COMPAGNIE GÉNÉRALE ÉLECTRIQUE DU CANADA / CANADIAN GENERAL ELECTRIC CO. LTD.
795, 1re Avenue / 795-1st Avenue
Lachine, Qc H8S 2B8

NOUVEAU DU 10 FÉV. '86 PANAFAX (514) 634-3329 EFFECTIVE 10 FEB. '86

DATE: May 21 1986

HEURE/TIME:

NOMBRE DE PAGES SUIVANT LA FEUILLE D'ENVOI:
NUMBER OF PAGES TO FOLLOW LEAD SHEET: One (1)

NOM/NAME: Mr. Bill Hynch [illegible] 5201

COMPAGNIE/COMPANY: Savannah District Corps. of Engineers

VILLE/CITY: Savannah, Georgia USA

NO. PANAFAX NO.: (912) 944-3350

DE/FROM: M.M. Gfeller

TELEPHONE: (514) 634-3411 Poste/Ext. 446 Dial Com 0*22500

SI VOUS NE RECEVEZ PAS CLAIREMENT L'UNE DE CES PAGES, VOUILLEZ APPELER:
IF YOU DON'T RECEIVE ONE OF THESE PAGES PROPERLY, PLEASE CALL:

Lise Shrivel, Ext. 225

COMMENTAIRES/COMMENTS:

FROM C.G.E. D.E.S. [illegible] 05/21/86 [illegible] P 1

May 21, 1986

Reference: R.B. Russell Dam Pump Turbines Units 5 to 8
Contract DACW 21-84-C 0015

This to confirm final prototype data for turbine output and axial thrusts as given over phone today:

1° Max turbine output at 157 ft net head with wicket gate opening corresponding to 30 mm on the model: 127 000 HP

Servomotor stroke will provide for this opening

2° Axial thrusts, according to final model tests and calculations:

- Maximum hydraulic thrust when operating as a turbine at normal speed : 505 000 lbs (downwards)
- Maximum hydraulic thrust when operating as a pump at normal speed : 1 035 000 lbs (downwards)
- Maximum hydraulic thrust at runaway speed : 1 [illegible] lbs (downwards)

Regards
WB Gfeller

cc: T Hotay
TH McOwen

FROM [illegible] MTL 05/21/86 12:55 P. 2 [illegible]
FROM [illegible] MTL 05/21/86 12:55 P. 1

Dominion
Engineering
Works

BY COURIER

March 7th, 1984

Department of the Army
Savannah District Corps of Engineers
200 East St-Julian Street
Savannah, Georgia 31402
U.S.A.

Att: Mr. S. Thomas Anderson
SASCB-SA

RE: RB Russell Dam Pump Turbines
Units 5 to 8
Contract DACW 21-84-C-0015

Dear Sir:

With reference to the above contract, we are providing the following outline for your review and consideration:

A) Contract Improvements:

1) Maximum pump power/Contract = 131,000
DEW power (Expected) 125,800

Power reduction for the motor/
Generators = 4,200 HP

2) Pumping Efficiency Contract Improvements

a) @ 161' head
89.7% test curve
89.5% contract requirements
+ 0.2% improvement

b) @ 148' head
88.9% test curve
88.8% contract requirements
+ 0.1%

Over 80% of the pumping operating head range has better efficiency than specified.

Dominion Engineering Works, 290 Ste Catherine, Lachine, Quebec, Canada H8S 2S8
Telephone (514) 634-3411 Cable Address: Dominion Montreal Telex 05-821220
A Division of Canadian General Electric Company Limited

3) Turbine Mode

| | |
|---|---|
| a) @ 179' head test curve performance | 186,000 HP |
| contractual requirements | 184,000 HP |
| Increased power available | 2,000 HP |
| b) @ 157' Test curve Perf. | 126,800 HP |
| Contractual requirements | 124,300 HP |
| Increased Power available | 2,500 HP |

B) Contract Deficiency:

Pumping Mode @ 141' head

| | |
|---|---|
| Contract requirements | 88.6% |
| Test Curves | 88.2% |
| Short fall | 0.4% |

C) Performance Curves

- Attached are the following performance curves:

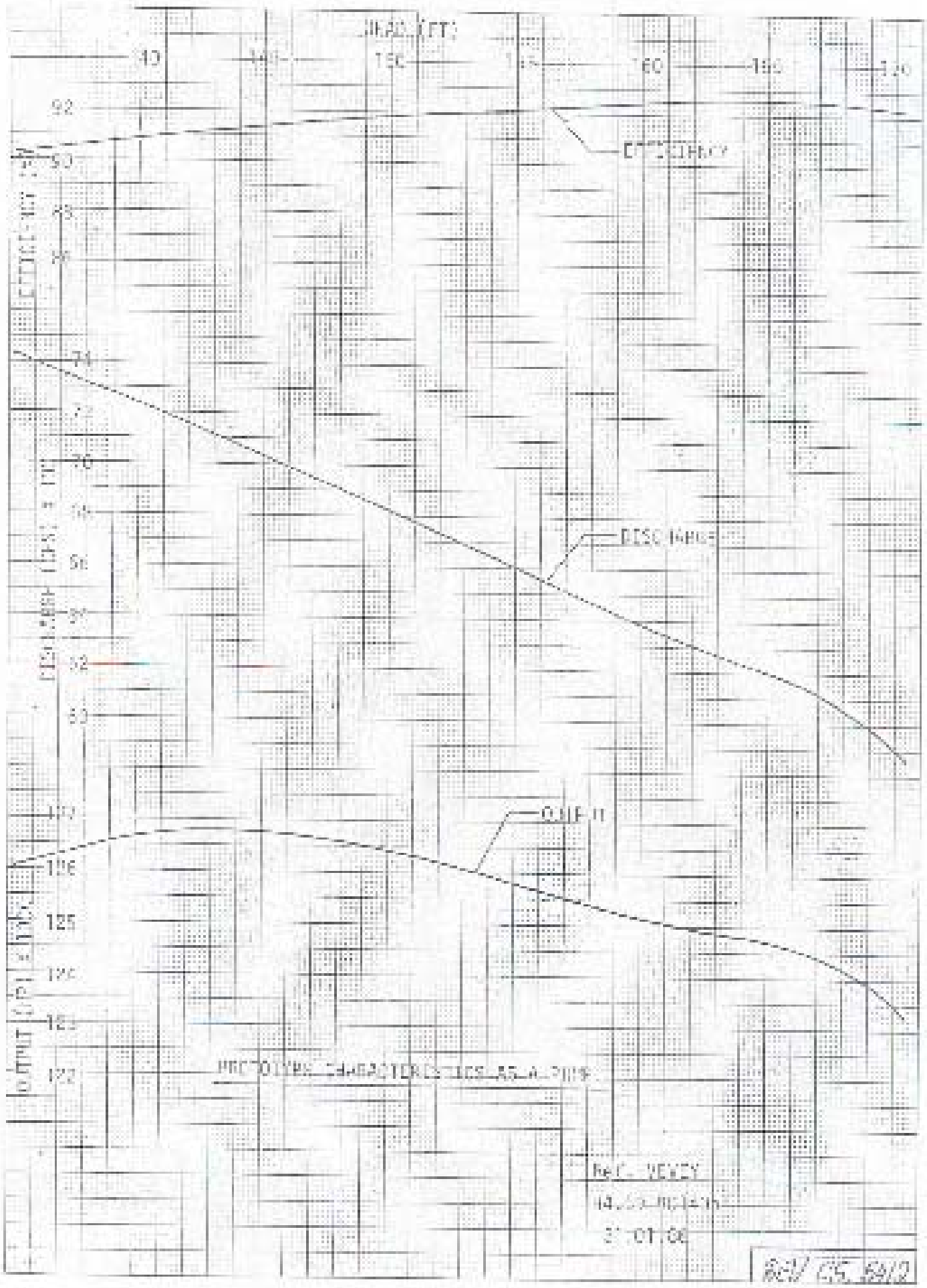
- DXW CS 5413 - Model characteristics as a Pump
- DXW CS 5417 - Prototype characteristics as a Turbine
- DXW CS 5412 - Prototype characteristics as a Pump

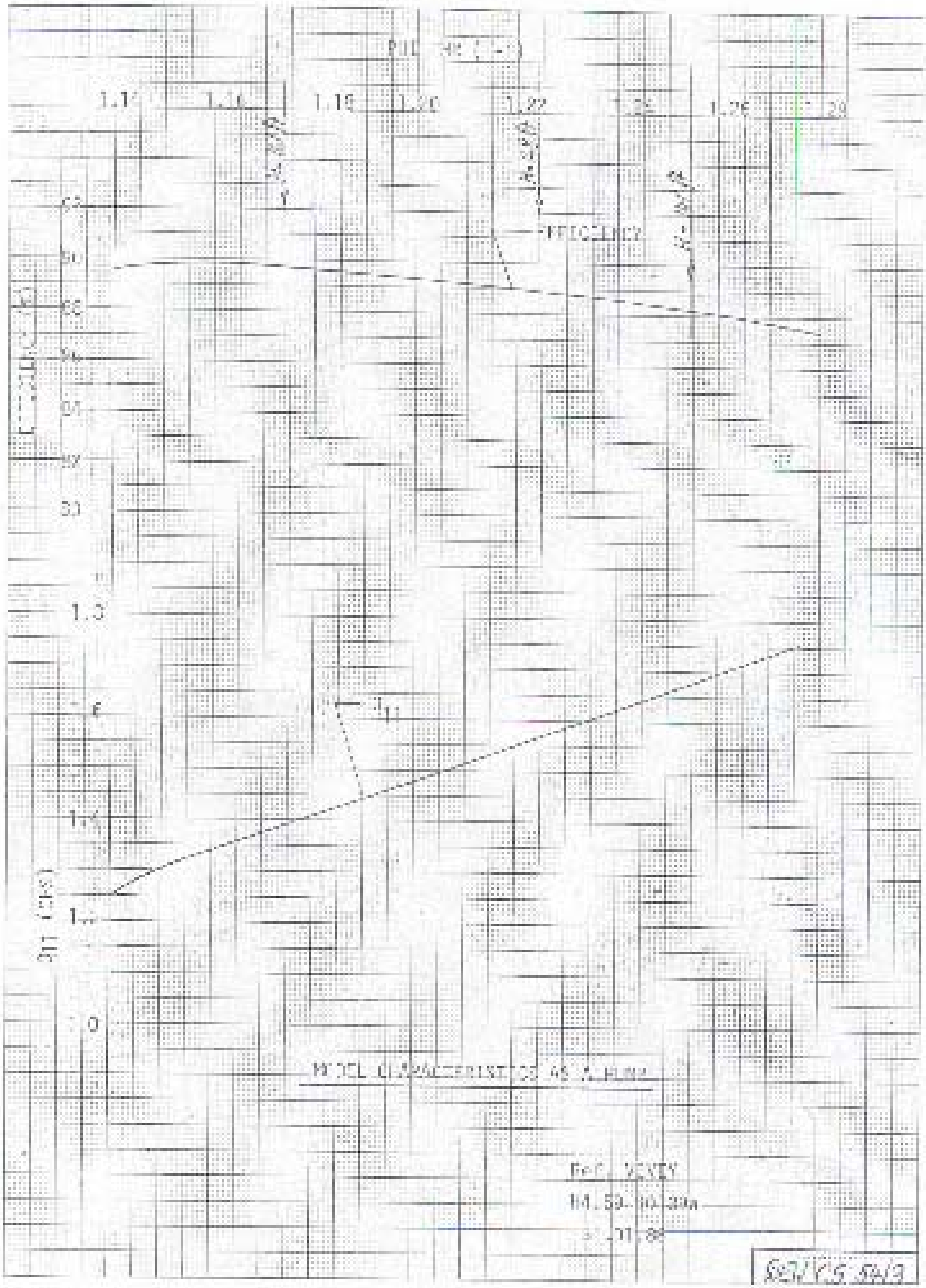
D) Contract Modifications

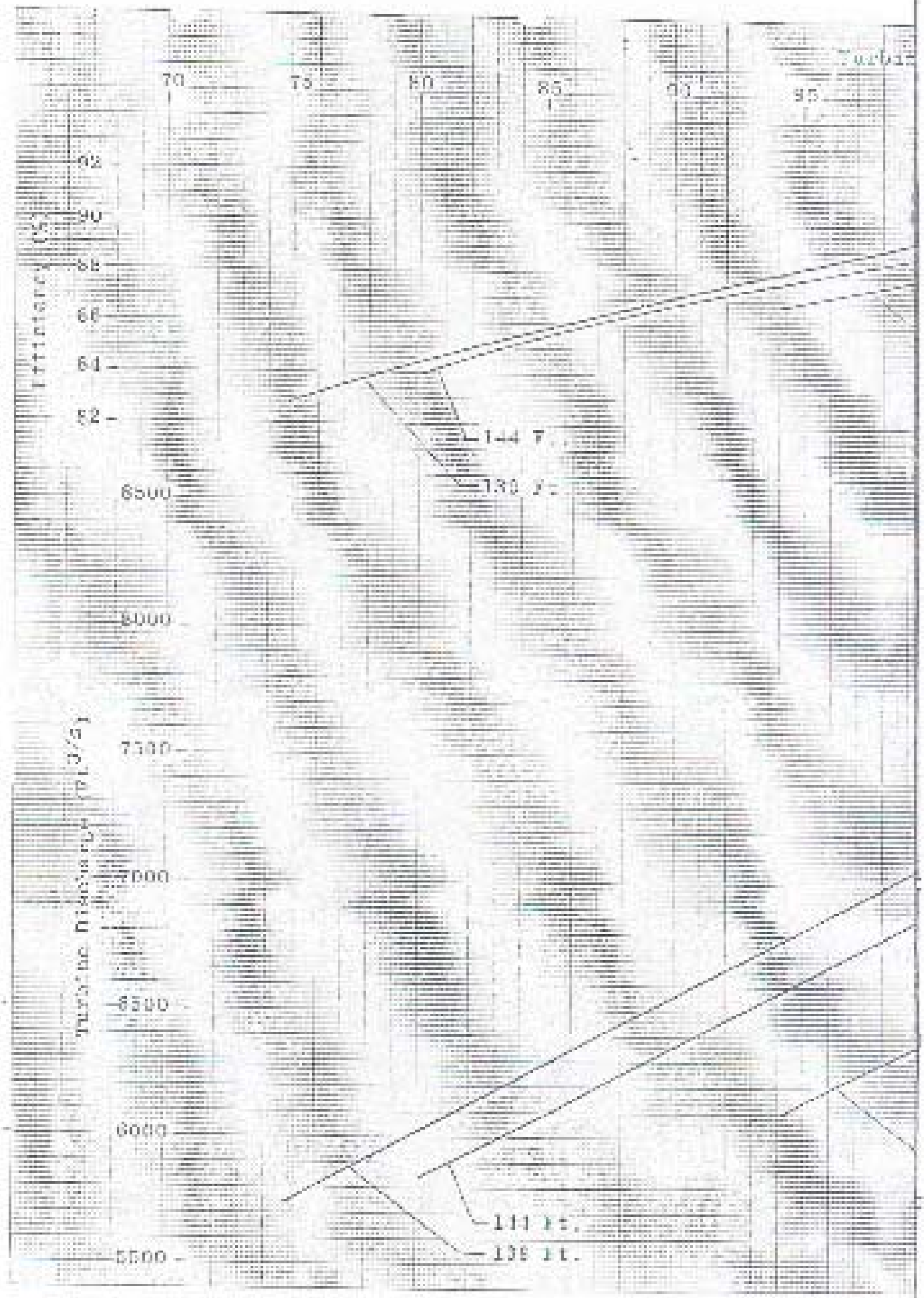
We believe that the Contract Improvements identified in Item A above, more than offset the Contract Deficiency outline in Item B above, therefore we respectfully request your modification of the contract in accordance with the foregoing information.

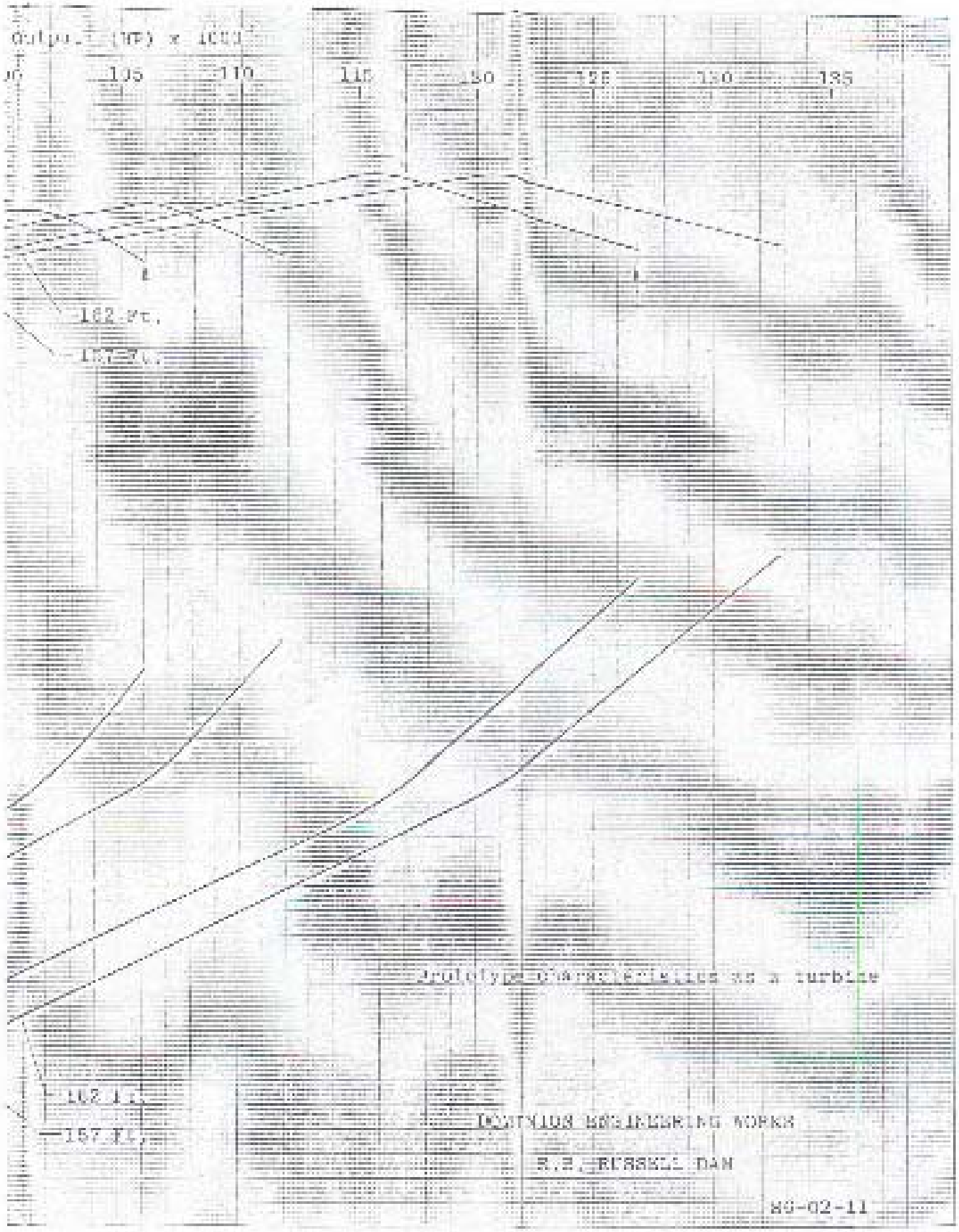
Yours truly,
DOMINION ENGINEERING WORKS
A Division of Canadian General Electric Co., Ltd.,

I. McKay, P.Eng.,
Manager-Turbine Sales Int'l.
Turbines & Generators

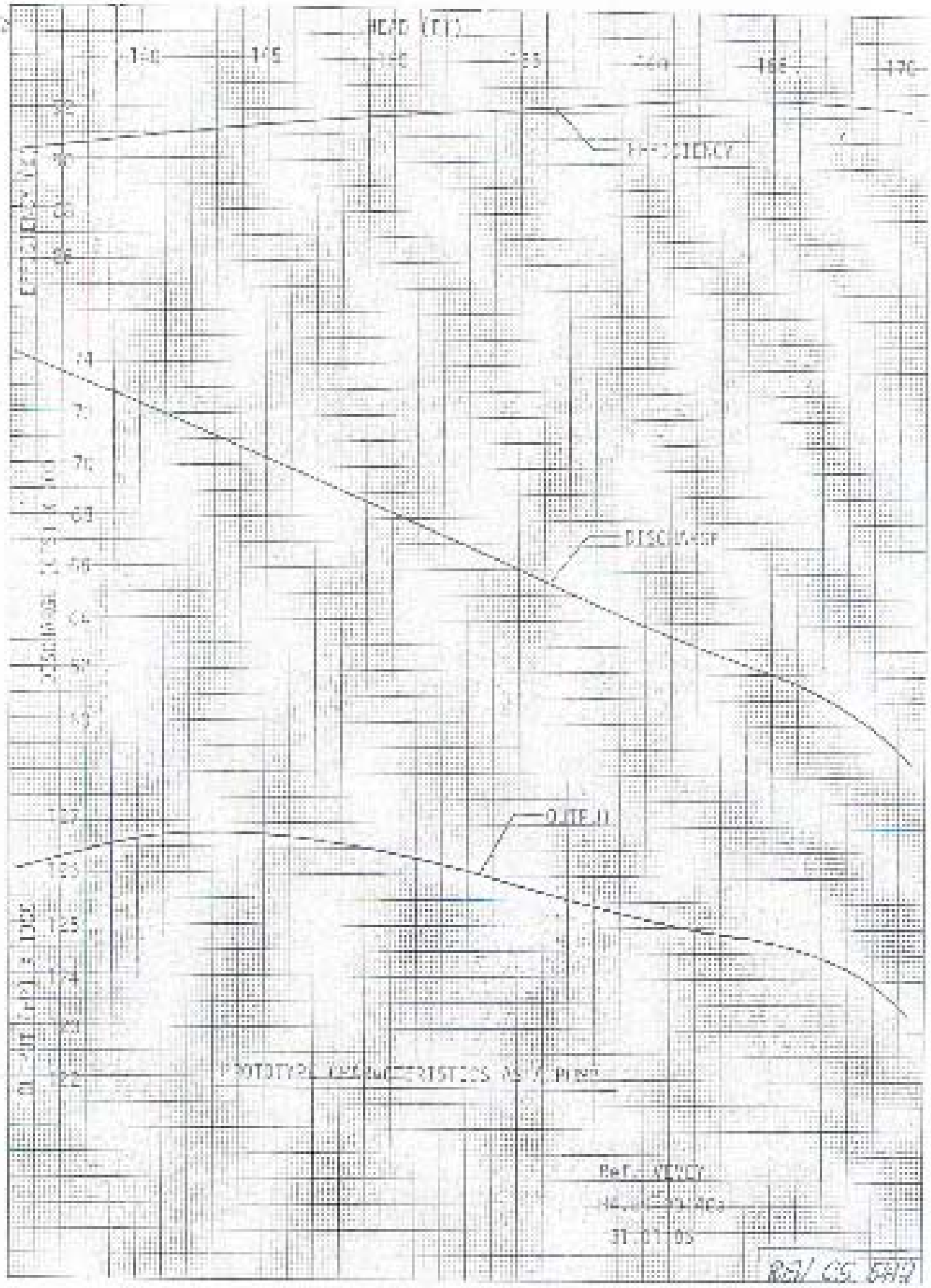

HEAD (FT)
140
150
160
170
180
190
120
92
90
88
86
74
72
70
68
66
64
62
60
EFFICIENCY
DISCHARGE
OUTPUT
126
125
124
123
122
PROTOTYPE CHARACTERISTICS AS A PUMP
REV. 05 5/12

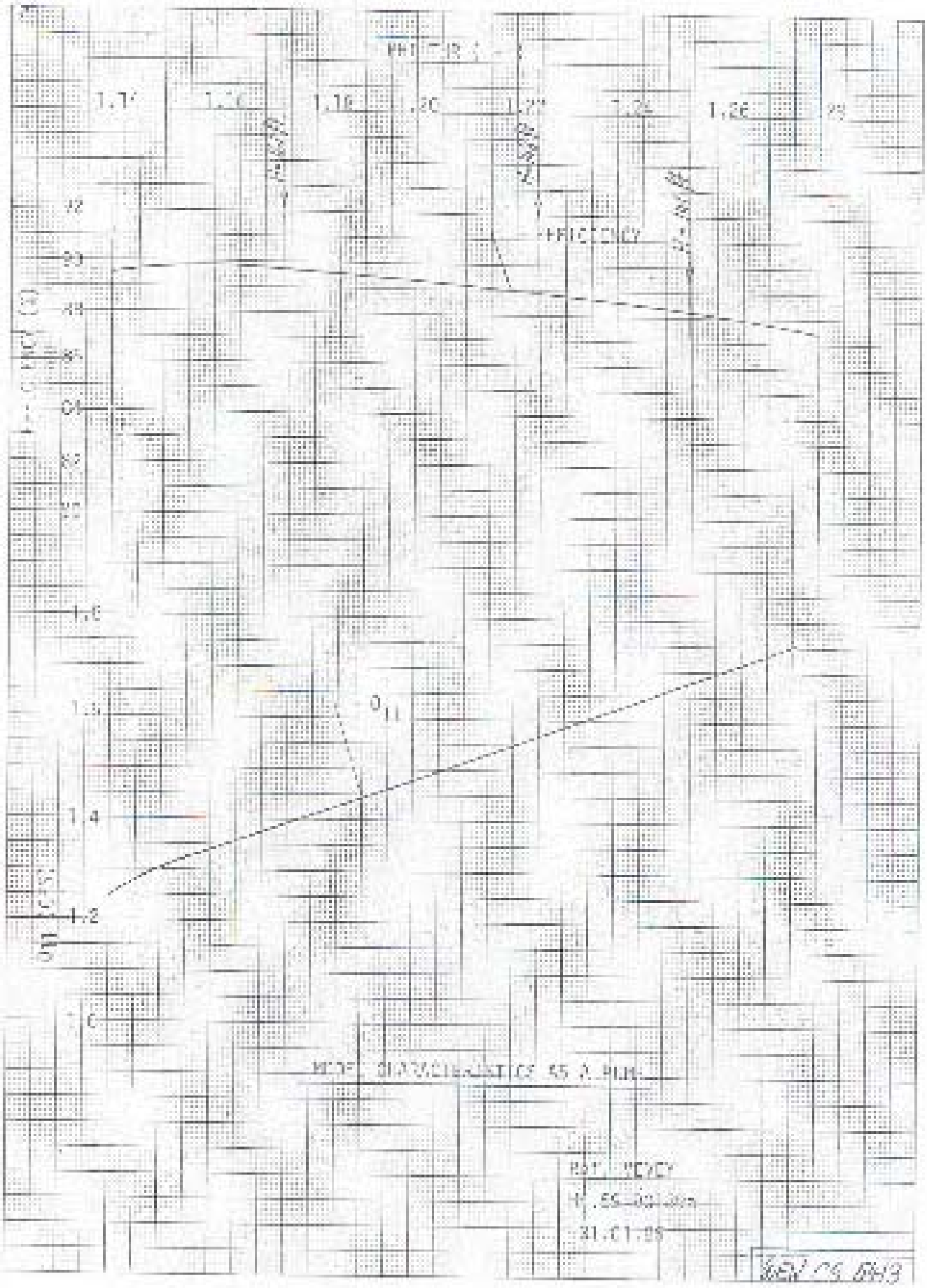

1.14
1.16
1.18
1.20
1.22
1.24
1.26
1.28
EFFICIENCY
1.0
MODEL CHARACTERISTICS AS A PUMP
Ref. VEVEY


70
75
80
85
90
95
92
90
88
86
84
82
8500
8000
7500
7000
6500
6000
5500
144 F.
130 F.
144 F.
130 F.

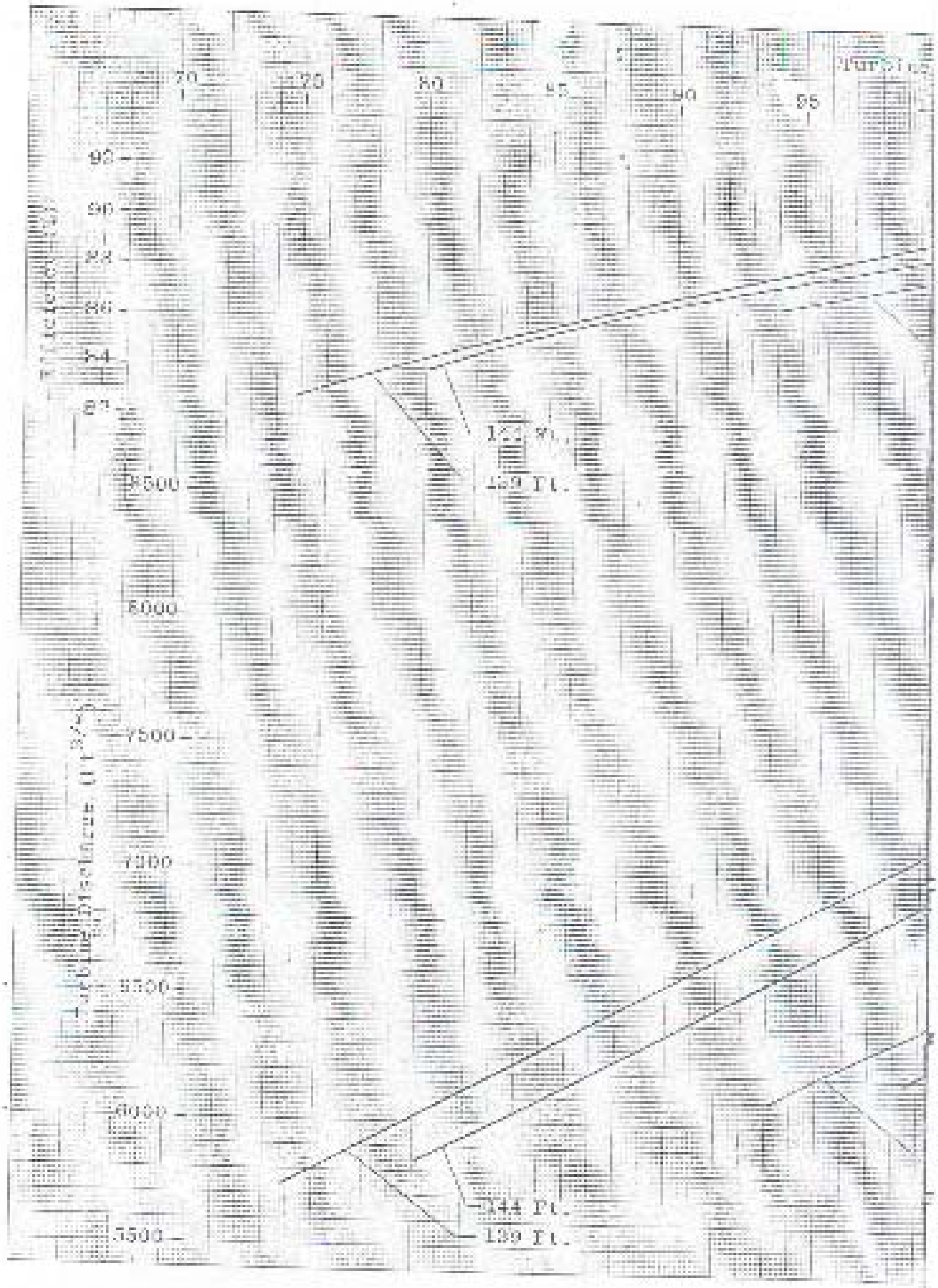

Output (HP) x 1000
105
110
115
120
125
130
135
162 Ft.
157 Ft.
Prototype characteristics as a turbine
162 Ft.
157 Ft.
DOMINION ENGINEERING WORKS
R.B. RUSSELL DAM
86-02-11

GEN C.S. 5417

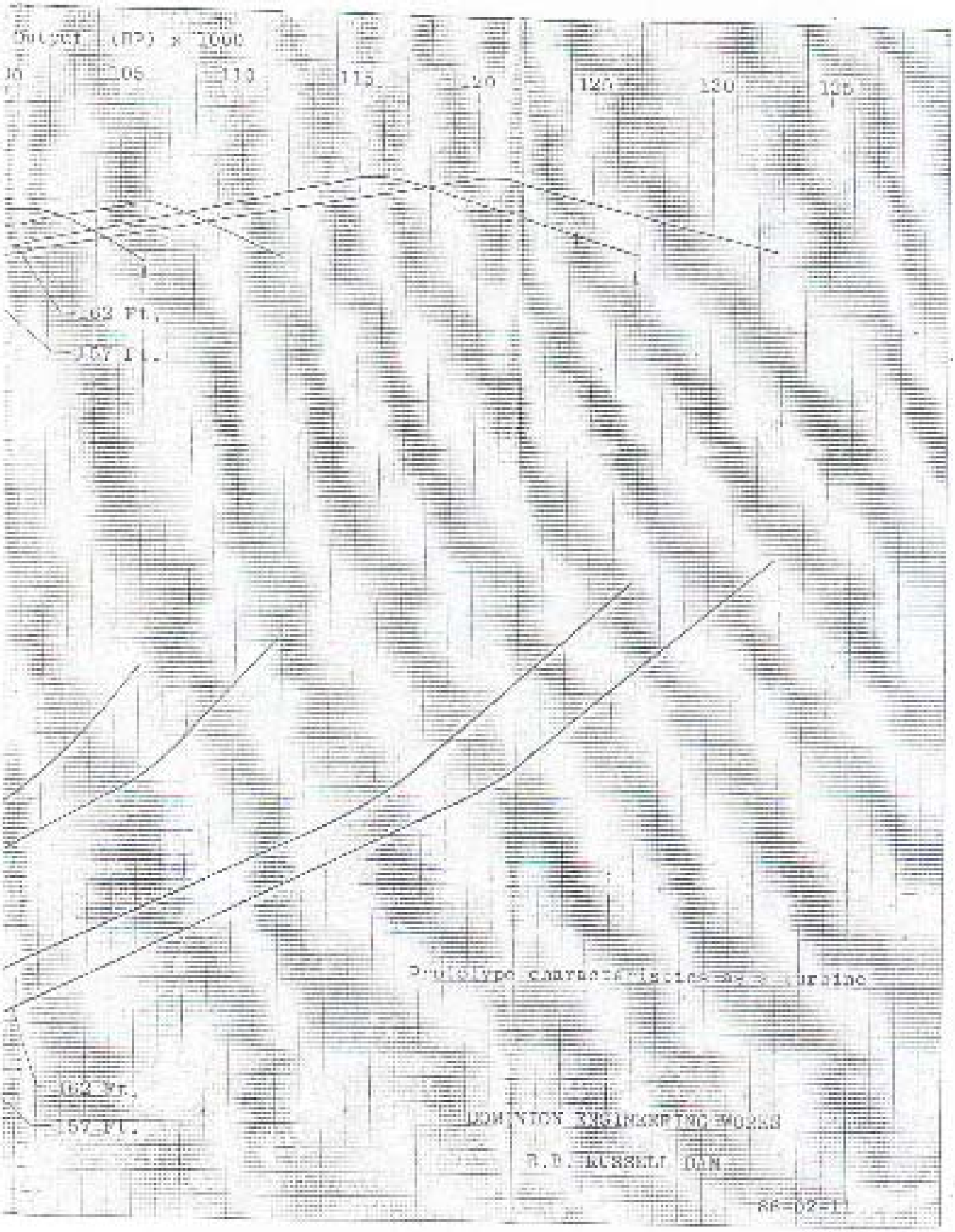

HEAD (FT)
EFFICIENCY
DISCHARGE
OUTPUT
PROTOTYPE CHARACTERISTICS



1.16
1.20
1.24
1.26
1.4
1.2
MODEL CHARACTERISTICS 55 A PUMP



Turbine
70
75
80
85
90
95
92
90
88
86
84
82
8500
8000
7500
7000
6500
6000
5500
144 Ft.
139 Ft.


Output (HP) x 1000
105
110
115
120
125
130
135
Prototype characteristics as a turbine
157 Ft.
DOMINION ENGINEERING WORKS
R.D. RUSSELL DAM
86-02-11

BEW C.S. 5417

DEPARTMENT OF THE ARMY
U.S. Army Corps of Engineers
WASHINGTON, D.C. 20314

REPLY TO
ATTENTION OF

DAEN-ECL

24 June 1988

SUBJECT: Richard B. Russell Pump Turbines; Contract DACW21-80-C-0015

Commander
U.S. Army Engineer Division, South Atlantic
ATTN: SADEN-IE
510 Title Bldg.
30 Pryor Street, S.W.
Atlanta, GA 30335-6801

1. Enclosed is a copy of the "Evaluation of Model Inches" report for the subject pump-turbines, dated 16 June 1988, prepared by DAEN-ECH-B. The report recommends (see page 12) that the model test be accepted even though the model fails to meet the guaranteed efficiency by 0.3% (88.3 vs. 88.5%) when pumping at 141 foot head. Subsequent to the witnessed model test, SADEN-IE investigated the amount of time that the operating head would be in this deficient range. Enclosed is a copy of SADEN-IE 08 dated 28 April 1988, subject: R. B. Russell Pumped Storage Head-Frequency. The conclusion is that it is unlikely that the units would be used in the pumping mode at less than 144 ft. of head because the 141 ft or less head occurs during flood stages. Also that in the unlikely event that the units were used in the pumping mode at the relatively low head that the annual loss to the government due to the 0.3% loss in efficiency would be less than $50 per year or a present worth over the life of the project of approximately $1,000.

2. The primary purpose for specifying a guaranteed efficiency at the 141 ft. minimum pumping head was to provide control over the shape of the pump efficiency curve from minimum to minimum pumping heads. Although the model is deficient in efficiency at the minimum head, the magnitude of the deficiency (0.3%) does not affect the overall shape of the curve inasmuch as the model exceeds the guaranteed efficiency at 146 ft. head by 0.3% and maximum pumping head of 161 ft. by 0.5%. In addition, the model results indicate that the units will exceed the power and efficiencies guarantees in the turbine mode and the pumping capacity guarantees in the pumping mode. As a result of the increase in power, the generator mode nameplate rating was increased from 91,055 to 94,500 kVA. Evaluation of benefits derived from improvements in performance in the normal operating range show they far exceed the theoretical loss due to the 0.3% inefficiency at 141 ft.

DAEN-[illegible]
SUBJECT: Richard B. Russell Pump Turbines; Contract DACW1-[illegible]-C-0015

head. There should not be any loss in operating benefits to the Government because its unlikely that the pumps will be operated at the 141 ft. head.

3. It is recommended that the Contracting Officer accept the withdrawal model test and that the contract by modified to change the guaranteed model pump efficiency at 141 ft. head from [illegible] to [illegible] with no change in contract price.

FOR THE COMMANDER:

2 Encls
1. Pump Model Test Rpt.
2. [illegible]-[illegible] by ctd
28 April 26

WILLIAM [illegible], JR.
Chief, Engineering Division
Directorate of Engineering and Construction

RICHARD B. RUSSELL

Pump - Turbines

Evaluation

of

Model Tests

June 10, 1986

by

DAEN-ECE-E
BOB KINSEL

BACKGROUND: The RICHARD B. RUSSELL PUMP-TURBINE model tests were conducted at Vevey Engineering Works, Vevey, Switzerland, during the week of 7-11 April 1986. Vevey Engineering Works performed these tests as a sub-contractor to Dominion Engineering Works. Mr. W. B. Sfeller represented Dominion Engineering Works at the tests. The following four representives from the Corps of Engineers witnessed the tests:

Mr. CHARLES CLELAND - Sav. Dist.
Mr. ROBERT KINSEL - OCE
Mr. JAMES PACE - SAD
Mr. JAMES KNOTNER - Mobile Dist.

Vevey Engineering did an extensive amount of development work on the turbine runner design. This is illustrated by the fact that six model runners were built and tested.

# Turbine Operation

When operating as a turbine the following conditions were specified:

## Power Output

104,000 HP @ 139 ft. net head

124,300 HP @ 157 ft. net head

## Turbine Model Efficiency

87.2 % Eff. @ 104,000 HP & 144 ft. head

86.0 % eff. @ 104,000 HP & 139 ft. head

The operating head range as a turbine is 139 - 161 feet.

The maximum horsepower of the turbine model occurred at 30 mm. model gate opening. The generator motor can utilize the larger capacity since they are used for the larger pumping horsepower requirements. Therefore, it was decided to make the "100 percent gate" equal to the model 30 mm gate.

Compute maximum power of turbine for various heads.

$$\phi = \frac{ND}{1837\sqrt{H}} = \frac{(120)(225)}{1837\sqrt{H}}$$

$$= \frac{14.89}{\sqrt{H}}$$

$$HP = HP_1 (Hd)^{\frac{3}{2}} (D_m)^2$$

Turbine Power at 100% Gate

| $\phi$ | Hd | $HP_n$ | HP |
|---|---|---|---|
| 1.188 | 157 | 0.1785 | 126,800 |
| 1.20 | 154.1 | 0.179 | 122,500 |
| 1.21 | 151.5 | 0.179 | 120,510 |
| 1.22 | 149.0 | 0.1795 | 117,400 |
| 1.23 | 146.6 | 0.1798 | 115,200 |
| 1.24 | 144.3 | 0.18 | 112,600 |
| 1.241 | 144.0 | 0.18 | 113,300 |
| 1.25 | 142.0 | 0.18 | 109,900 |
| 1.26 | 139.7 | 0.18 | 107,300 |
| 1.243 | 137.0 | 0.18 | 105,500 |

The predicted power output of the turbine at the specified heads of 137' & 157' feet exceed the specified power values. In general, the prototype turbine power output will exceed the the calculated power output by 2-3 percent.

Turbine performance at 144 ft. net head. At 109,000 HP the efficiency should be a minimum of 87.2 percent.

$\phi = 1.241$ at 144 ft head

144' Head

| Model Gate Opening | % Gate Opening | $HP_1$ | $Q_1$ | Eff. % | HP | Q cfs |
|---|---|---|---|---|---|---|
| 22mm | 73.3 | 0.155 | 160 | 85.3 | 96,700 | 6930 |
| 24 | 80.0 | 0.168 | 167 | 87.6 | 104,500 | 7320 |
| 26 | 86.6 | 0.174 | 171 | 86.6 | 108,500 | 7670 |
| 28 | 93.5 | 0.178 | 182 | 85.7 | 111,000 | 7930 |
| 30 | 100 | 0.18 | 189 | 83.9 | 112,300 | 8170 |

$$HP = HP_1 \, D^2 \, H^{3/2}$$

$$= HP_1 \, (19)^2 \, (144)^{1.5}$$

$$Q = Q_1 \, H^{1/2} \, D^2 = Q_1 \, (144)^{0.5} (19)^2$$

The efficiency of the turbine at rated power (104000 HP) and 144 ft head exceeds the specified efficiency.

Turbine performance at 139 ft. head is specified to be 86.0 % efficiency at 104000 horsepower.

139 ft Head

| Model Gate Opening | % Gate Opening | HP | Q | Eff in % | HP | Q cfs |
|---|---|---|---|---|---|---|
| 22 mm | 73.3 | 0.153 | 1.60 | 84.2 | 46,500 | 6810 |
| 24 | 80.0 | 0.168 | 1.70 | 87.1 | 47,400 | 7235 |
| 26 | 86.6 | 0.175 | 1.78 | 86.5 | 103500 | 7576 |
| 28 | 93.3 | 0.178 | 1.84 | 85.2 | 105300 | 7871 |
| 30 | 100 | 0.180 | 1.90 | 83.5 | 106500 | 8087 |

At 104,000 HP the efficiency is greater than 86 %. Therefore efficiency requirements of the unit as a turbine are satisfactory.

Turbine Cavitation: The specifications require the turbine to have 12 feet of margin for tailwater elevations above 318.0'. For tailwater elevations between 312.0' to 318.0' the cavitation shall vary from 6 to 12 feet respectively. Cavitation shall be based on 80°F water.

The model showed excellent cavitation characteristics when operating as a turbine. Only the turbine tests at a $\phi$ value of 1.190 had a decrease in performance sufficient enough to determine the cavitation margin. To show that the cavitation requirements that were specified have been met, the margin will be calculated for TWL of 318' at 151' head.

At plant sigma there were not any cavitation bubbles present on the model runner.

$\phi = 1.190$ , Head = 157'

Plant $\sigma_p$

$$\sigma_p = \frac{H_a - H_v - H_{sp}}{H} = \frac{32.15 - (18)}{157} = 0.314$$

Critical $\sigma_c$

$$\sigma_c = \frac{H_a - H_v - H_{sc}}{H}$$

Cavitation margin, if submergence

$$\sigma_p - \sigma_c = \frac{H_a - H_v - H_{sp}}{H} - \frac{H_a - H_v - H_{sc}}{H}$$

$$H(\sigma_p - \sigma_c) = H_a - H_v - (300 - TWL_p) - [H_a - H_v - (300 - TWL_c)]$$

$$= TWL_p - TWL_c$$

$$(157)(0.319 - 0.12) = \text{margin}$$

margin = 31.0 feet

Hysteresis: This phenomenon is a hydraulic condition which causes the performance of the unit to follow different operating conditions depending on whether $\phi$ $\left(\phi = \frac{ND}{1827\sqrt{H}}\right)$ is being increased or decreased. At the lower power levels, the hydraulic performance of the unit is very poor. If operated at this condition, the unit would be subject to vibration and poor hydraulic conditions. This condition can only happen on the installed units at R.R. Russell at speeds above synchronous speeds. Therefore, it should present no operational problems at that project.

# Pumping Performance

## Specified Capacity and efficiency

| Net Head | Specified Efficiency | Capacity $ft^3/sec$ |
|---|---|---|
| 161' | 59.5 | 6821 |
| 148' | 58.8 | 6763 |
| 41 | 86.4 | 7159 |

Best efficiency of the unit, when pumping, occurs at a model wicket gate opening of 20 mm. With a model to prototype ratio of 19, the prototype gate opening should be 380 mm (19x20) or 14.96 inches. Performance of the unit varies very little with wicket gate openings ± 0.5 mm on the model. Therefore the pumping mode should have the

wicket gate opening set between 14 9/16 inches and 15 5/16 inches. The wicket gates should be set to operate at the quietest or least vibration point within this range.

Pumping Performance

| Head | $\phi$ | $Q_m$ cfs | $Q_p$ cfs | Specified Q | Model Eff. | Specified Eff. |
|---|---|---|---|---|---|---|
| 161 | 1.173 | 1.38 | 632.1 | 5800 | 90.0 | 89.5 |
| 148 | 1.224 | 1.57 | 689.5 | 6200 | 89.1 | 88.8 |
| 141 | 1.254 | 1.67 | 715.4 | 6450 | 88.3 | 88.6 |

$$Q_p = Q_m \, H^{1/2} \, D^2$$

$$eff_{pump} = \frac{Q\,H}{8.81\ HP}$$

The previous chart shows that the turbine exceeds the pumping and efficiency requirements for all conditions except the efficiency requirement at 141 ft head.

It is recommended that the government accept the less than specified efficiency at the minimum head, for the following reasons:

1. The 141 ft head is an extremely rare head condition that briefly (or never) will occur. Therefore, the economic loss to the government is very low or nothing. This is the main justification for accepting the model tests.

2. Developing the model for a pump-turbine is a very complex job. It is without a doubt that very Engineering spent a considerable

amount of money and time to develop a high quality hydraulic design of the unit. Many of the problems they encountered, such as unstable pumping range and the hysteresis in the turbine operation, required considerable effort in order to move these out of the operating range of the unit. It is a very subjective part of the turbine contract to determine what is an acceptable margin. The contractor's motivation was always that they wanted to develop the best hydraulic design for the project. They built six model runners in order to achieve this goal and then furnished a runner larger than the minimum size specified in order to increase the margin on the pumping unstable region of the performance curve. One of the effects, however, of furnishing the larger runner is to decrease the pumping efficiency at the minimum head. With a smaller runner the turbine would be very close to meeting the

performance guarantees. It does not seem fair to penalize the contractor when he is furnishing a larger than specified size turbine runner (and more costly) in order to provide a better unit.

3. The specifications were developed from pump-turbine data furnished by one manufacturer. This was the case since no other manufacturer had tested pump-turbines for this head range. With the limited data available it was not possible to determine performance requirements that several manufacturers could meet which is our normal way of preparing specifications. While we feel certain that the other manufacturers could have met the specification, we would have specified a lower efficiency for the unit had this test data been available prior to preparing the specification

4. This is a very subjective evaluation of the model runner, but it is quite possible that had the manufacturer used a different method of constructing the model, then the model efficiency could have been increased enough to meet the specifications. The turbine runner was made of cast aluminum with a thin heating of epoxy applied to the runner. The epoxy could not be polished the same as a brass runner. The epoxy did not appear to be as smooth as a polished runner, which could have been enough of a factor to pass the test. Achieving a higher model efficiency by polishing the runner would not help the protype units. It is standard practice for manufacturers to use highly polished brass runners for the model acceptance tests.

Pumping Instability: At pumping heads above the maximum plant head, the pump-turbine has an unstable region. This unstable region is caused by recirculation of water within the runner and is typical for many pumps. Operation of the unit would be very rough. This condition also is a function of gate opening. There is slight chance that the unit could pass through this range during start-up of the pump when the wicket gates are opening. Increasing the wicket gate opening time will decrease the chance of passing through this rough range.

Cavitation When Pumping: The pump met all the cavitation specifications. However, there were some cavitation bubbles present in the pump mode. Stainless steel will be placed on the runner in the areas where the bubbles appeared.