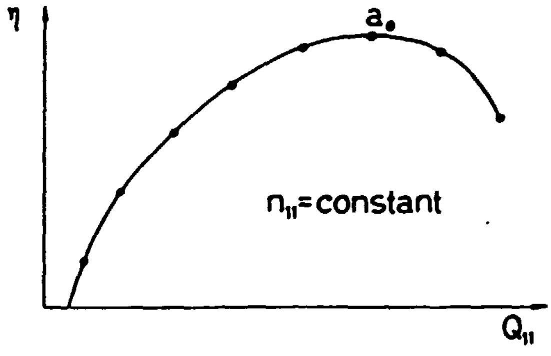
The turbine is operated within desired speed range with constant wicket gate opening and under practically constant head. (The head is somewhat reduced at increasing discharge due to the fixed weir.)

The figures obtained at this test are converted to unit values and plotted into a diagram as function of n_{11}:

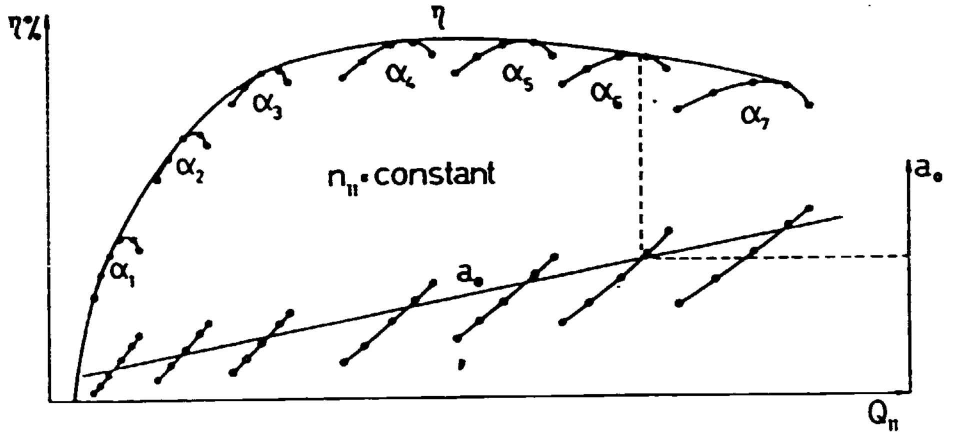


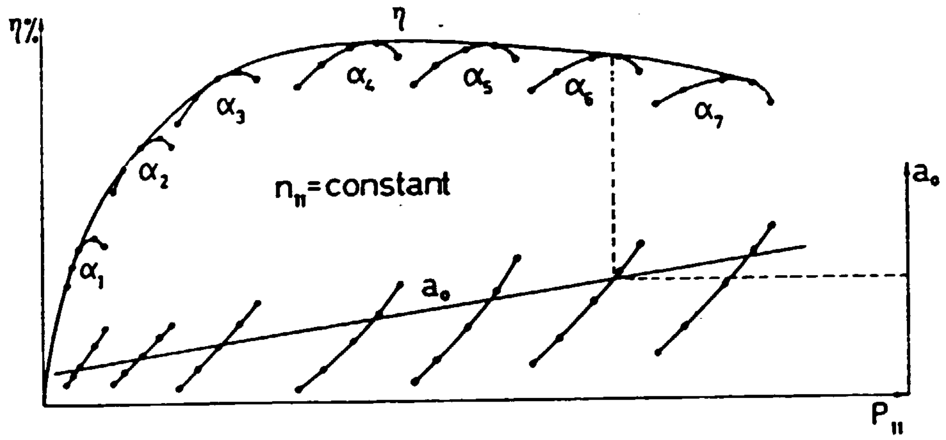
For each constant runner blade angle α the tests are repeated for a great number of constant wicket gate openings from closed to fully open position.

From the diagram thus obtained are taken the values of η and Q_{11} for a number of n_{11} which are spread over the entire speed range. A diagram showing efficiency v.s. unit discharge at constant runner blade angle and constant unit speed is drawn up.



These curves, one for each blade angle, are plotted together at a selected number of unit speed. A number of diagrams as shown below is obtained. If desired it is also possible to run the turbine at a constant speed and thus get the efficiency discharge relationship directly at any chosen speed. See diagram below.





The same type of diagram but having output P kW, instead of discharge Q, as the abscissa is also plotted.

Normally the following 7 runner blade angles are tested:

α_1 = minimum blade angle ($\alpha \simeq -2^\circ \rightarrow +5^\circ$)

$\alpha_2 = 7{,}5^\circ$

$\alpha_3 = 12{,}5^\circ$

$\alpha_4 = 17{,}5^\circ$

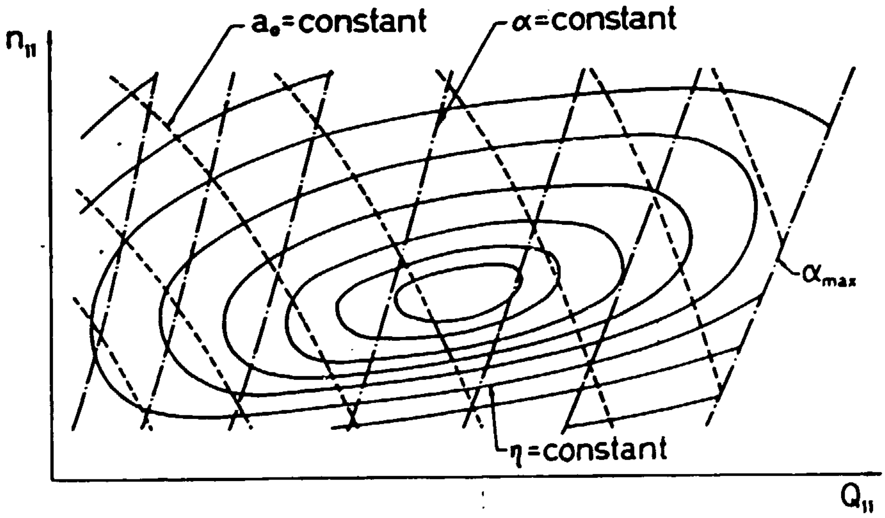
$\alpha_5 = 22{,}5^\circ$

$\alpha_6 = 27{,}5^\circ$

α_7 = maximum blade angle ($\alpha \simeq 32^\circ \rightarrow 33^\circ$)

$\alpha = 0^\circ$ indicates that the inlet and outlet corners at the outer periphery of the blades are in a plane perpendicular to the turbine shaft.

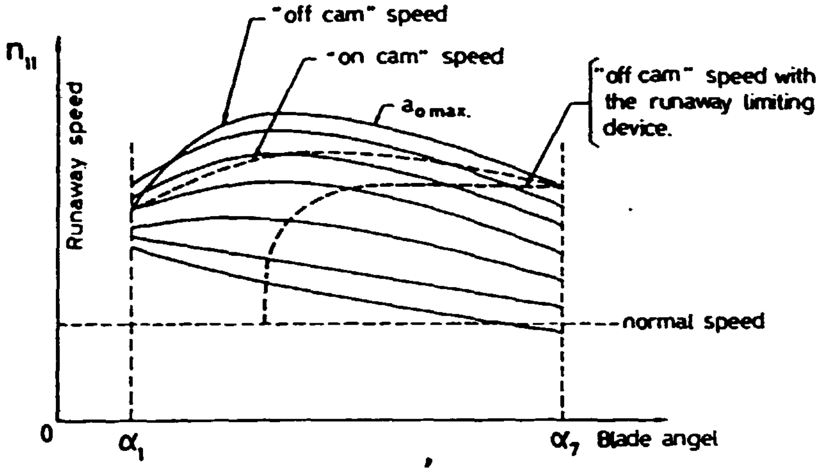
A turbine performance diagram - unit speed v.s. unit output - is drawn up on the basis of the envelope curve for constant n_{11}. One axis representing n_{11}, the other Q_{11} and the efficiency as the parameter.



The unit speed v.s. unit discharge - or output- diagram are supplemented with curves for constant α and a_0. These curves are obtained as follows:

1.3.1 Runaway speed

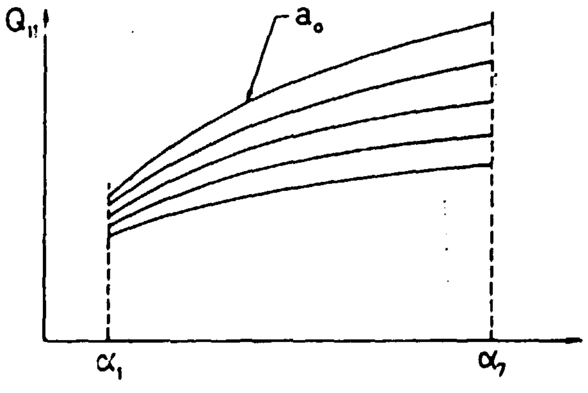
With the aid of these diagrams an assembly diagram of the runaway speed is made up as follows with $n_{11\ \text{runaway}}$ and α along the axis and a_o as the parameter:



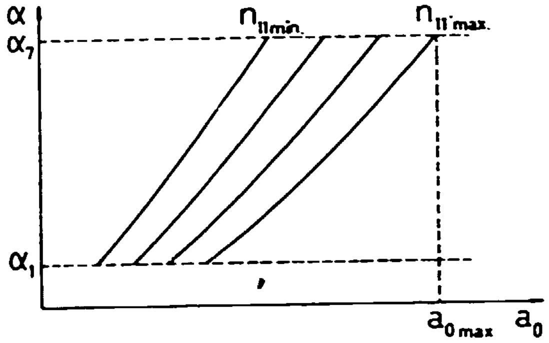
This diagram can be supplemented with curves for a_o, valid for optimum combination for the n_{11} corresponding to the absolutely highest head for a certain plant. In this way it is possible to obtain directly from the diagram the maximum theoretical runaway speed, i.e. "off cam" speed as well as "on cam" speed and maximum practical runaway speed with NOHAB's patented runaway speed limiting device.

1.3.2 Discharge at runaway speed

In analogy with the procedure for runaway speed an assembly diagram over the discharge at runaway speed is made up with $Q_{11\ \text{runaway}}$ and α along the axes and a_o as the parameter:

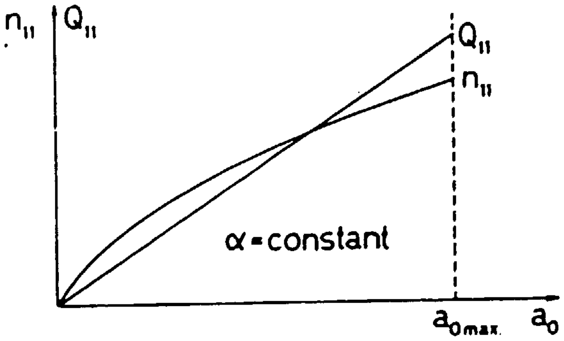


The envelope curve is transmitted to a curve on which the gate opening is a function of the discharge or of the output. It will be possible to read the combination between the gate opening and the runner blade angle resulting in optimum efficiency. From these combination curves is obtained a combination diagram giving the shape of the combination cam.



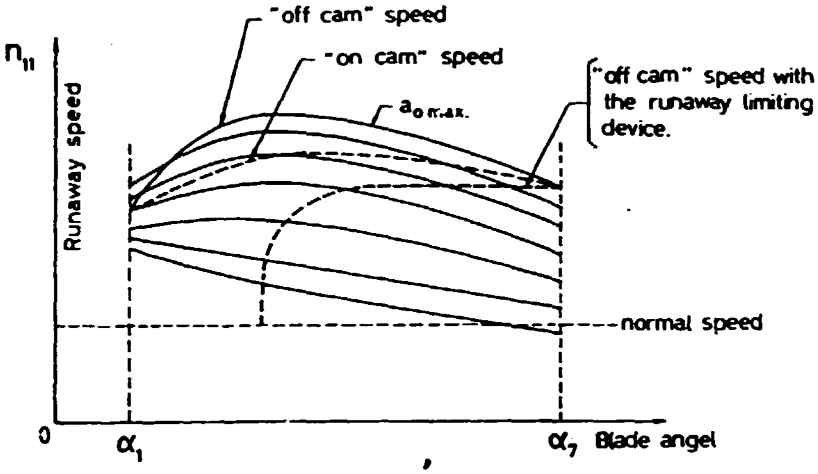
1.3 Runaway speed test

Runaway speed tests are carried out with the turbine shaft completely free from all brake action with the exception of such due to bearing friction and windage losses. Speed, discharge and head are measured at a number of constant runner blade angles (α) and gate openings (a_o). The test data are converted to unit values. For each blade angle the following diagram is made up.



1.3.1 Runaway speed

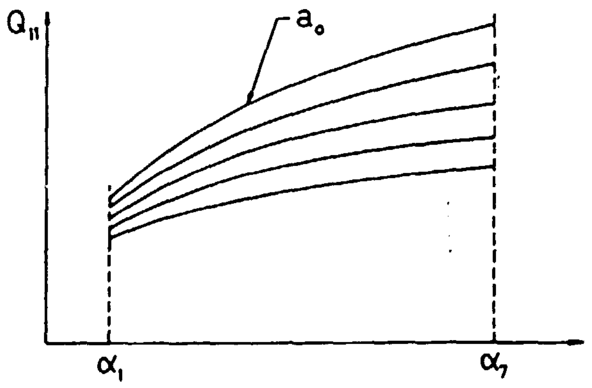
With the aid of these diagrams an assembly diagram of the runaway speed is made up as follows with $n_{11\ \mathrm{runaway}}$ and α along the axis and a_o as the parameter:



This diagram can be supplemented with curves for a_o, valid for optimum combination for the n_{11} corresponding to the absolutely highest head for a certain plant. In this way it is possible to obtain directly from the diagram the maximum theoretical runaway speed, i.e. "off cam" speed as well as "on cam" speed and maximum practical runaway speed with NOHAB's patented runaway speed limiting device.

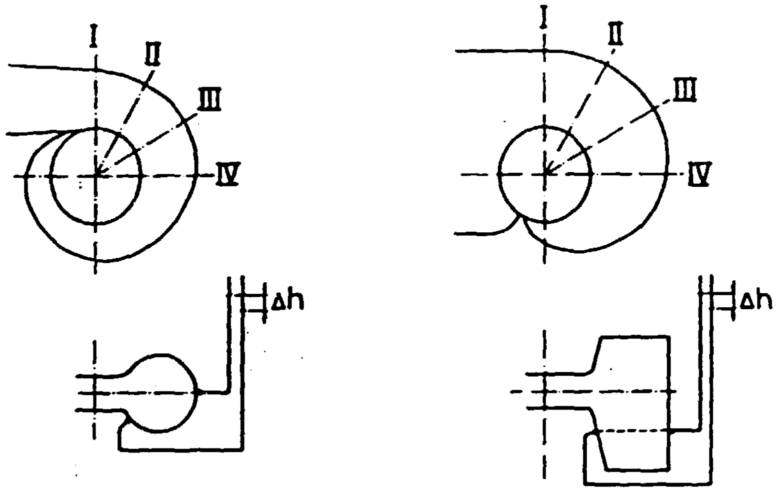
1.3.2 Discharge at runaway speed

In analogy with the procedure for runaway speed an assembly diagram over the discharge at runaway speed is made up with $Q_{11\ \mathrm{runaway}}$ and α along the axes and a_o as the parameter:



1.6 Differential pressure

At differential pressure measurements investigations are carried out on how the pressure difference Δh between two points of measurement on the spiral casing varies with the turbine discharge (Q). The points of measurement are located on the same radial cross section at the spiral casing but at different distances from the turbine centerline. In order to obtain as correct a picture as possible of the flow pattern, in the laboratory, tests measurements are taken at several cross sections. The figure below shows the schematic location of the points of measurements on a turbine having steel plate spiral casing as well as on one designed with concrete semi spiral casing. The location of the pressure taps corresponds here practically to the location recommended by the Winter-Kennedy method.



On a panel are arranged a number of glass tubes which are connected to the pressure taps and on which the pressure readings are taken. In order to obtain a reference level, two glass tubes on the panel are connected to the flume. In this way the reference level and the head water level will be indicated. Simultaneously with the differential pressure readings (Δh) in the various cross sections are taken readings of head (H) and turbine discharge (Q).

In order to determine $Q = f(\Delta h)$

let

$$Q = K(\Delta h)^n$$

where

Q = discharge $m^3/sec.$

K = seeked factor

h = differential pressure mm water column

n = seeked exponent

After transmitting to logarithmic form is obtained:

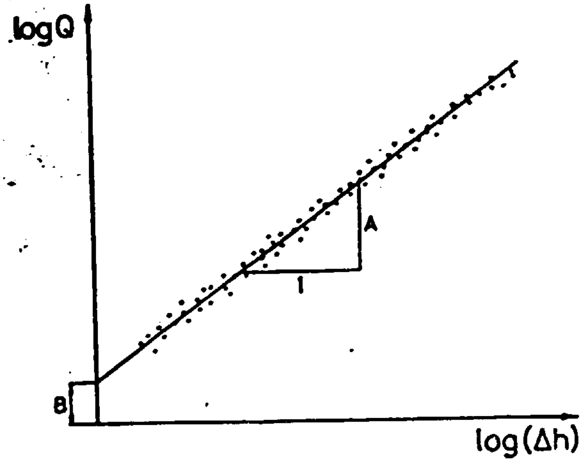
$$\log Q = n \log(\Delta h) + \log K$$

This equation signifies a straight line in accordance with

$$y = Ax + B$$

The coefficients A and B are determined graphically.

The diagram is found below:



ENCL. 11

SVENSK STANDARD

SIS 14 57 16

MNC METALLNORMCENTRALEN

Utgåva 1 Sida 1 (2)

Första giltighetsdag 1977 - 07 - 01

Koppargjutlegering
ALUMINIUMBRONS 57 16
Typ CuAl10Fe5Ni5

Cast copper alloy (aluminium bronze)

UDK 669 35

Aluminiumbrons enligt denna standard overensstammer med alloy CuAl10Fe5Ni5 enligt det internationella standardforslaget ISO DIS 1338, med undantag enligt tabell 1 under Kompletterande upplysningar.

Standarden motsvarar ovriga nordiska standarder enligt tabell 2 under Kompletterande upplysningar

I standarden anvands hållfasthetsbeteckningar enligt SIS 01 66 02 och mattenhetsbeteckningar enligt standarder fortecknade i SIS 01 61 11.

1 N/mm² = 1 MPa

Oversikt over svensk standard for koppargjutlegeringar se MNC 51

Leveransformer Tackor, sandgjutgods, kokillgjutgods, centrifugalgjutgods, stranggjutgods

Materialfordringar

Sammansättning

Leveransform		Al %	Cr %	Cu[1] %	Fe %	Mn %	Ni[1] %	Pb[2] %	Si %	Sn %	Zn %	Mg %
Tackor	Nom	10	–	80	5	–	5	–	–	–	–	–
	Min	9,0	–	77,0	3,5	–	4,5	–	–	–	–	–
	Max	10,0	0,005	82,0	5,3	2,3	6,3	0,02	0,07	0,10	0,20	0,04
Gjutgods	Nom	10	–	80	5	–	5	–	–	–	–	–
	Min	8,8	–	77,0	3,5	–	4,5	–	–	–	–	–
	Max	10,0	0,005	82,0	5,5	2,5	6,5	0,05	0,10	0,20	0,20	0,05

[1] Ni får inräknas i Cu-halten

[2] För svetsade konstruktioner får Pb ej överstiga 0,02 %

Hållfasthet

SIS brons	Tillstånd	Leveransform	Draghållfasthet SIS 11 21 10			Hårdhet SIS 11 25 10
			$R_{p0.2}$ N/mm² min	R_m N/mm² min	A_5 % min	HB min
57 16-00	Obehandlat	Tackor	–	–	–	–
57 16-03	Obehandlat	Sandgjutgods[3]	(250)	(640)	(13)	(140)
57 16-06	Obehandlat	Kokillgjutgods	(250)	(650)	(13)	(150)
57 16-15	Obehandlat	Centrifugalgjutgods, stranggjutgods	(250)	(670)	(13)	(160)

Värden inom parentes är inte bindande

[3] Avser hållfasthet på separatgjuten provstav

Övriga fordringar

SIS 11 00 01, Allmänna tekniska leverans- och kontrollbestämmelser för metalliska varor, gäller i tillämpliga delar.

Dessutom gäller SIS 21 98 51. Gjutgods av kopparlegeringar. Tekniska leveransbestämmelser.

Kontroll vid leverans

Tackor Kontroll med avseende på sammansättning är obligatorisk vid leverans och redovisas normalt med intyg rörande chargeanalys.

Gjutgods Eventuell överenskommelse om kontroll vid leverans skall ingå i köpeavtal och innehålla bestämmelser om vilka egenskaper som skall kontrolleras och om hur kontrollen skall gå till. (Jfr SIS 11 00 01.)

Beteckning

Brons 57 16-xx enligt SIS 14 57 16 eller SIS brons 57 16-xx

KOMPLETTERANDE UPPLYSNINGAR

Nedanstående upplysningar är ej fastställda som svensk standard och får ej anses bindande. Ytterligare upplysningar lämnas i MNC handbok nr 3. Gjutlegeringar. facklitteraturen eller av tillverkaren.

Kemisk sammansättning. Avvikande ISO-värden

		Al %	Cu %	Mn %	Pb %	Si %	Zn %
Tackor	Min	8.2	> 76.0	–	–	–	–
	Max	10.7		3.0	0.10	0.08	0.50
Gjutgods	Min	8.0	> 76.0	–	–	–	–
	Max	11.0		3.0	0.10	–	0.50

Cr och Mg finns ej i ISO

Tabell 1

Tekniska data

Densitet, ρ, vid 20 °C	7.6 g/cm^3
Smältintervall	1025–1040 °C
Värmeledningsförmåga, λ, vid 20 °C	42 W/(m · °C)
Längdutvidgningskoefficient, α, vid 20–200 °C	$18 \cdot 10^{-6}$ °C^{-1}
Resistivitet, ρ, vid 20 °C	200 nΩm
Elasticitetsmodul, E	120 000 N/mm^2

Motsvarigheter i dansk, finsk och norsk standard

SIS brons	Danmark DS 3001	Finland SFS	Norge NS
57 16	5716	2212	16 570

Tabell 2

TABLE 2 *Continued*

Grade	Type	Composition, percent														
		Carbon, max	Manganese, max	Silicon, max	Phosphorus,* max	Sulfur,* max	Chromium	Nickel	Molybdenum	Columbium	Selenium	Copper	Tungsten, max	Vanadium, max	Cobalt, max	Iron, max
CN-7M	20 Chromium, 29 Nickel, with Copper and Molybdenum	0.07	1.50	1.50	0.04	0.04	19.0-22.0	27.5-30.5	2.0-3.0	...	...	3.0-4.0	...	...	...	...
CN-7MS	19 Chromium, 24 Nickel, with Copper and Molybdenum	0.07	1.00	2.50–3.50	0.04	0.03	18.0–20.0	22.0–25.0	2.5–3.0	...	...	1.5–2.0	...	...	...	...
CW-12M	Nickel, Molybdenum, Chromium	0.12	1.00	1.50	0.040	0.030	15.50-20.00	remainder	16.00-20.00	...	...	...	5.25	0.40	...	7.50
CY-40	Nickel, Chromium, Iron	0.40	1.50	3.00	0.030	0.030	14.00-17.00	remainder	...	...	...	...	...	...	...	11.00
CZ-100	Nickel Alloy	1.00	1.50	2.00	0.030	0.030	...	remainder	...			1.25 max		...	...	3.00
M-35	Nickel-Copper Alloy	0.35	1.50	2.00	0.030	0.030	...	remainder	...			26.0-33.0		...	...	3.50
N-12M	Nickel-Molybdenum Alloy	0.12	1.00	1.00	0.040	0.030	1.00 max	remainder	26.0-33.0			...		0.60	...	6.00
CA-6NM	...	0.06	1.00	1.00	0.04	0.03	11.5-14.0	3.5-4.5	0.40-1.0	...	...	...	...	...	...	
CD4MCu		0.04	1.00	1.00	0.04	0.04	24.5-26.5	4.75-6.00	1.75-2.25			2.75-3.25				
CA6N		0.06	0.50	1.00	0.02	0.02	10.5-12.5	6.0-8.0								

* For free machining properties the composition of Grade CF-16F may contain suitable combinations of selenium, phosphorus, and molybdenum (Grade CF-16F) or of sulfur and molybdenum (Grade CF-16Fa) as follows.

Selenium, phosphorus, and molybdenum:	
Selenium, percent	0.20 0.35
Phosphorus, max, percent	0.17
Molybdenum, max, percent	1.50
Sulfur and molybdenum:	
Sulfur, percent	0.20 0.40
Molybdenum, percent	0.40 0.80

Other combinations of elements for free-machining properties may be agreed upon by the manufacturer and the purchaser.

* For the more severe general corrosive conditions, and when so specified, the carbon content shall not exceed 0.10 percent. This low-carbon grade shall be designated as Grade CH-10.

ENCL. 12

TABLE 2 *Continued*

* Chemical analysis is not normally required for the elements phosphorus, sulfur, and molybdenum, but if they are present in amounts over those stated they may be cause for rejection.

* Grade CF-8C shall have a columbium content of not less than 8 times the carbon content and not more than 1.0 percent. If a columbium-plus-tantalum alloy in the approximate Cb:Ta ratio of 3:1 is used for stabilizing this grade, the total columbium plus tantalum content shall not be less than [illegible] times the carbon content and shall not exceed 1.1 percent.

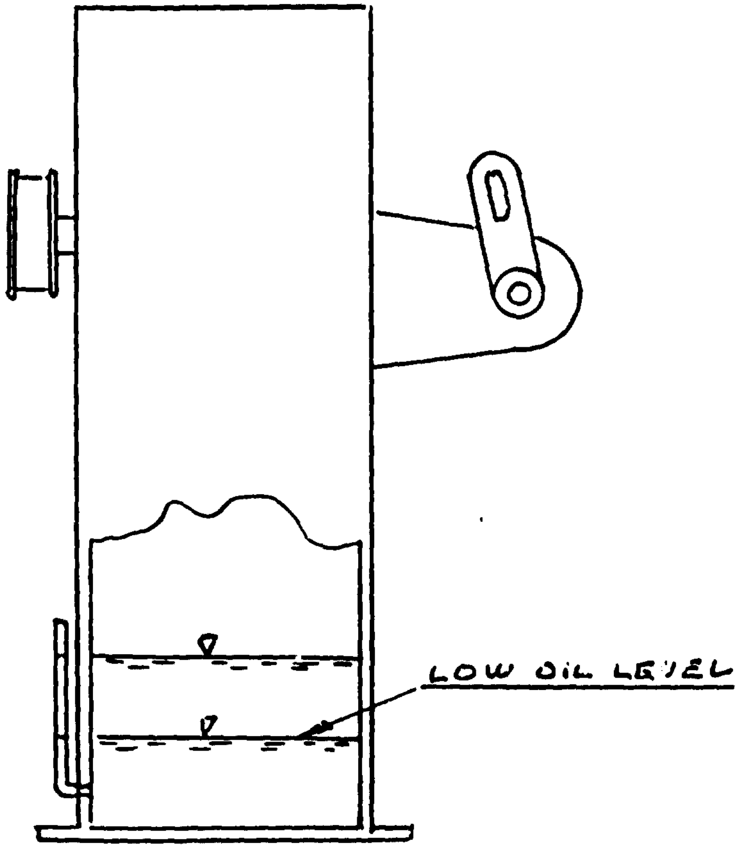
[illegible]

TABLE 3 Tensile Requirements

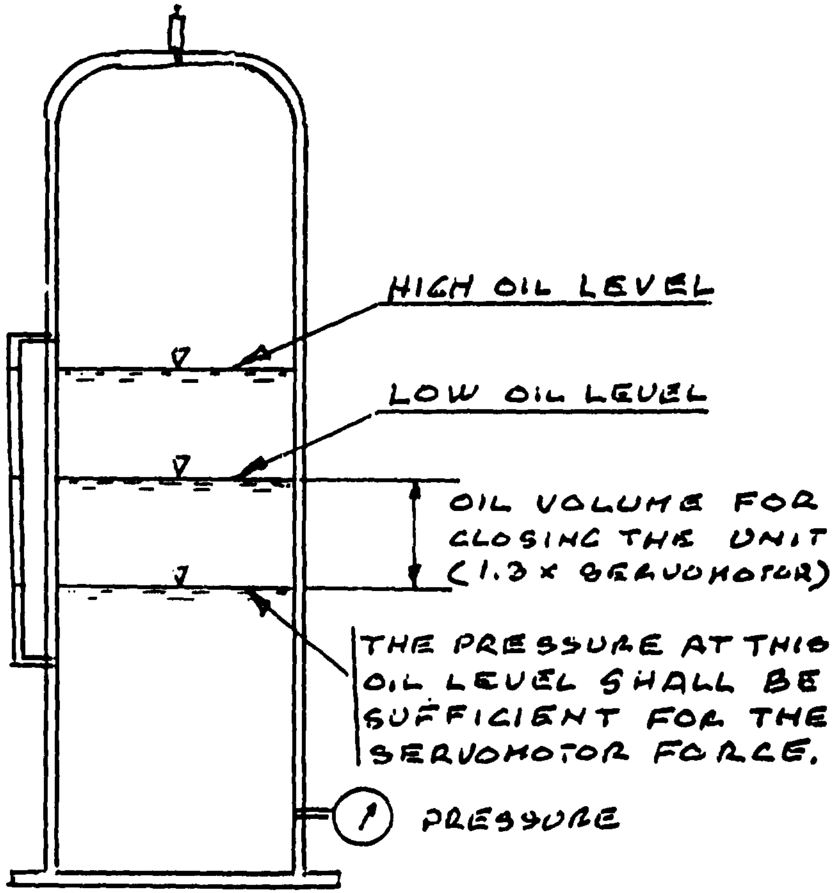
Grade	Type	Tensile Strength, min		Yield Point, min		Elongation in 2 in. or 50 mm, min, percent	Reduction of Area, min, percent
		ksi	MPa	ksi	MPa		
CF-8	19 Chromium, 9 Nickel	65	450	28	195	35	. . .
CG-12	22 Chromium, 12 Nickel	70	485	28	195	35	. . .
CF-20	19 Chromium, 9 Nickel	70	485	30	205	30	. . .
CF-8M	19 Chromium, 10 Nickel, with Molybdenum	70	485	30	205	30	. . .
CF-8C	19 Chromium, 10 Nickel, with Columbium	70	485	30	205	30	. . .
CF-16 F and CF-16Fa	19 Chromium, 9 Nickel, Free Machining	70	485	30	205	25	. . .
CH-20 and CH-10	25 Chromium, 12 Nickel	70[a]	485[a]	30	205	30	. . . .
CK-20	25 Chromium, 20 Nickel	65	450	28	195	30	. . .
CE-30	29 Chromium, 9 Nickel	80	550	40	275	10	. . .
CA-15 and Ca-15M	12 Chromium	90	620	65	450	18	30
CB-30	20 Chromium	65	450	30	205	. . .	. . .
CC-50	28 Chromium	55	380	. . .	. . .	. . .	. . .
CA-40	12 Chromium	100	690	70	485	15	25
CF-3	19 Chromium, 9 Nickel	65	450	28	195	35	. . .
CF-3M	19 Chromium, 10 Nickel, with Molybdenum	70	485	30	205	30	. . .
CG-8M	19 Chromium, 11 Nickel, with Molybdenum	75	520	35	240	25	. . .
CN-7M	20 Chromium, 29 Nickel, with Copper and Molybdenum	62.5	430	25	170	35	. . .
CN-7MS	19 Chromium, 24 Nickel, with Copper and Molybdenum	70	485	30	205	35	. . .
CW-12M	Nickel, Molybdenum, Chromium	72	495	46	320	4.0	. . .
CY-40	Nickel, Chromium, Iron	70	485	28	195	30	. . .
CZ-100	Nickel Alloy	50	345	18	125	10	. . .
M-35	Nickel-Copper Alloy	65	450	30[a]	205[a]	25	. . .
N-12M	Nickel-Molybdenum Alloy	72	495	46	320	6	. . .
CA-6NM	. . .	110	760	80	550	15	35
CD4MCu		100	689	70	483	16	
CA-6N		140	965	135	930	15	50

[a] When adequate weldability is stipulated the silicon content may have to be lowered, in which case the minimum required yield strength shall be 25 ksi (180 MPa).

ENCL. 13



LOW OIL LEVEL



HIGH OIL LEVEL
LOW OIL LEVEL
OIL VOLUME FOR CLOSING THE UNIT (1.3 x SERVOMOTOR)
THE PRESSURE AT THIS OIL LEVEL SHALL BE SUFFICIENT FOR THE SERVOMOTOR FORCE.
PRESSURE

Departamento Fuentes No Convencionales
de Energia

Attn lng. Flavio Ferran

Re. Design and Manufacture of
Hydraulic Mini Turbines

Many thanks for all the kind hospitality I encountered during my visit to the Institute in Cuernavaca.

There were two points left open in my recommendations - points 6. and 41. I have been in contact with specialists from SWECO and will now give the following recommendations.

For point 6., the Swedish practice is to use penstock supports according to Fig. 1 in Enclosure 1. When the expansion is rather big, an arrangement according to Fig. 2 can be used.

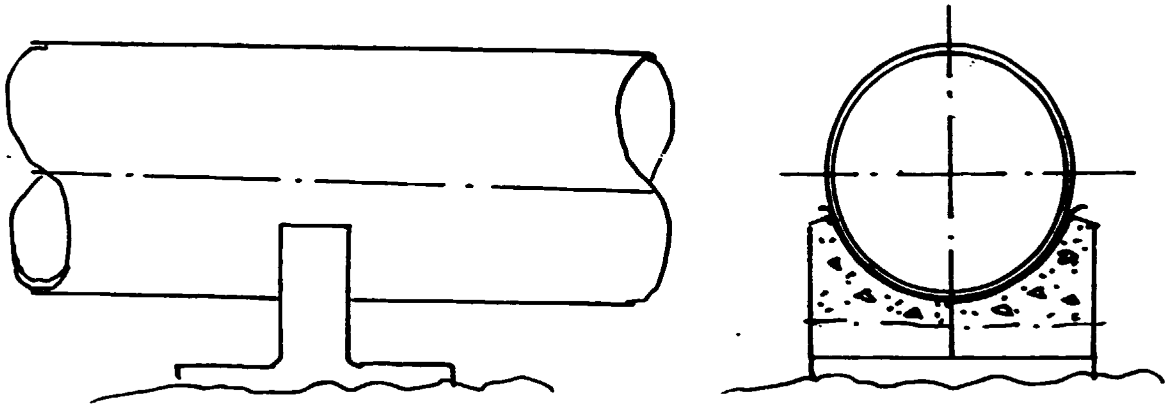
The questions about personnel safety raised in point 41 are serious. A protection device has to be installed, that prevents the unit from starting up, unless there are normal voltage on the grid-line-side of the breaker. Se Enclosure 2.

My best regards to you and all the personnel I met during my visit.

Very truly yours,

Evald Holmén

ENCL 1.



LAYERS OF BETUMIN-PAPER TO
A THICKNESS OF ∼5 mm.

FIG. 1

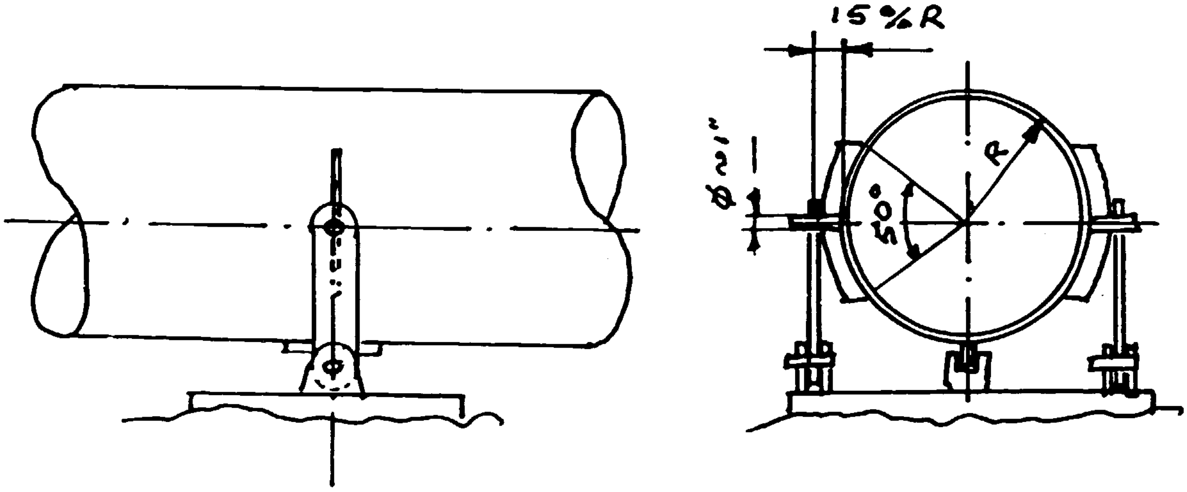
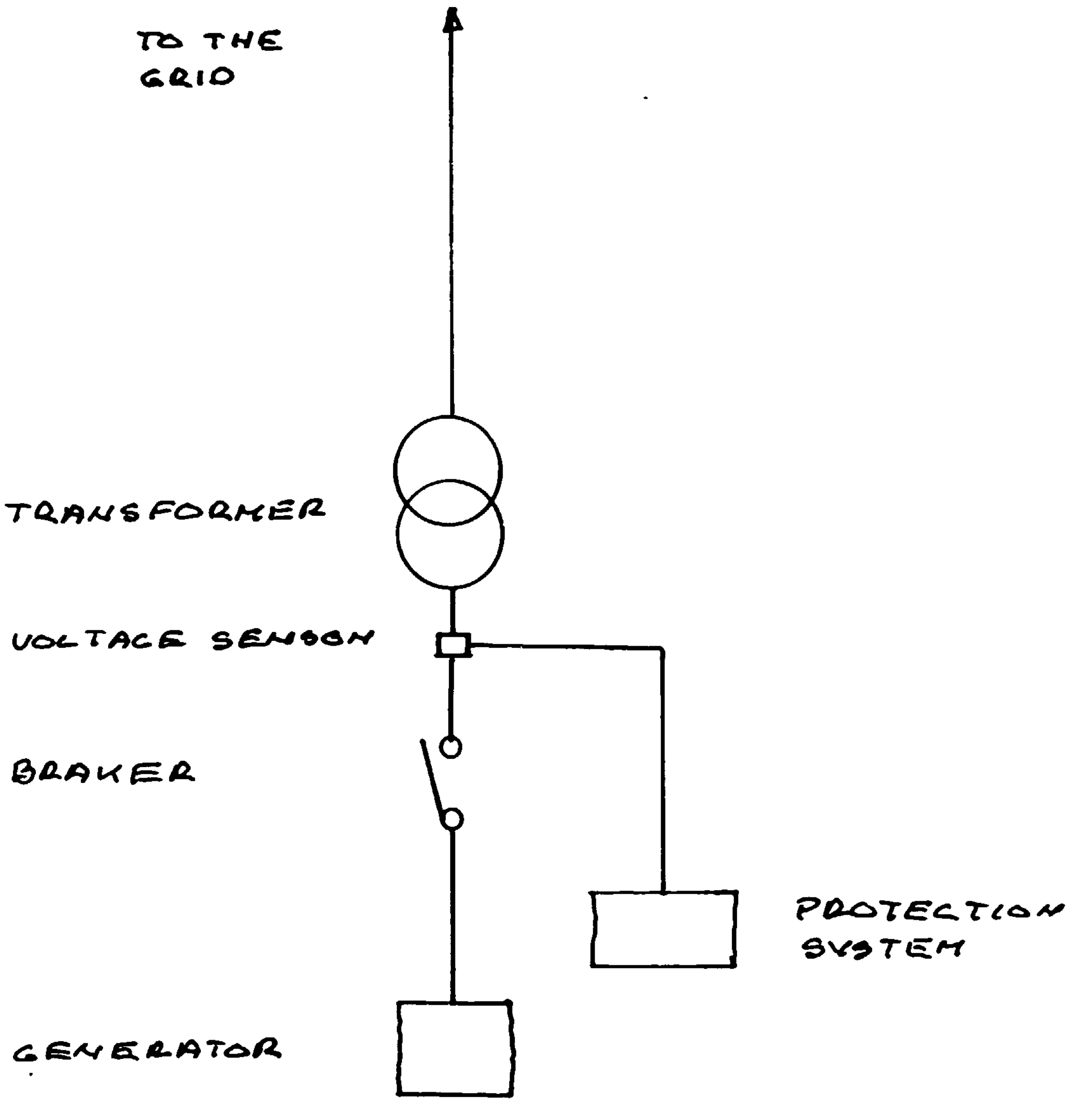


FIG. 2.

ENCL. 2.


TO THE GRID
TRANSFORMER
VOLTAGE SENSOR
BRAKER
PROTECTION SYSTEM
GENERATOR