AN ECONOMIC STUDY OF TURBO-GENERATOR UNITS TO MEET THE FUTURE DEMANDS OF THE VIRGINIA POLYTECHNIC INSTITUTE HEATING AND POWER PLANT

by

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II

INTRODUCTION

The primary function of the Heating and Power Plant of Virginia Polytechnic Institute is to produce steam to heat the various buildings on the campus and to provide process steam for the dining hall, the infirmary, and the college creamery and laundry. The plant also generates electric power as a by-product. This power is used by the campus and sold to the community of Blacksburg. The present plant facilities are not capable of supplying the total electric load and therefore over half of the power is purchased from the Appalachian Electric Company. The electric power produced by the college plant is controlled by the demand for process and heating steam. Power in excess of this is supplied by Appalachian Electric Power Company.

The college power plant is now operating at approximately peak steam load during the heating season. The heating season is considered to be six months, October through March. The two present turbo-generators in use are reaching the end of their expected life. It is believed that only one more overhaul on these turbines will be of economic value.

According to reports on future college enrollment and the expected building program, the process and heating steam load and the electric power load will increase beyond the capacity of present plant output. This situation has created a problem dealing with the replacement of the present equipment to meet the future demands. It is felt that at some time in the near future it will be economically feasible to install a larger, more efficient turbo-generator unit than those now in use. In view of this possibility the question arises as to the most economical size and type of unit to purchase.

It is considered best to approach this problem through an economic study based on the expected future steam and electric energy loads for the average heating season only.

This thesis will (1) investigate the effect of using several different throttle pressures and temperatures on the selection of the most efficient and economical unit, and (2) determine the effect that one, two, and three stage feedwater heating has on the selection of the unit and the savings involved by the use of a more efficient cycle. It will be assumed that the new unit selected would be installed in the plant for the heating season of 1959-60, and that the present turbines will not be in operation.

Since the investment required for such a unit would probably depend upon the business cycle, this investigation will not be approached with the view of the unit paying for itself in a given time. The expected life of a turbo-generator unit is usually considered to be 25 years. If the selected unit is installed in 1959, the year 1966 would represent approximately one quarter of its expected life. Therefore, the heating season of 1966-1967 will be used in this investigation to compare the savings in the cost to the college of steam and electric energy.

III

REVIEW OF LITERATURE

In order to determine the future load demands on the Heating and Power Plant a study must be made of the expected building program of the college, which will be governed by the future enrollment. Mr. P. H. Farrier states in his report on future college enrollment:

"It seems obvious that fairly soon the college will be called upon to take care of a greatly increased number of students, probably 5,000 five years from now. If we had facilities we probably could have from 7,000 to 10,000 students fifteen years from now. To take care of that many would mean doubling our physical plant."

"After this fall (1952), enrollment will gradually increase for a period of five years and then may sharply increase until 1960 and will continue to increase for at least another decade." [1]

The electric load will increase with the college expansion, but the greatest increase will come from the town of Blacksburg. Therefore, the increase in electric load will not be determined directly by the growth of the college. In order to determine this load growth a study must be made of past records.

"Tremendous load growth of postwar years has taxed the ingenuity of planning engineers and required the utmost cooperation of interconnected electric systems and industrial customers."

"Houston Lighting and Power Company's system has almost trebled since 1946, setting an annual growth rate of 16.2 per cent."

"Load growth since 1916 has averaged 13.9 per cent per year. During the pre-depression period of 1916-1930, the rate was 20.6 per cent. The depression period reduced it to 15.3 per cent. But since 1932, the growth has averaged 12.8 percent; this is considered more representative of present day conditions." [2]

Three other investigations have been made on turbo-generators units to meet the future demands of the Heating and Power Plant. The first investigation was by J. E. McMurrer, Jr. (14) on an economic study of a 2000 Kw unit. Mr. McMurrer's study was carried to the heating season of 1958-1959. The second investigation was made by V. F. Anderson (15) on an economic study of a high pressure boiler and turbo-generator. His investigation was also carried to the heating season of 1958-1959. The third investigation was by C. C. Wagoner (16) on an economic study of a 5000 Kw three stage extraction condensing turbo-generator. His investigation was carried through to the year 1959.

The results of these investigations were favorable toward the installation of a new unit, but it was felt that a further investigation should be made as to different size turbines with different throttle conditions. Since any unit decided upon will probably not be installed before 1959, it was felt that this investigation should be carried to some time in the future past the year 1959, in order to determine values for future loads after the unit is installed.

In the study of back pressure turbo-generator units it may be said that "by generating steam at a pressure above the process level and using the expansion energy in a turbine, power can be produced at a very low incremental fuel rate. The large economic gains obtained from combining the need for power and process steam have led to the development of a wide variety of steam turbines to supply both uses."

"Basically, the turbine accommodates itself to the steam flows required by the process, while extracting as much power as practical. The flow through the turbine must balance exactly the process requirements."

"When steam must be generated to serve the needs of the process, by-product power can be made by a process turbine at an additional fuel consumption of approximately 4250 B.T.U. per Kw-hr as compared with 11,500 B.T.U. per Kw-hr for prime power in an equivalent plant." [3]

"Electric power will be provided most economically in those plants where steam is used for manufacturing process and where all the required power is generated, as a by-product, by the steam before it is used in processing." [4]

"It is safe to conclude that industrial generation of electric power can be justified on an economic basis when process steam requirements are equal to or greater than that required to produce all the plant's electric demands on a by-product basis." [5]

All these stated facts apply to the Heating and Power Plant.

In order to determine the most efficient operating conditions for a proposed turbo-generator unit, the following statements were reviewed:

"When planning a new installation or trying to improve an existing one and both power and steam are required in substantial quantities, it is well to keep in mind.

(1) "Power can be made at a low fuel cost whenever sizeable quantities of steam must be reduced from a high pressure to a low pressure."

(2) "When buying new boilers consider the large possible savings in power costs which can sometimes be made by using conservatively high steam pressures and temperatures." [6]

"Many combinations of steam pressures and temperatures are available, and have been used in the past. However, the hundreds of thousands of kilowatts of turbine capacity purchased prior to 1940, practically all of them can be grouped into four general classes of initial steam conditions, as follows:

400 psig. , 750 F
600 psig. , 825 F
850 psig. , 900 F
1250 psig. , 950 F

"These permitted a choice of four steam conditions, any one of which could result in economical steam plants. They are matched to give reasonable moisture conditions in the lower pressure turbine stages with resulting long exhaust-end blade life." [7]

It was felt that the use of feedwater heaters in a regenerative cycle would increase the efficiency of the plant cycle and be of economic value. This is backed up by the following statements:

"The use of feedwater heaters has contributed more than any other single item to the remarkable increase of efficiency of power stations." [8]

"The advantages of the regenerative cycle are (1) the reduction of temperature stresses in the boiler by introducing hot feedwater rather than cold and (2) the increased economy." [9]

Most of the calculations of this investigation are based on the results of heat balances. Mr. P. J. Potter says, "A common rule of thumb is to select extraction pressures that will give approximately equal temperature rise in each feedwater heater."

"The importance of the heat balance in the design of steam power plants cannot be overestimated. Once the capacity of the station is determined the proper size of the boiler, heaters, pumps, piping - in fact practically all the mechanical equipment - can be determined solely by a heat balance." [9]

When selecting the number of heaters to use in a regenerative cycle it should be kept in mind that "the number of heaters used and the feedwater temperature attained both have an important bearing on the thermal performance of the turbine and station. The complication and expense generally keep the number of heaters down to 2 or 3 for medium size units."

"In general, each heater added in the system improves the performance, but in less amount than does the proceding one, whereas it costs just as much; hence, according to the law of diminishing returns, there is, in each case, an economic limit to the number of heaters that should be used." [10]

IV

SUMMARY

This investigation was conducted to determine the indicated average heating season saving which could be expected from several different size turbines with different throttle conditions and varying stages of feedwater heating when operating in the Virginia Polytechnic Institute Central Heating and Power Plant. Such a saving in cost of power produced locally is possible because of the higher cost of power when purchased from the Appalachian Electric and Power Company. This investigation covers the heating season of 1966-1967, and assumes that the present turbo-generators will not be in use at that time.

The electric load and steam demand load for the average heating season day of 1952-1953 were taken from existing records and were projected into the future to the heating season of 1966-1967. Two values were used as factors to project the steam demand into the future because of the uncertainty of the future college expansion. From this data electric load versus time curves and steam demand versus time curves were plotted for the average heating season day. This information, together with steam rates and exhaust rates for one, two, and three stage feedwater heating cycles for each proposed turbo-generator unit was used. Four curves were plotted for each throttle condition for the average heating season day in order to determine the kilowatt hours which could be generated by each proposed unit of sufficient size to supply the expected future steam demand. These curves were: Curve No. 1, the expected total electric load; Curve No. 2,

the expected electric power generated by the proposed unit with one stage feedwater heating; Curve No. 3, the expected electric power generated by the proposed unit with two stage feedwater heating; Curve No. 4, the expected electric power generated by the proposed unit with three stage feedwater heating. The areas under curves two, three, and four represent the kilowatt hours which could be generated by the proposed unit supplying the expected steam demand.

The value of these areas in terms of dollars saved was determined by multiplying the areas, in square inches, by a scale constant, 1000 Kw-hrs per square inch, and by $0.008 per Kw-hr minus the fuel cost per Kw-hr. The value $0.008 is the minimum rate of energy paid by the college for purchased power from the Appalachian Electric Power Company. The resulting figure represented the average daily saving for the heating season. This figure was used to calculate the total saving for the heating season.

The indicated savings which might be expected for the heating season of 1966-1967 ranged from $89,000.00 for a throttle condition of 250 psig. and 500 F and a 3750 Kw unit with one stage feedwater heating to $249,480.00 for a throttle condition of 1200 psig. and 950 F and a 9375 Kw unit with one stage feedwater heating. The results for each throttle condition and turbine with varying heater stages are tabulated in part C of the investigation.

V

THE INVESTIGATION

A. Object

The object of this investigation is to determine the indicated savings for the average heating season of 1966-1967, resulting from the installation of a selected back-pressure turbo-generator unit in the Virginia Polytechnic Institute Heating and Power Plant.

B. Procedure

This investigation consisted of the collection, analysis, and projection of data concerning the various steam and electric loads on the Virginia Polytechnic Institute Heating and Power Plant to determine the savings that might be obtained during a heating season by the installation of a back-pressure turbo-generator unit exhausting at 15 psig. In order to determine the most efficient size turbine to install and the most efficient cycle to use, five standard steam pressures and temperatures were used along with one, two, and three stage feedwater heating cycles. The steam pressures and temperatures used were: (1) 250 psig. and 500 F, (2) 400 psig. and 750 F, (3) 600 psig. and 825 F, (4) 900 psig. and 900 F, (5) 1200 psig. and 950 F.

The heating season of 1952-1953 was chosen as the base heating season because the plant records on this year were available and compared favorably with the 51-year average outdoor temperatures. The heating season is considered to be 180 days in length and runs from October through March. The hourly low pressure and high pressure steam demand was averaged for the 180 days and a typical heating season day steam load curve was plotted. It was found that the high pressure 75 psig. steam load remained fairly constant for the 24 hour period while the variation in steam load came from the low pressure steam demand. The hourly electric load, which is the sum of the electric power generated in the Heating and Power Plant and the electric power purchased from the Appalachian Electric Power Company, was averaged for the 180 day period and an electric load curve was plotted for a typical heating season day.

The average electric load was projected into the future at a yearly 12.5 per cent increase. The yearly increase of 12.5 per cent was decided upon after discussion with the college utility engineer and a study of local and national expected load increases. The low and high pressure steam demands were projected into the future in proportion to the expected increase in college enrollment for the year 1966-1967, over the enrollment of 1952-1953. This expansion factor came out to be 2.14. The information on the future college enrollment of Virginia Polytechnic Institute came from an investigation made by Dr. P. H. Farrier of this institute. After a discussion with the college utility engineer, it was felt that the college facilities would possibly not increase in proportion with the expected future enrollment due to financial problems which might arise, so another expansion factor of 1.50 was used to project the steam loads from 1952-1953 to 1966-1967. It was assumed the electric load increase would remain the same as mentioned. This now gave two possible future steam loads from which to determine the possible electric power production.

In this investigation several facts must be kept in mind. The electric power produced is controlled directly by the demand for process and heating steam. There is both a high and a low pressure steam demand. The high pressure steam is at 75 psig. and the low pressure at 15 psig. There must be at least one stage of feedwater heating for deaeration purposes in the plant cycle.

The next step in the investigation was to calculate performance curves on a selected turbo-generator unit for each of the five throttle conditions. Calculations were made to determine the steam rate, the extraction factor, and the amount of exhaust flow for a standard turbo-generator unit. These values were calculated at 40 per cent and 80 per cent loading of the generator. The 40 and 80 per cent generator loading points were recommended by General Electric for plotting performance curves on steam turbines.[12]

The extraction point used was 75 psig. and the quantity of steam extracted was assumed to remain constant for all turbine loadings. A heat balance was calculated for a deaerator heater operating off the exhaust line of the turbine. The quantity of steam necessary for deaeration was subtracted from the exhaust flow at the 40 per cent and 80 per cent load points. From these values two straight line curves were drawn; a steam rate curve and a corresponding exhaust flow curve. In order to select the size turbine for each throttle condition, several different size turbines were tried and the one which operated at the highest efficiency and load with the expected exhaust steam demand was selected. Since only the average steam load was taken, a 10 per cent overload was allowed for in the selection of each turbine.

For each turbo-generator selected a second feedwater heating stage was added. This heater will be of the surface type and will be fed from the 75 psig. extraction line. A heat balance was calculated around the deaerator and the second heater to determine the new steam rate curves and the exhaust flow available. In these calculations a five degree terminal difference was used for the surface heater and a 10 per cent steam pressure drop from the turbine to the heater was assumed. These values were also recommended by General Electric for standard feedwater heater performance.[12]

The same procedure as above was used in calculating the effects of a third stage feedwater heater of the surface type. In order to determine the extraction pressure of the third heater, it was given the same feedwater temperature rise as the second heater using a five degree terminal difference and a 10 per cent steam pressure drop from the turbine to the heater. In both cases the drains from the surface heaters were cascaded back to the deaerator. This method of handling drains eliminates the use of drain pumps.

From the steam demand, electric load, and the different turbo-generator performance curves, a set of four curves were plotted for the average heating season day. Curve No. 1 shows the expected total electric load. Curve No. 2 shows the electric power that would be generated with one stage feedwater heating while extracting and exhausting the process and heating steam. Curve No. 3 shows the electric power that would be generated with two stage feedwater heating while extracting and exhausting the process and heating steam.

The next step in the investigation was to determine the kilowatt loads. This was done by measuring the areas under these curves. The area under Curve No. 1 represents the total electric load in Kw-hrs on the system for a day. The areas under Curves 2, 3, and 4 represent the total electric power in Kw-hrs that could be generated by the proposed unit. The scale constant used on these curves was 1000 Kw-hrs per square inch. In cases where the generated power curve was higher than the electric load curve, it was considered dump power and the area between these two curves was subtracted from the total generated power. Dump power is electric energy generated in excess to the electric load. Since it is not known whether this dump power can be sold or not, it was not considered in the calculation of the daily savings.

The total electric power generated in each case was multiplied by \$0.008 per Kw-hr minus the fuel cost per Kw-hr. The value \$0.008 is the minimum rate of energy paid by the college for purchased power to the local utility. To calculate the fuel cost per kilowatt hour, the average steam load per hour was used. The fuel was considered to have a heating value of 11,500 B.T.U. per lb. and a cost of \$2.50 per ton. The value 11,500 B.T.U. per lb. was taken from a fuel analysis of the local coal used for steam generation in the plant. The cost of this fuel at present is \$2.10 per ton and from this value a cost of \$2.50 per ton was assumed for the heating season of 1966-1967. A boiler efficiency of 80 per cent was assumed in the fuel cost calculations. In all probability the boiler efficiency will be higher than 80 per cent, but considering the present boiler efficiencies of 70 to 78 per cent, the low grade coal that is burned, and the wide range of operation the value of 80 per cent was thought to be a safe assumption.

The value obtained by multiplying the cost per Kw-hr times the generated load would be the savings that could be expected from the operation of the selected turbo-generator unit for one day during the heating season of 1966-1967. The total savings for the heating season were calculated along with the additional savings encountered by using one, two, and three stage feedwater heating.

The cost of labor and maintenance was not figured in the savings. It will not require any additional labor to operate such a unit. The same number of personnel are required to operate the plant whether it produces electric power or not. It has been found that the yearly cost of maintenance on such a unit is very small and therefore it was neglected.

No attempt will be made in this investigation to select one definite turbo-generator as the most economical unit since this will involve many factors which cannot be covered in the scope of this thesis. These factors are such things as initial cost, additional cost of extraction points, installation cost, cost of feedwater heaters, cost of high pressure piping, etc.

C. Results

Indicated Savings For The Heating Season of 1966-1967 Based

On The Cost Of Purchased Power

Throttle Conditions	Turbine Size	One Stage Feedwater Heating	Additional Saving With Two Stages	Additional Saving With Three Stages
Expansion Factor 2.14				
250 psig.	5000 KW	$130,757.00		
400 psig.	7500 KW	188,440.00	$ 6,489.00	$4,997.00
600 psig.	9375 KW	212,095.00	10,952.00	3,816.00
900 psig.	9375 KW	234,580.00	9,420.00	4,731.00
1200 psig.	9375 KW	249.480.00		
Expansion Factor 1.50				
250 psig.	3750 KW	$ 89,100.00		
400 psig.	5000 KW	131,299.00	$ 9,121.00	$6,311.00
600 psig.	6250 KW	151,738.00	8,642.00	5,130.00
900 psig.	7500 KW	171,180.00	8,307.00	6,068.00
1200 psig.	9375 KW	181,710.00	7,440.00	6,547.00

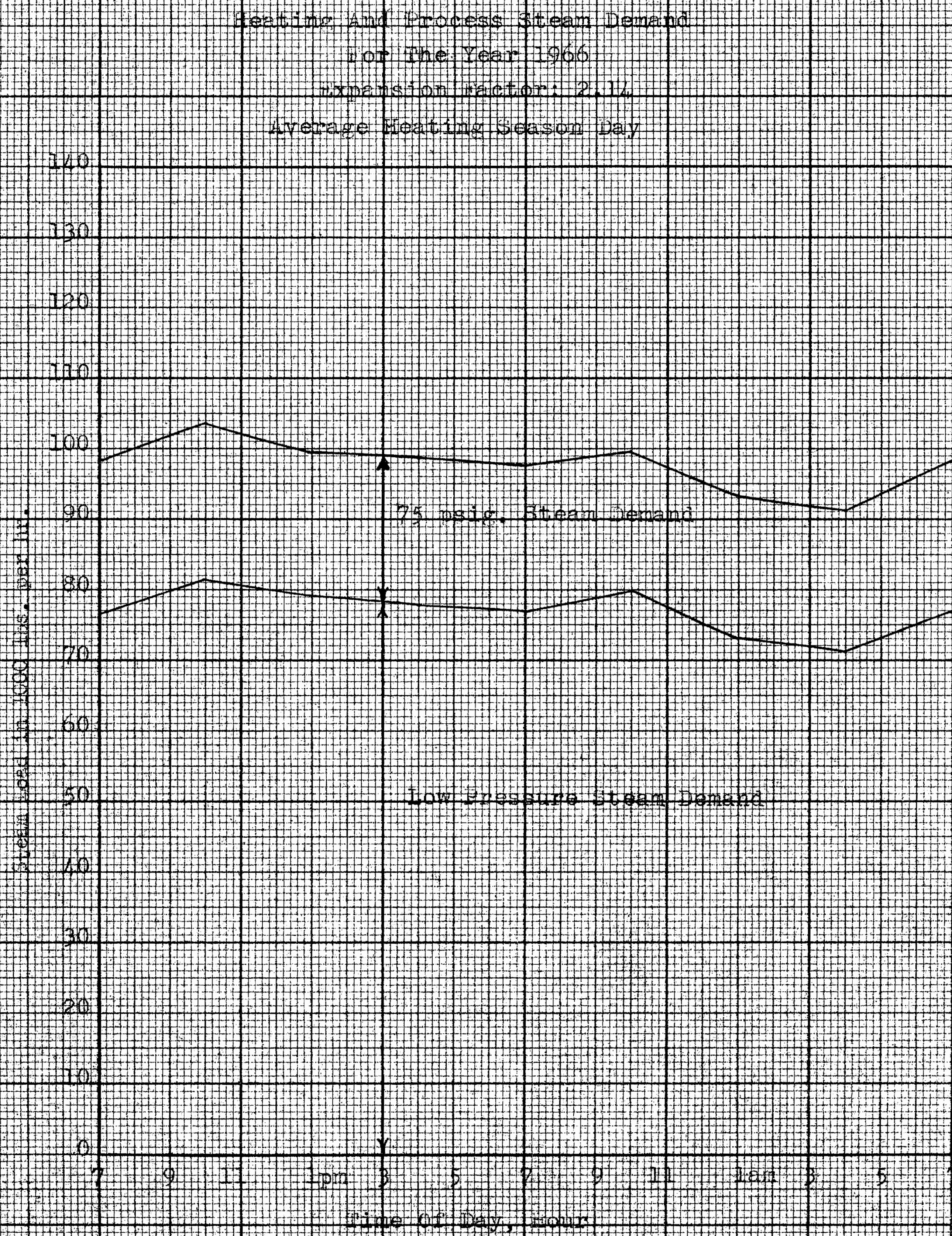


Fig. 1 Load Chart

D.H.W. 3/23/52

Heating And Process Steam Demand
For The Year 1966
Expansion Factor: 1.50
Average Heating Season Day

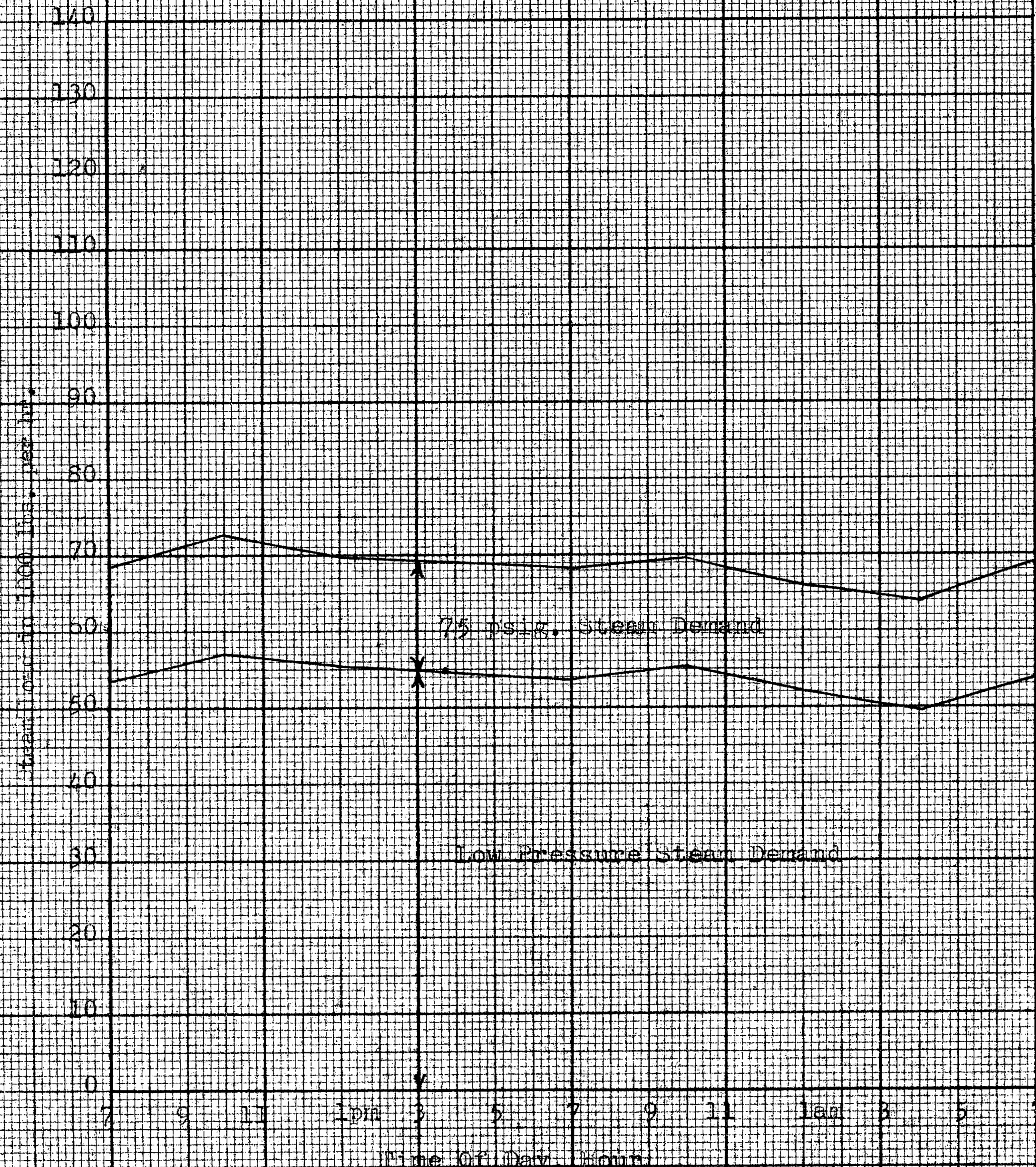


Fig. 2 Load Chart

D.H.W.

3/23/54

Throttle Conditions: 250 psig. & 500 F.T.T.
Unit: 5000 KW Max. Rated - Exhaust: 15 psig.
Extraction: 75 psig.
One Stage Feedwater Heating

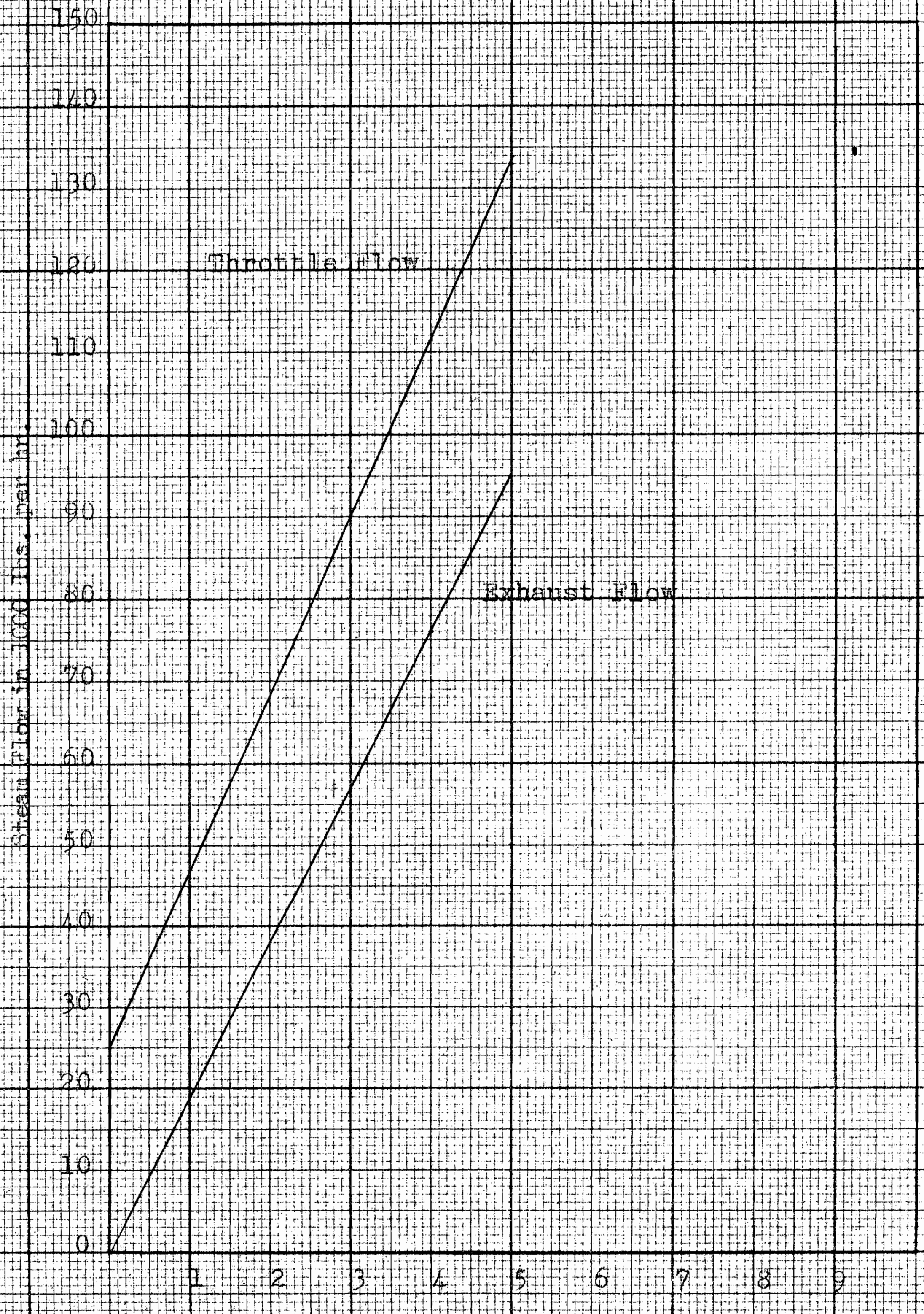


Fig. 3 Performance Curves(Expansion Factor 2.14)

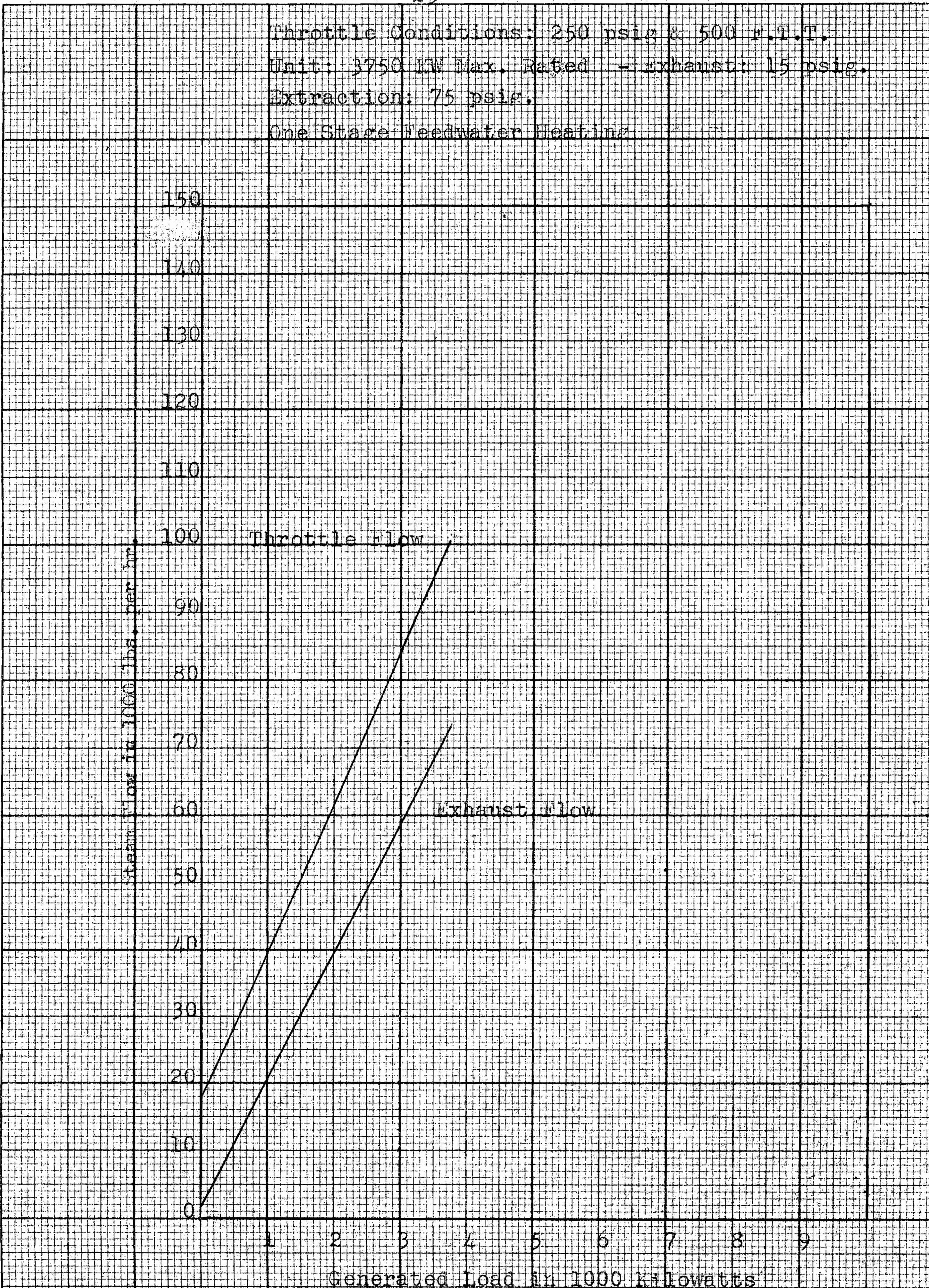


Fig. 4 Performance Curves (Expansion Factor 1,50)

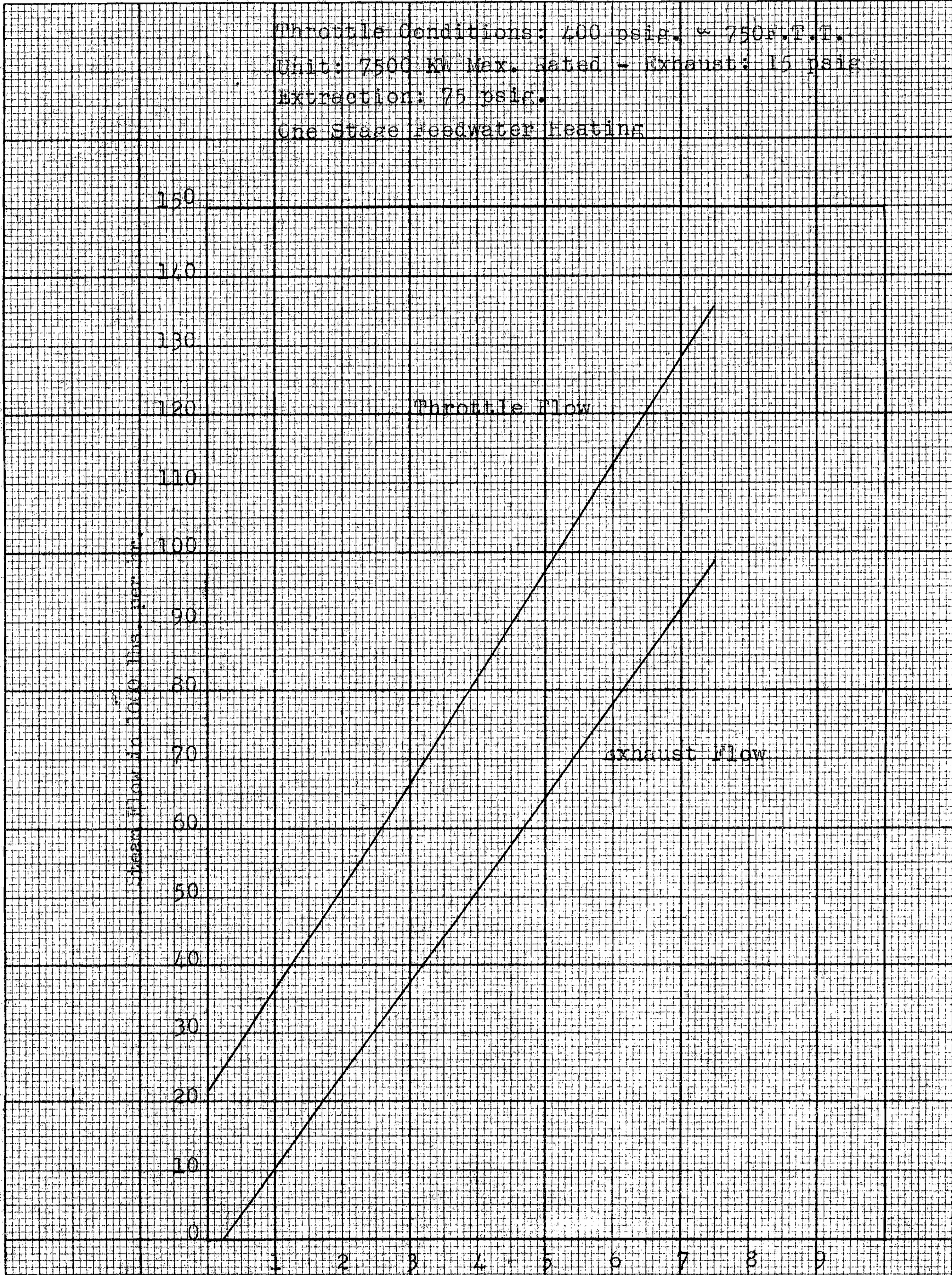


Fig. 5 Performance Curves (Expansion Factor 2.14)

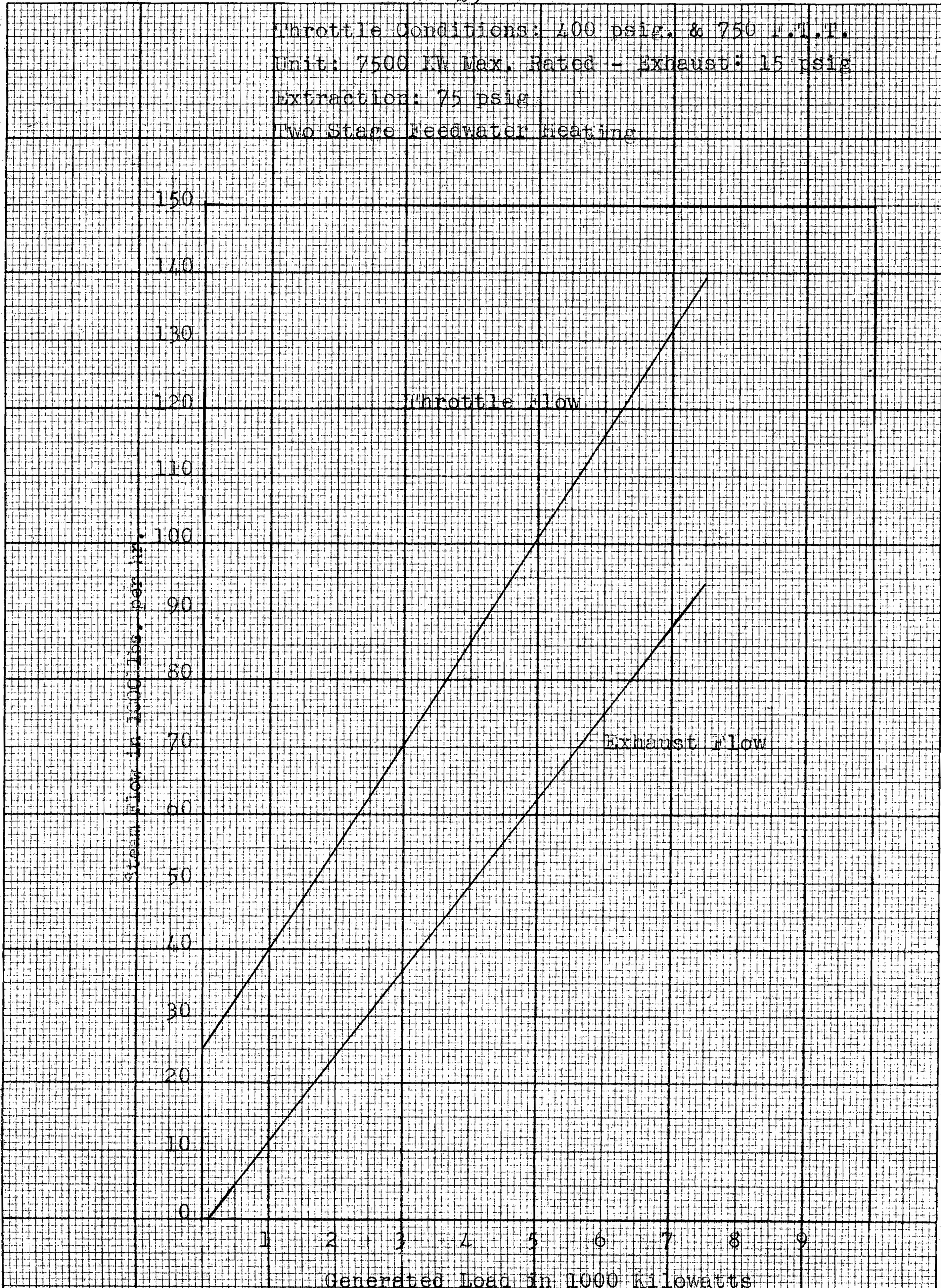


Fig. 6 Performance Curves (Expansion Factor 2.14)

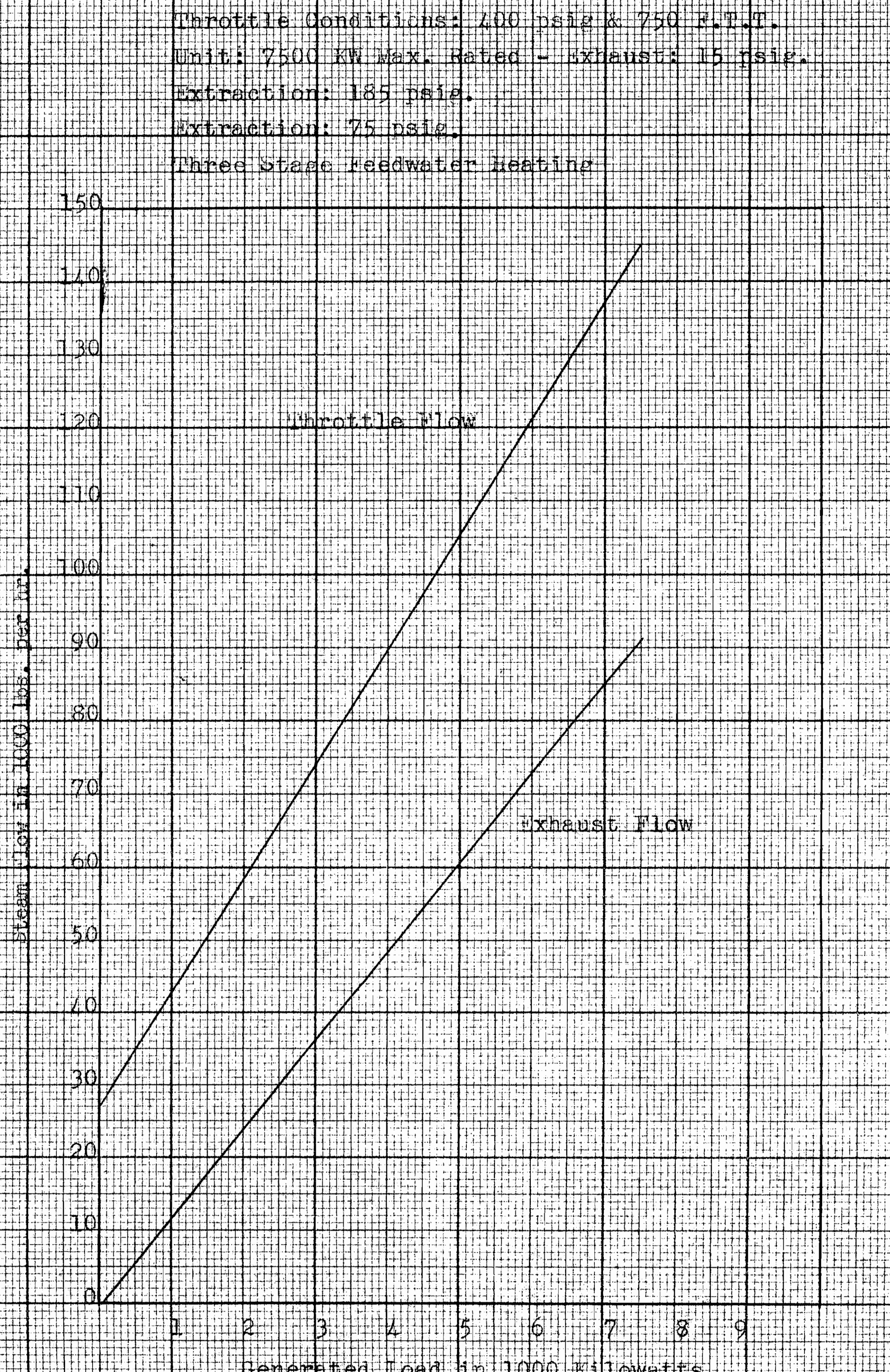


Fig. 7   Performance Curves (Expansion Factor 2.14)

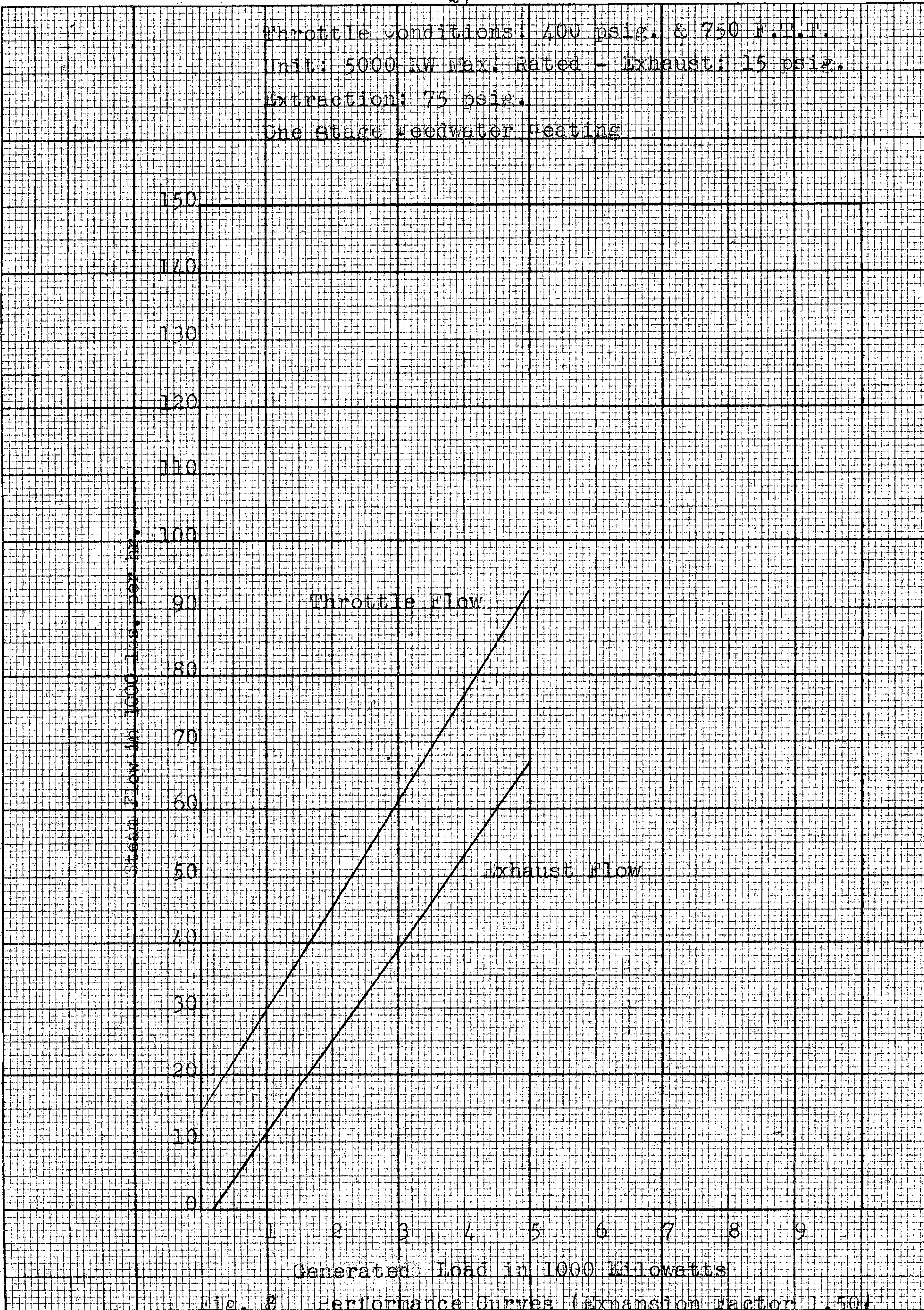


Fig. 8 Performance Curves (Expansion Factor 1.50)

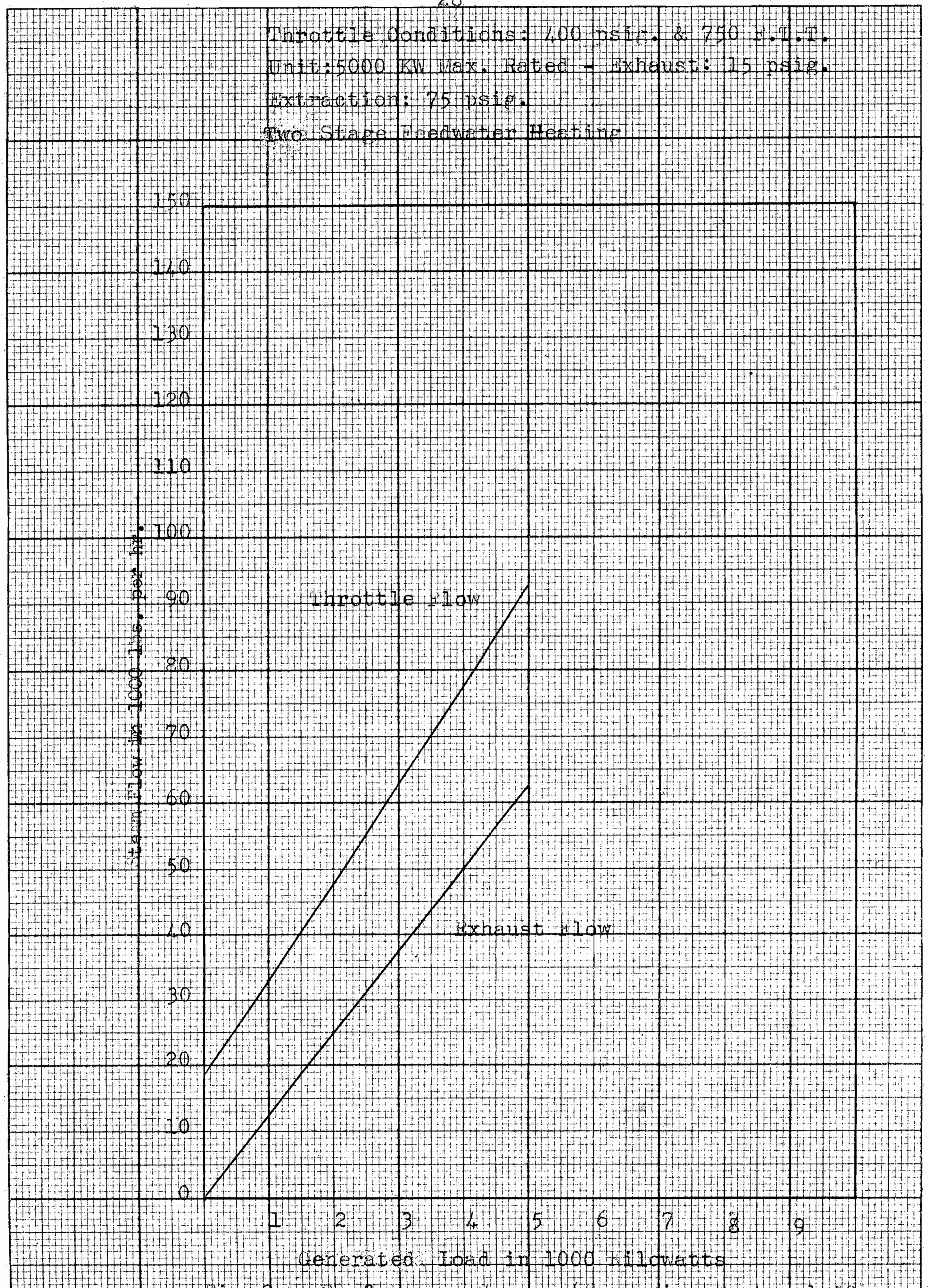


Fig. 9 Performance Curves (Expansion Factor 1.50)

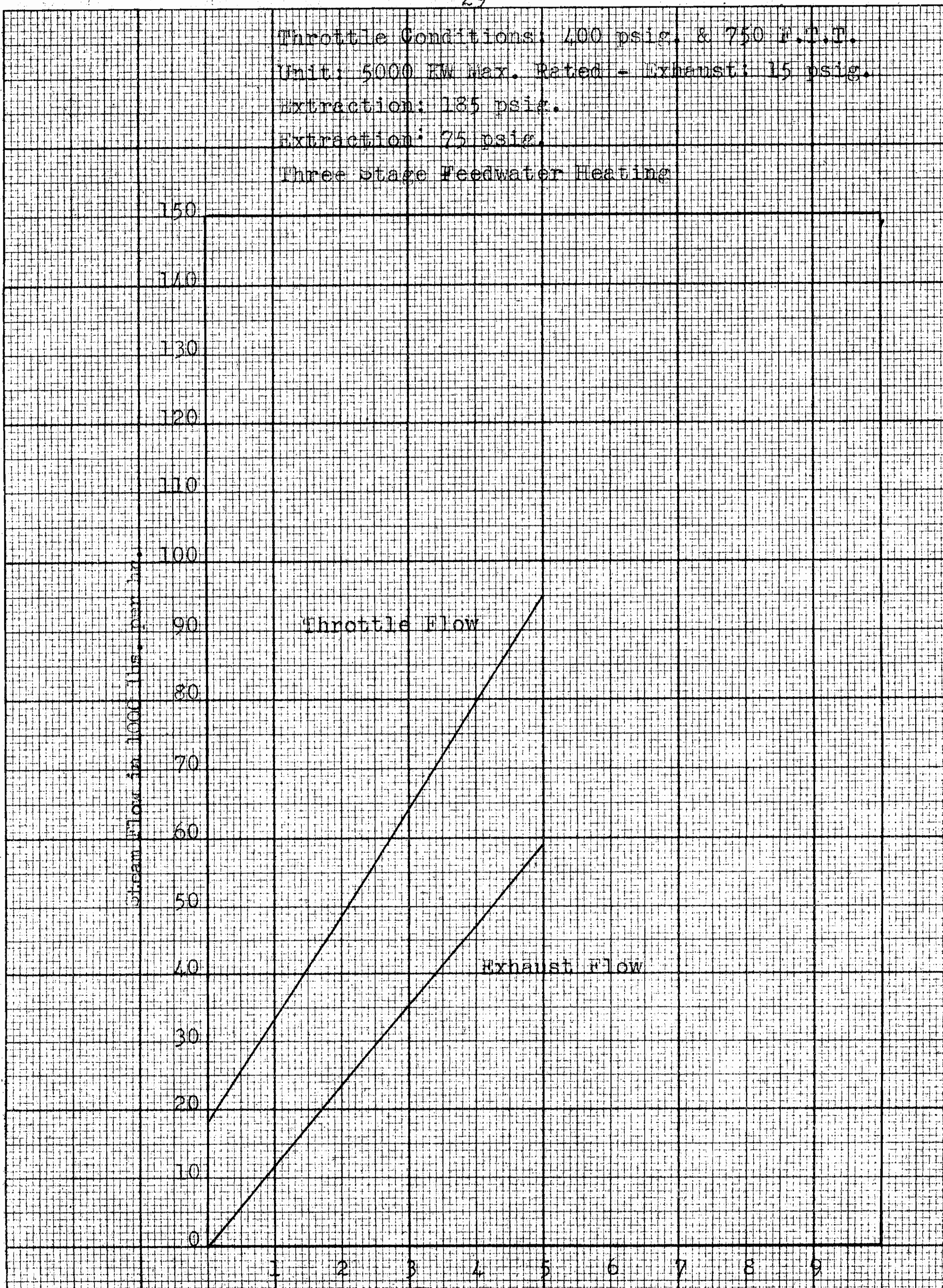


Fig. 10 Performance Curves (Expansion Factor 1.50)

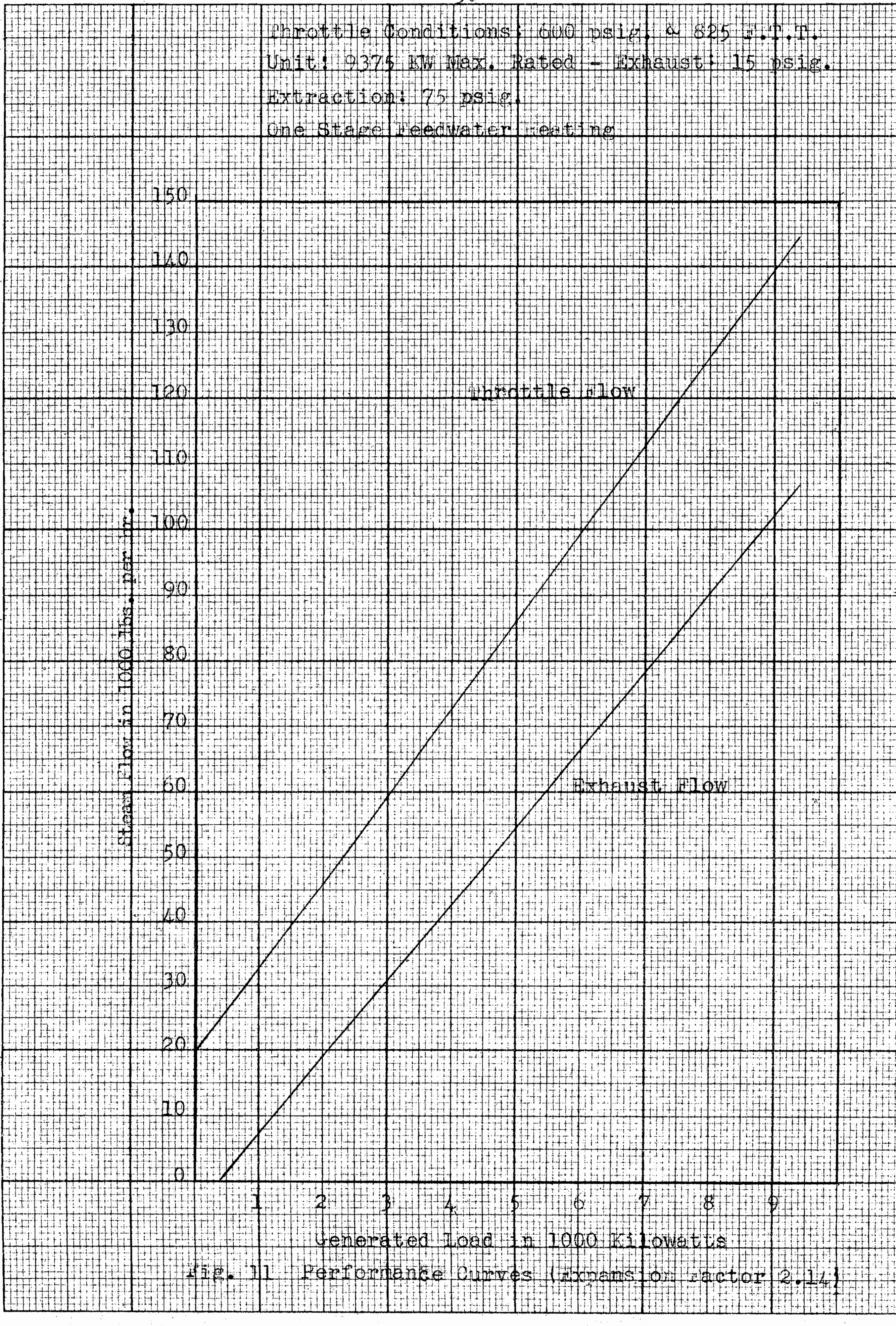


Fig. 11 Performance Curves (Expansion Factor 2.14)

Throttle Conditions: 600 psig. & 825 F.T.T.
Unit: 9375 KW Max. Rated - Exhaust: 15 psig.
Extraction: 75 psig.
Two Stage Feedwater Heating

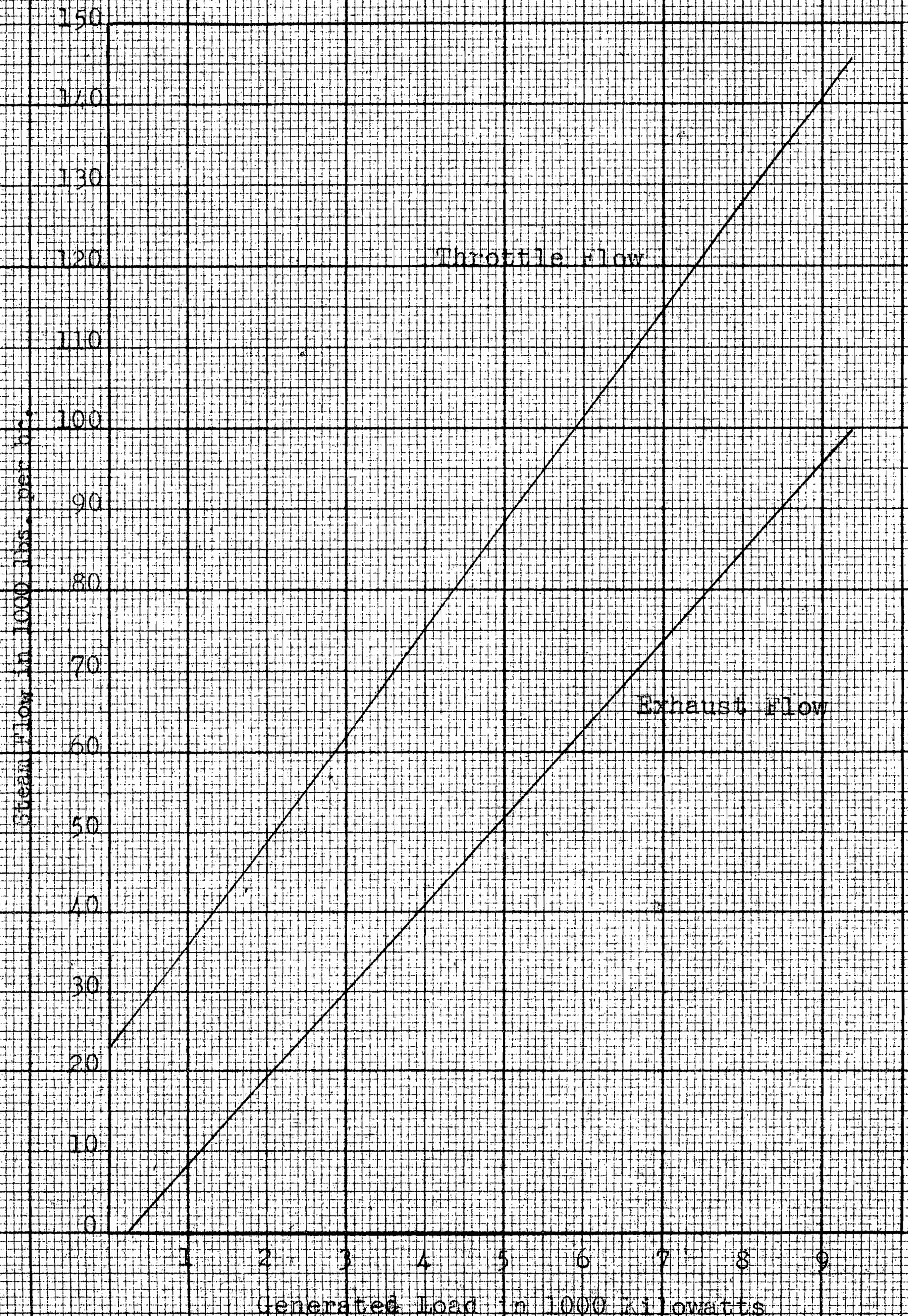


Fig. 12 Performance Curves (Expansion Factor 2.14)

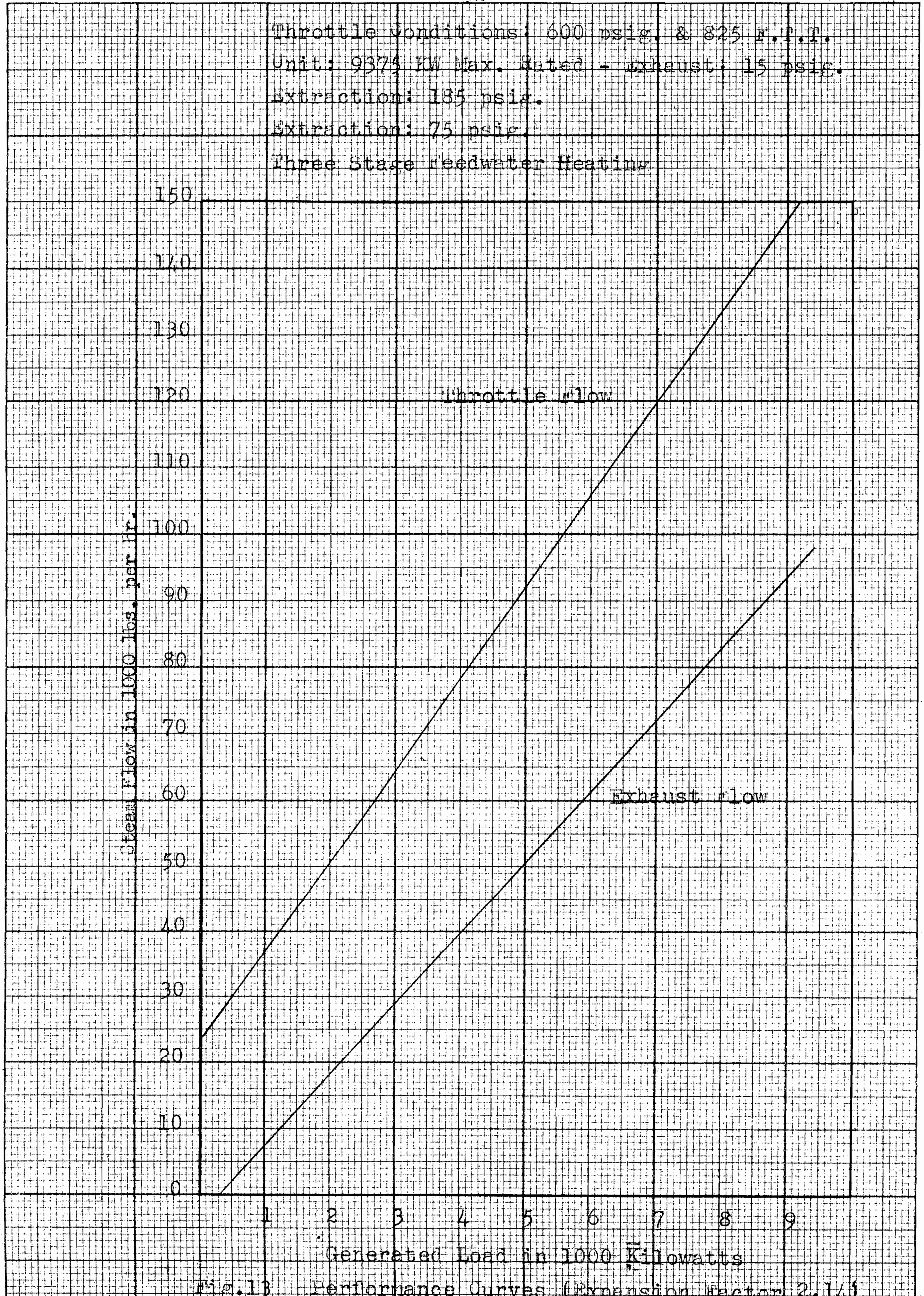


Fig.13 Performance Curves (Expansion Factor 2.14)

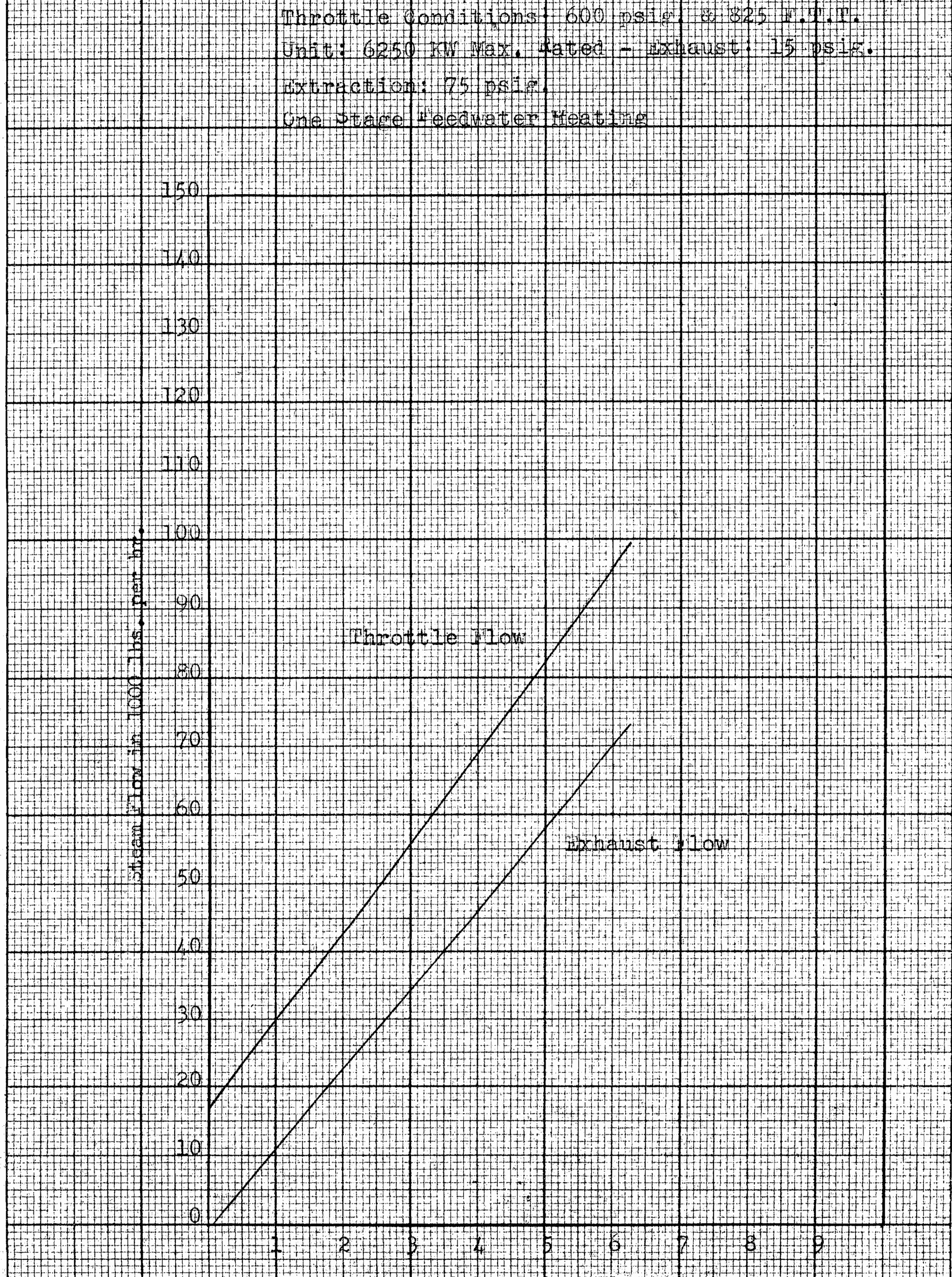


Fig. 14 Performance Curves (Expansion Factor 1.50)

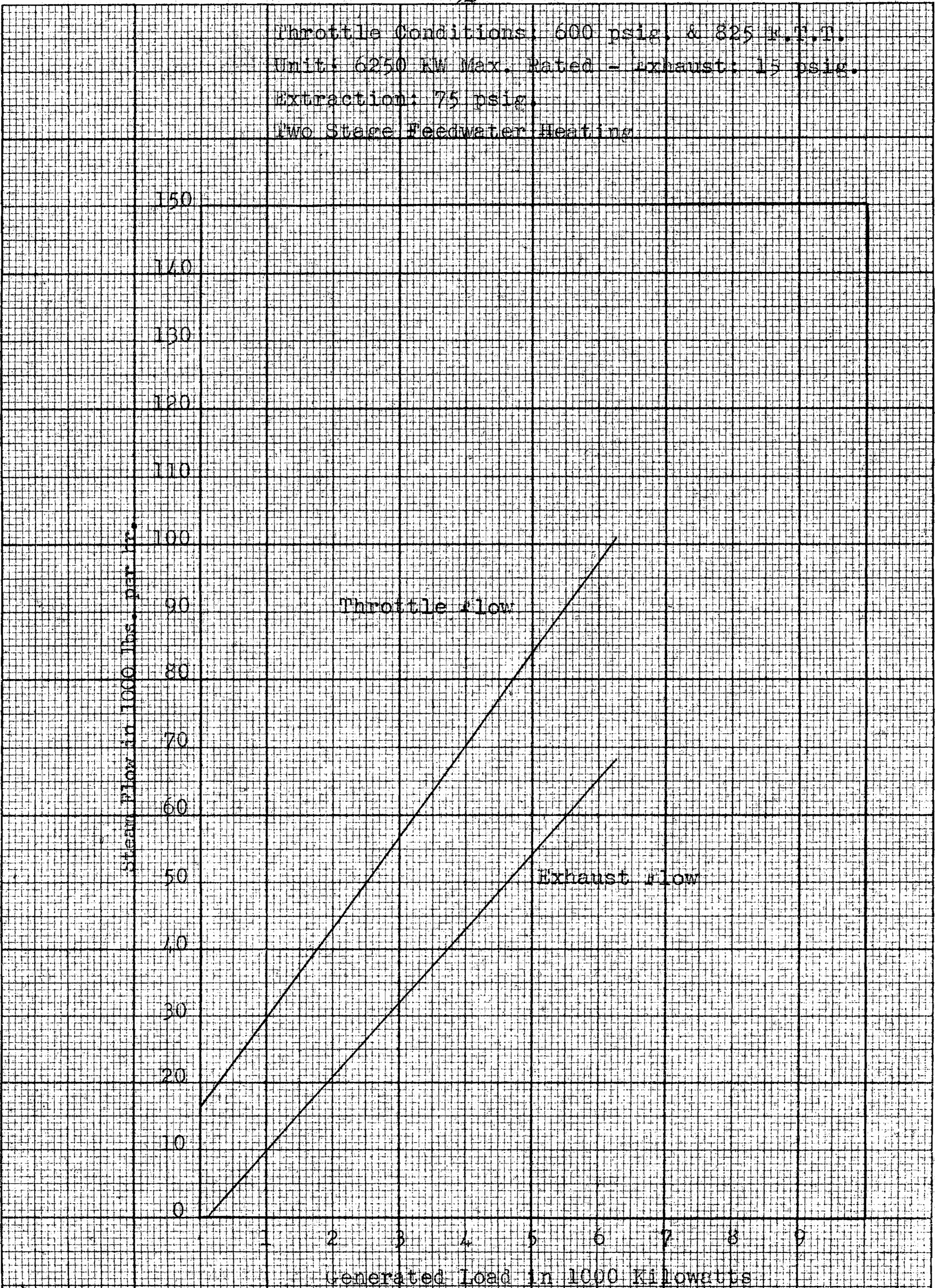


Fig. 15 Performance Curves (Expansion Factor 1.50)

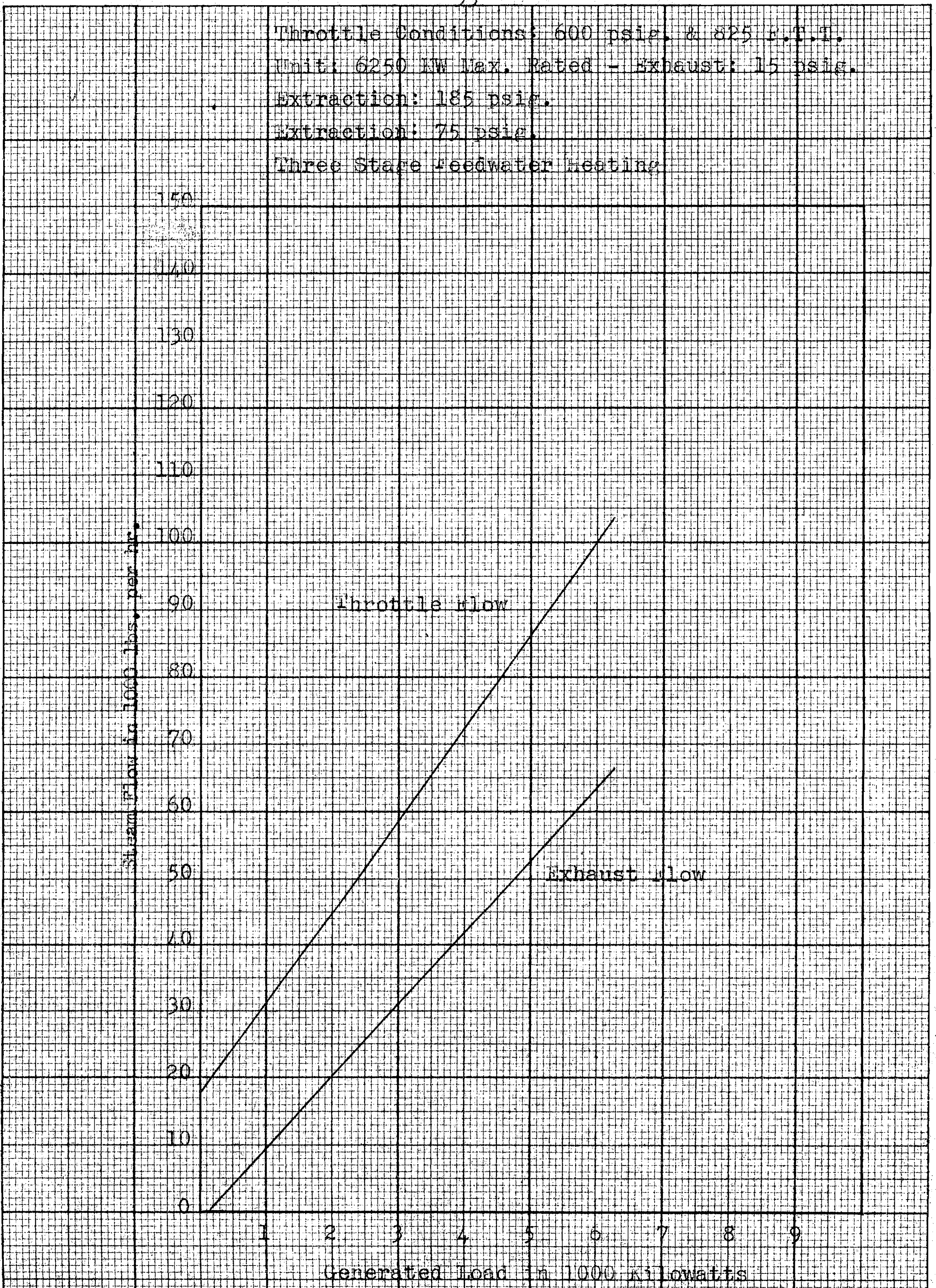


Fig. 16 Performance Curves (Expansion Factor 1.50)

Throttle Conditions: 900 psig. & 900 F.T.T.
Unit: 9375 KW Max. Rated - Exhaust: 15 psig.
Extraction: 75 psig.
One Stage Feedwater Heating

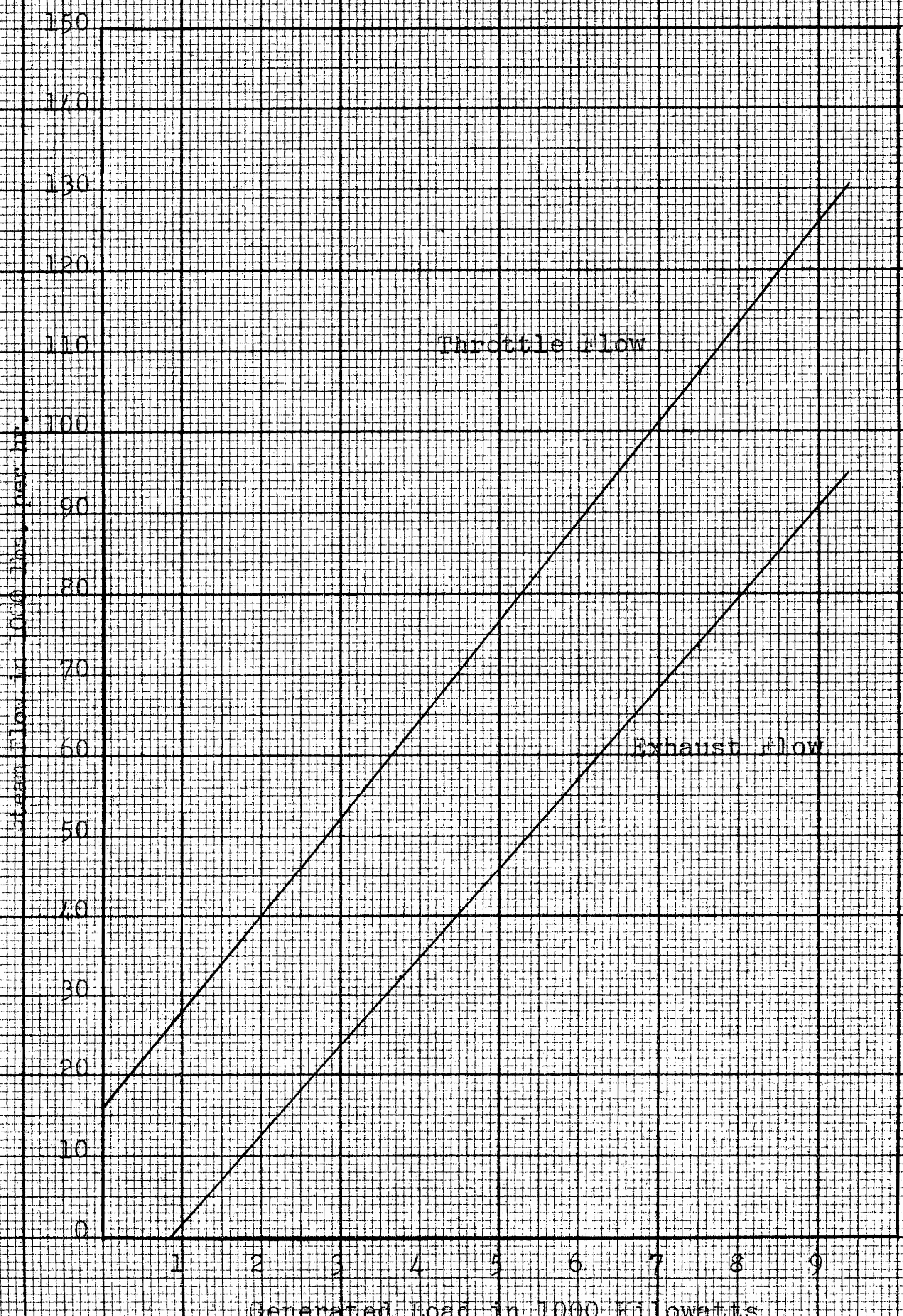


Fig.17 Performance Curves (Expansion Factor 2.14)

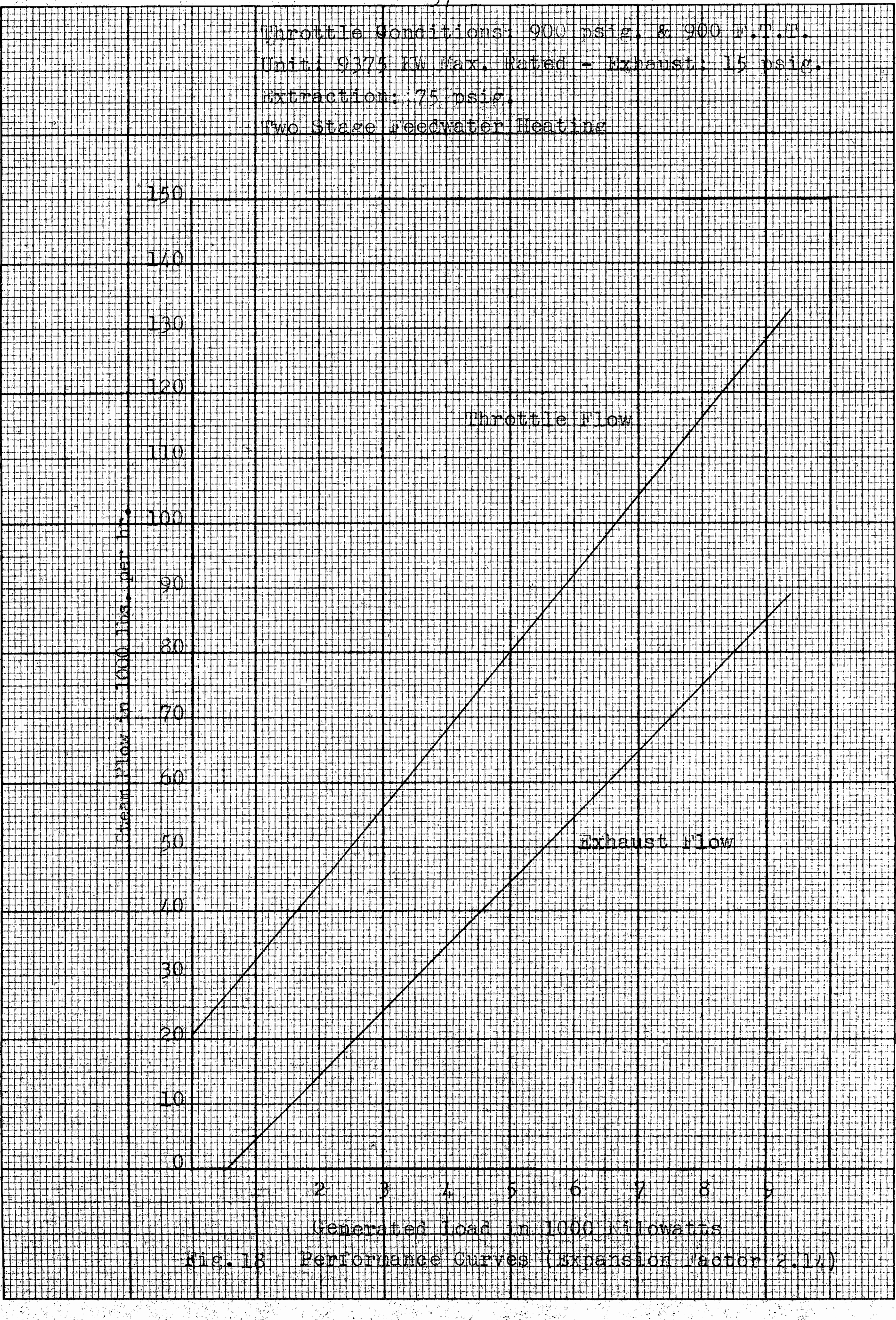


Fig. 18 Performance Curves (Expansion Factor 2.14)

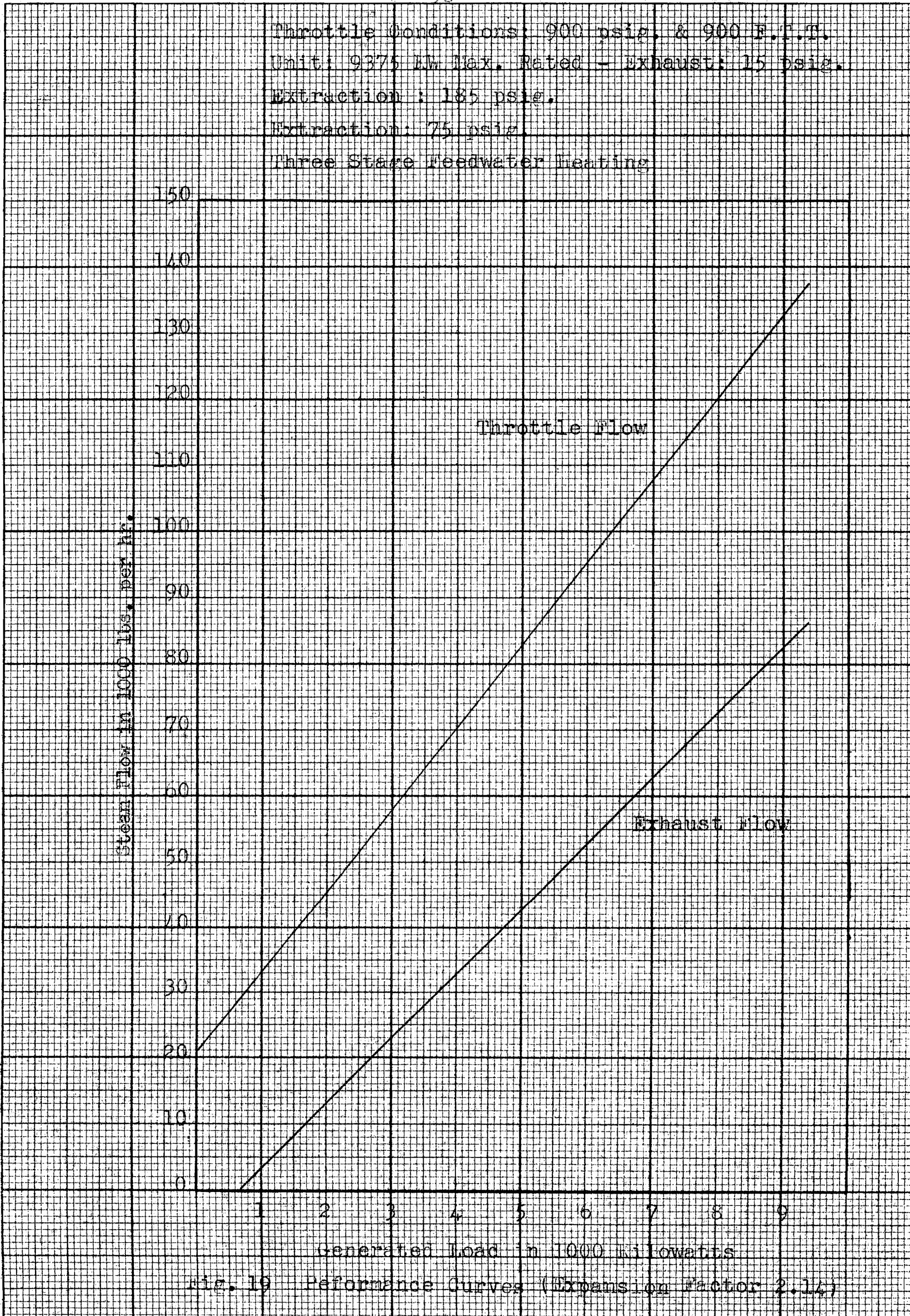


Fig. 19 Peformance Curves (Expansion Factor 2.14)

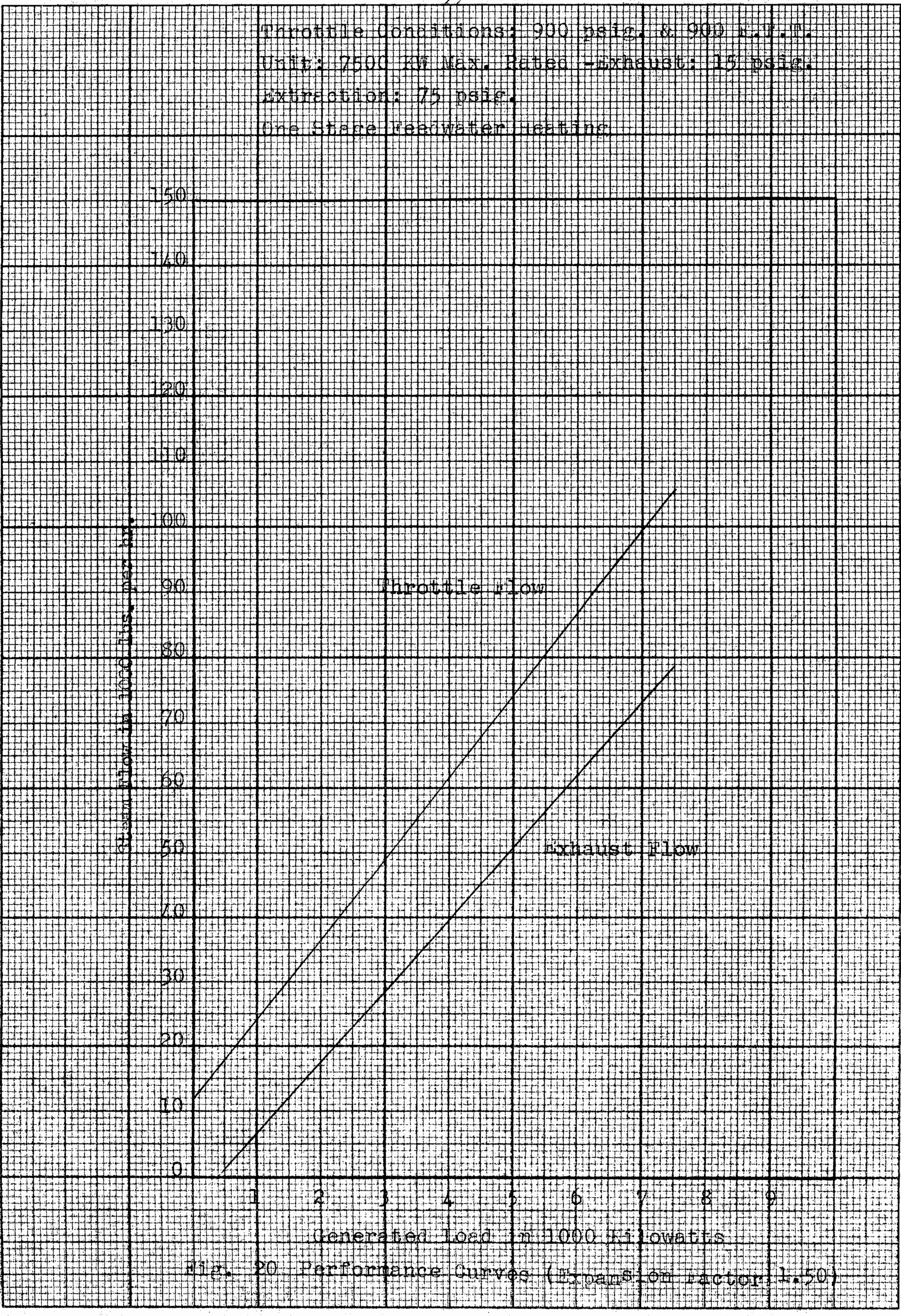


Fig. 20 Performance Curves (Expansion Factor 1.50)

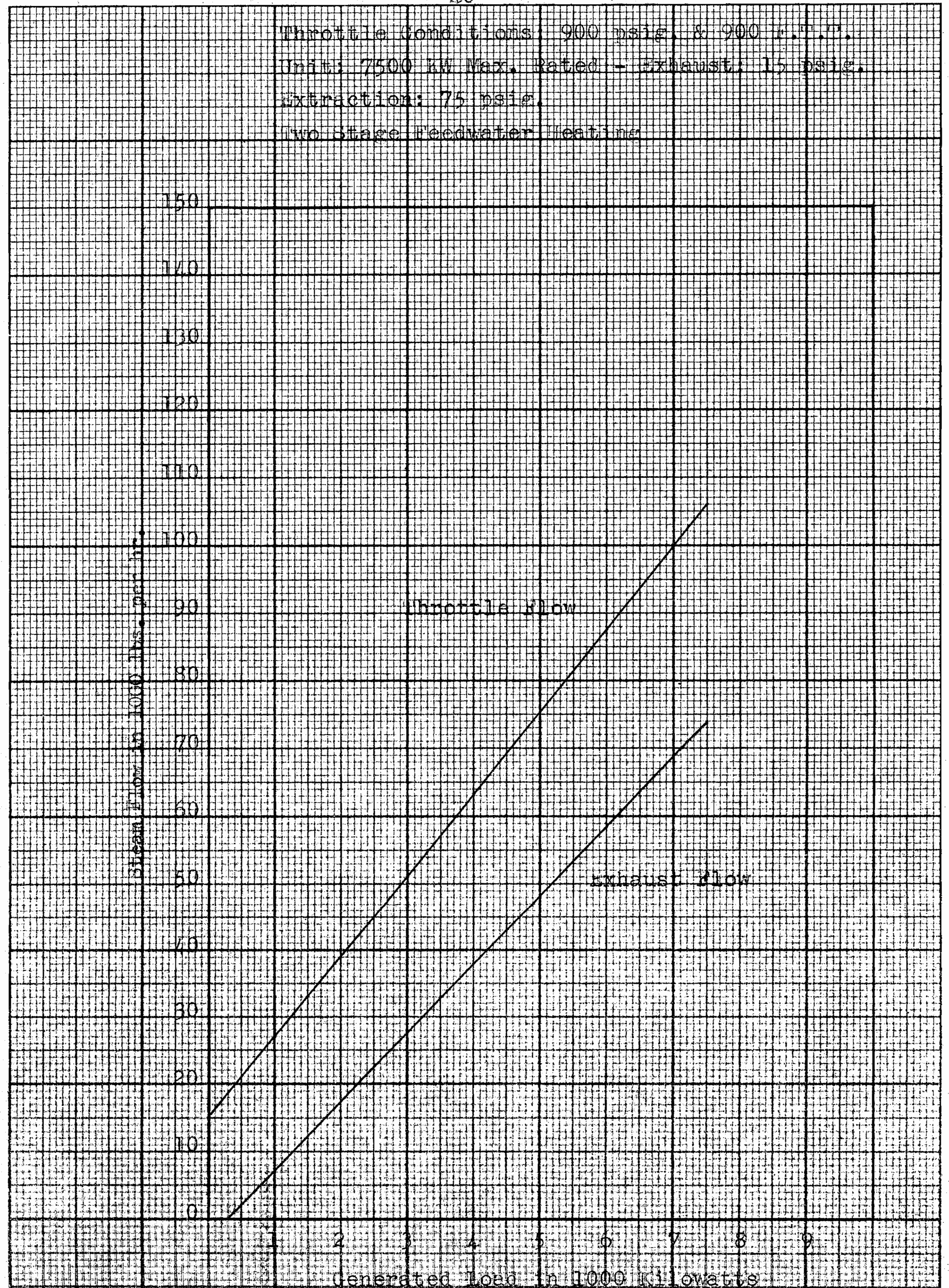


Fig. 21 Performance Curves (Expansion Factor 1.50)

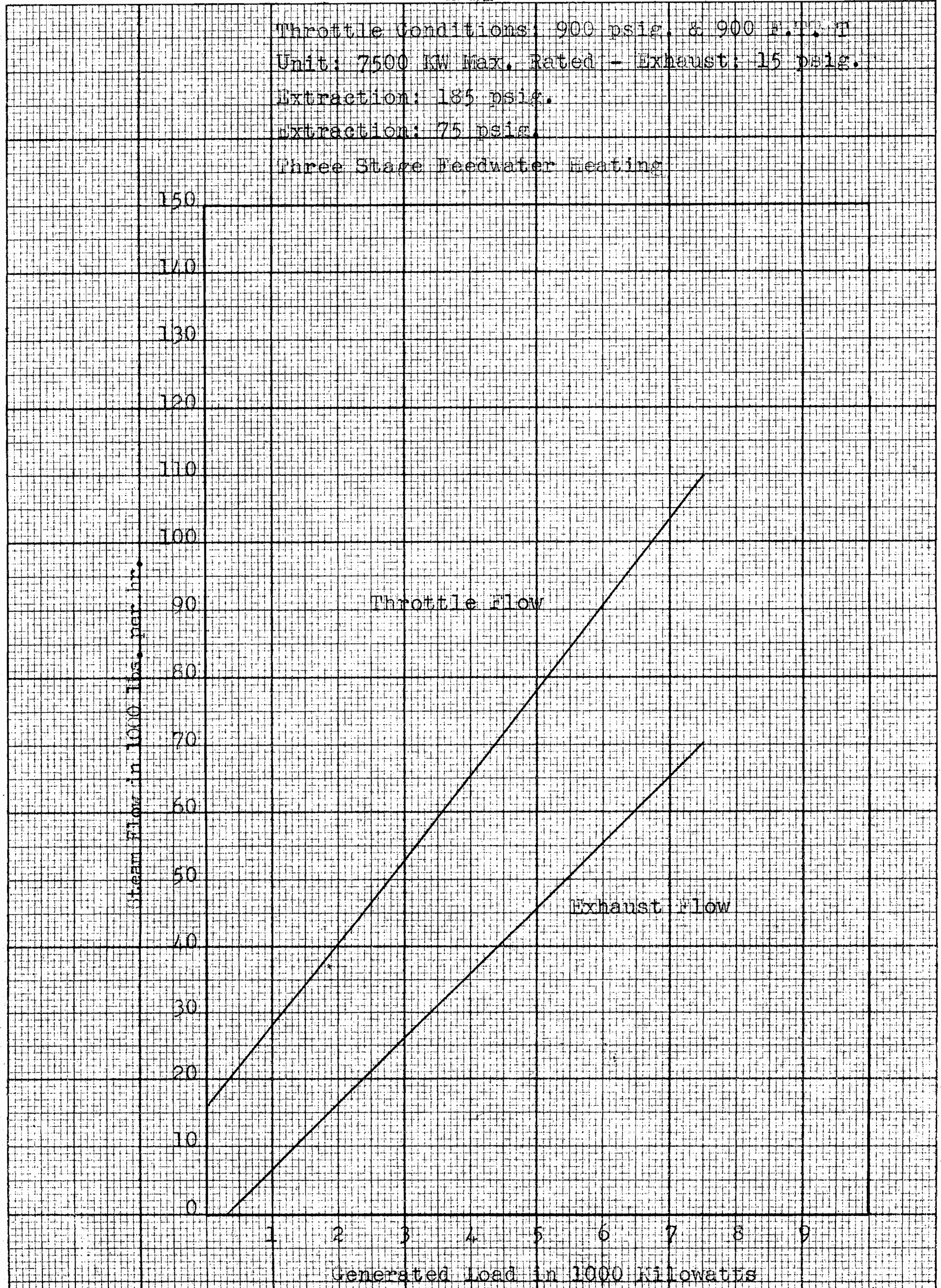


Fig. 22 Performance Curves (Expansion factor 1.50)

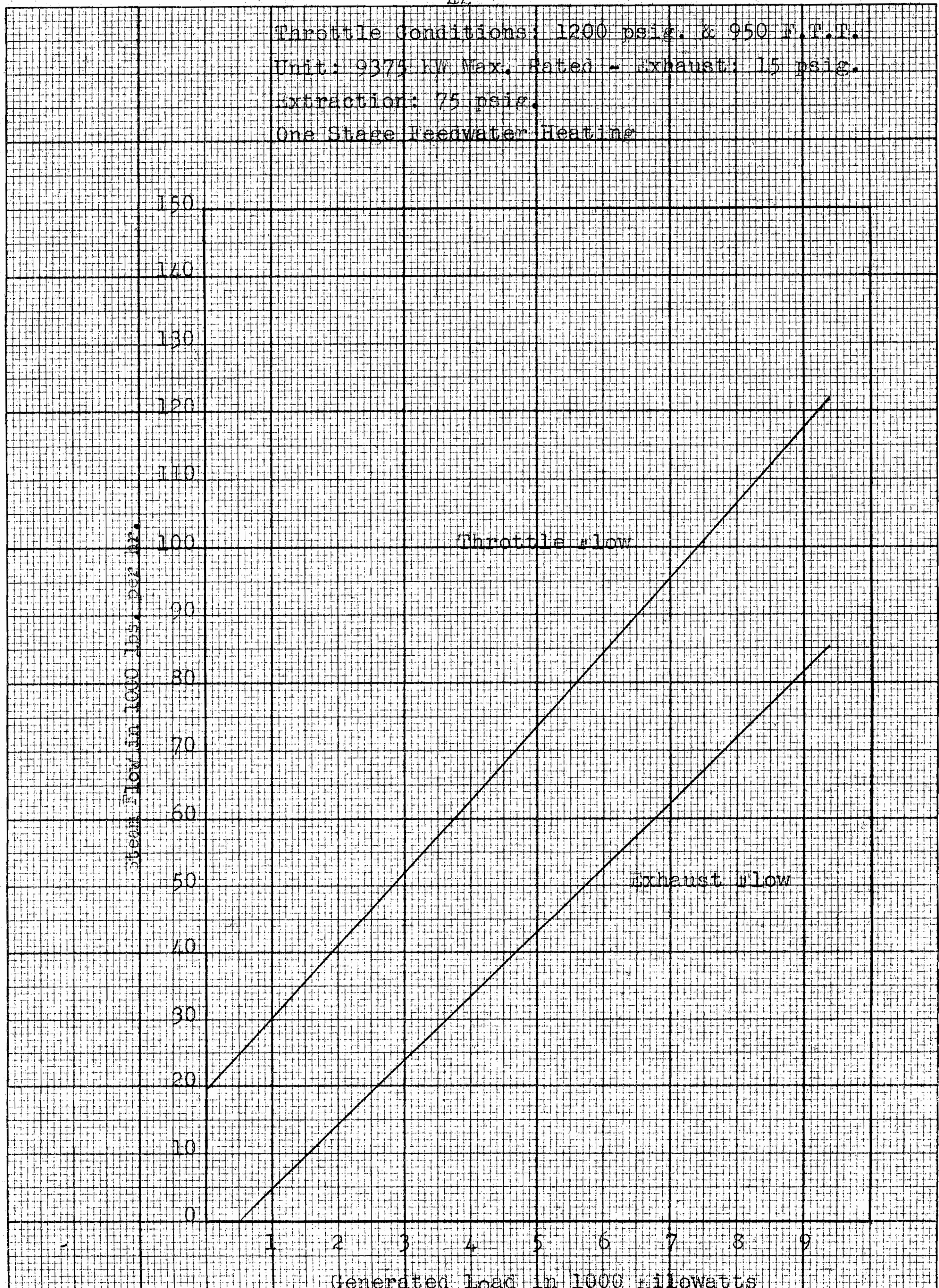


Fig. 23 Performance Curves (Expansion Factor 2.14)

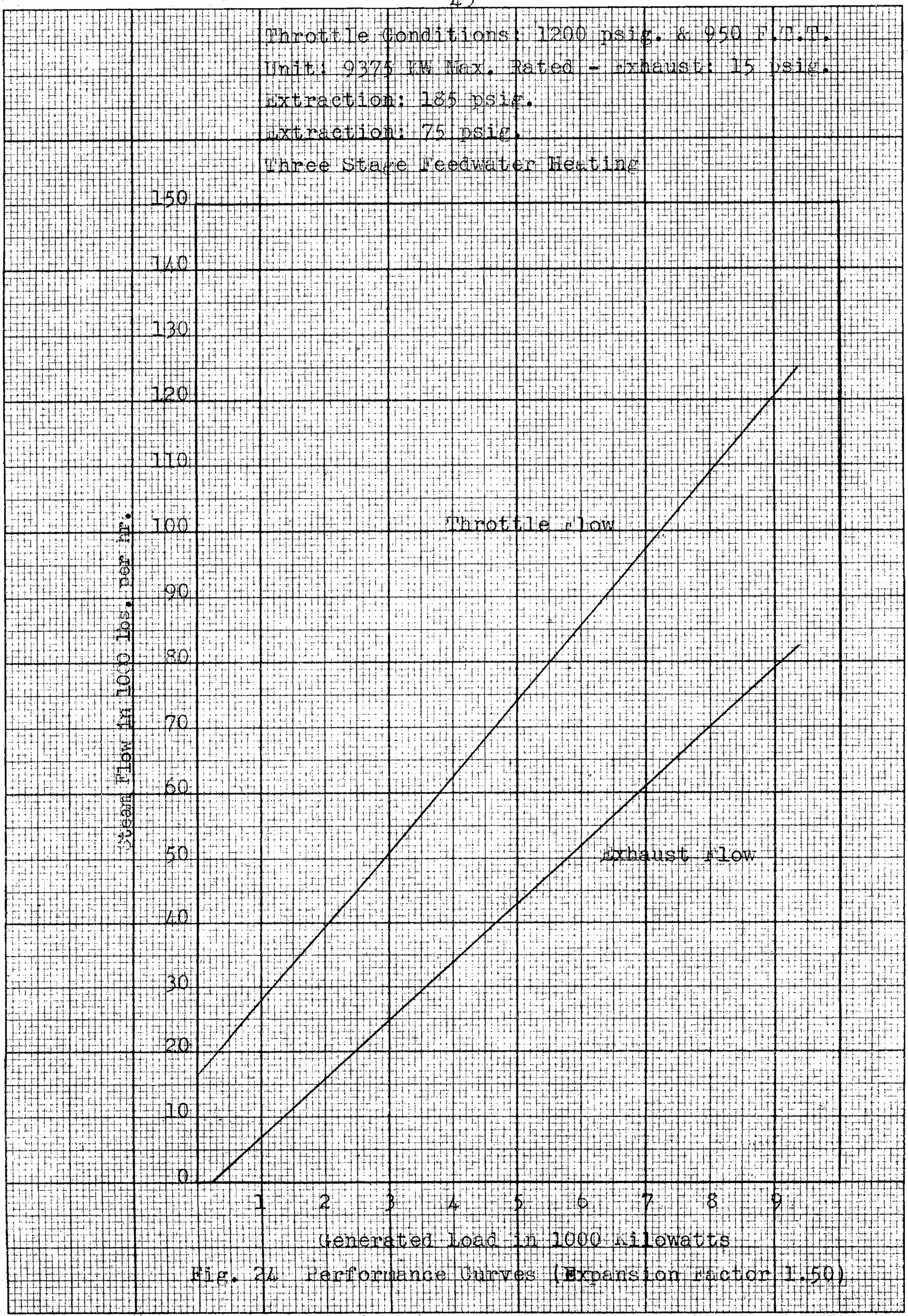


Fig. 24 Performance Curves (Expansion Factor 1.50)

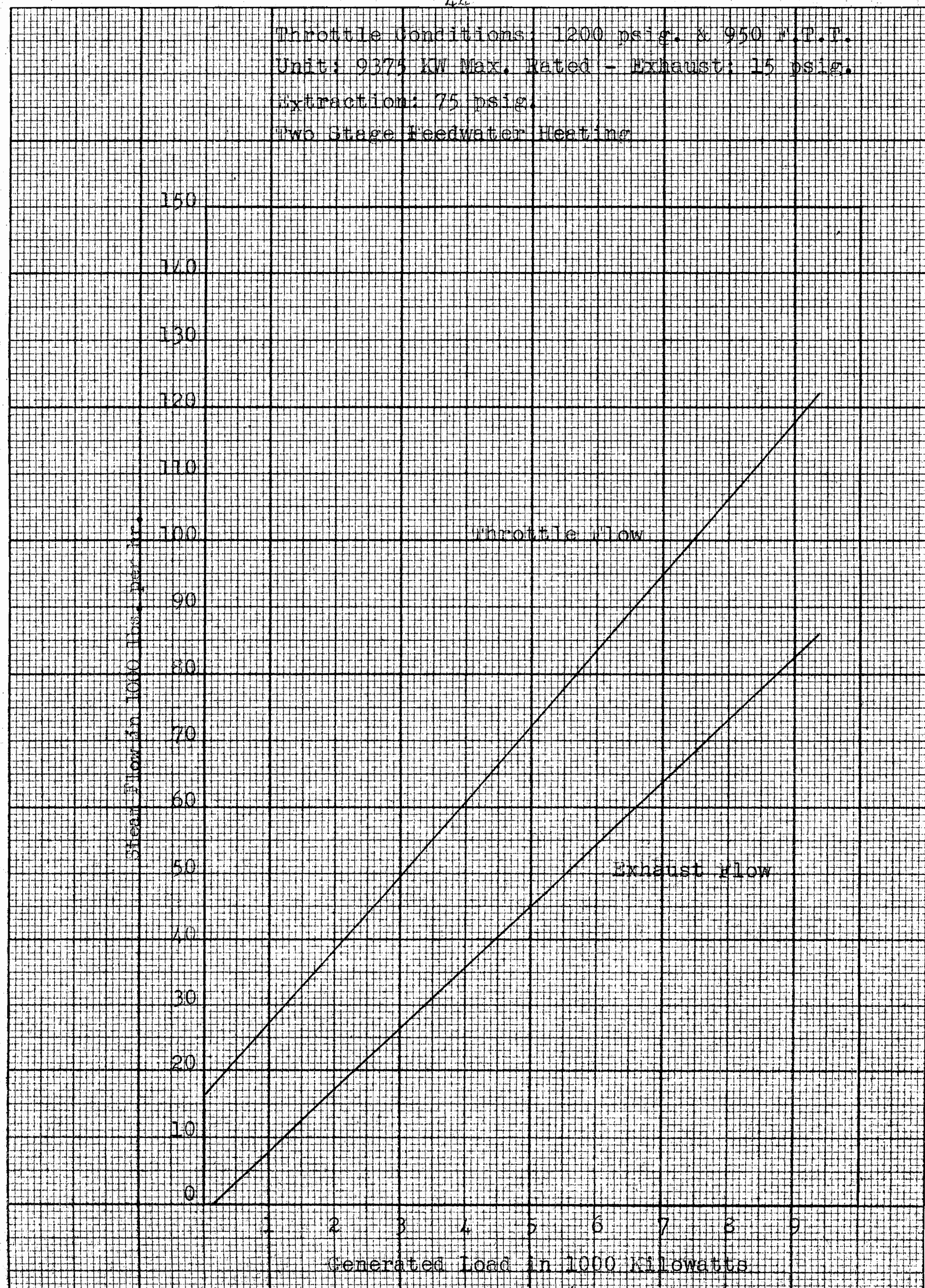


Fig. 25 Performance Curves (Expansion Factor 1.50)

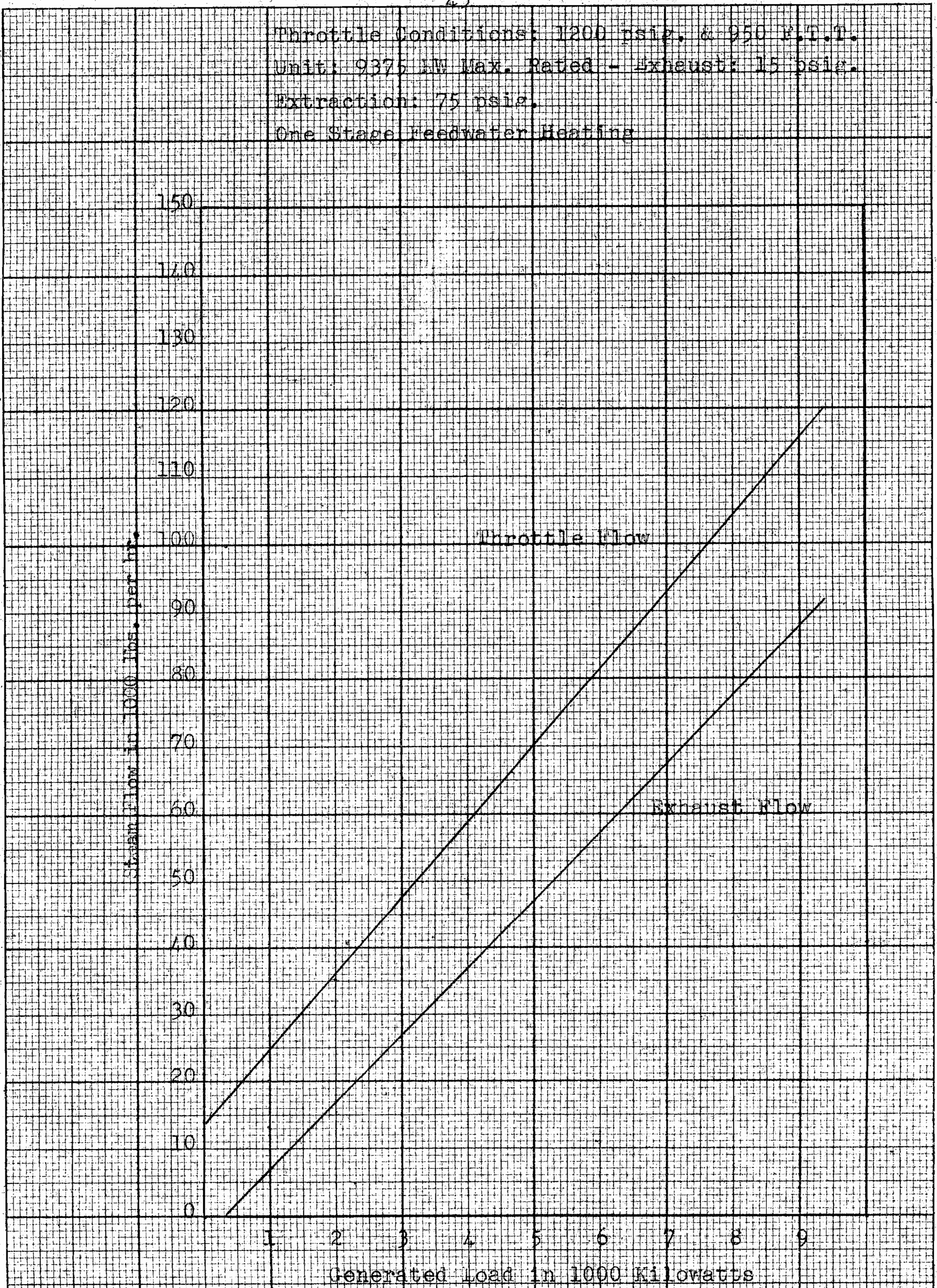


Fig. 26 Performance Curves (Expansion Factor 1.50)

CURVE CHART

Curve No. 1 =	The expected total electric load on the Heating and Power Plant.
Curve No. 2 =	The expected electric power generated by the proposed turbine with one stage feedwater heating.
Curve No. 3 =	The expected electric power generated by the proposed turbine with two stage feedwater heating.
Curve No. 4 =	The expected electric power generated by the proposed turbine with three stage feedwater heating.

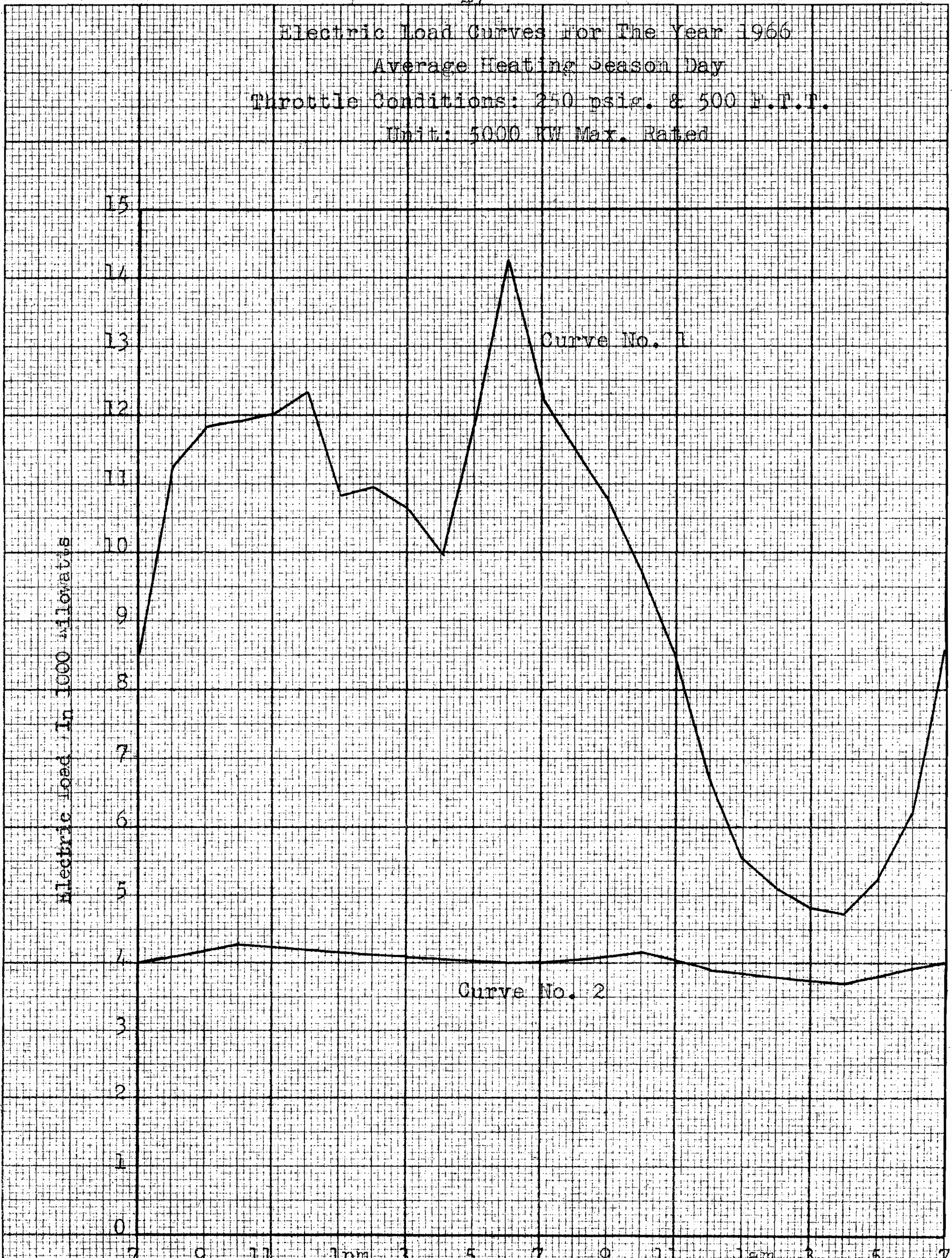


Fig. 27 Load Chart (Expansion Factor 2.14)

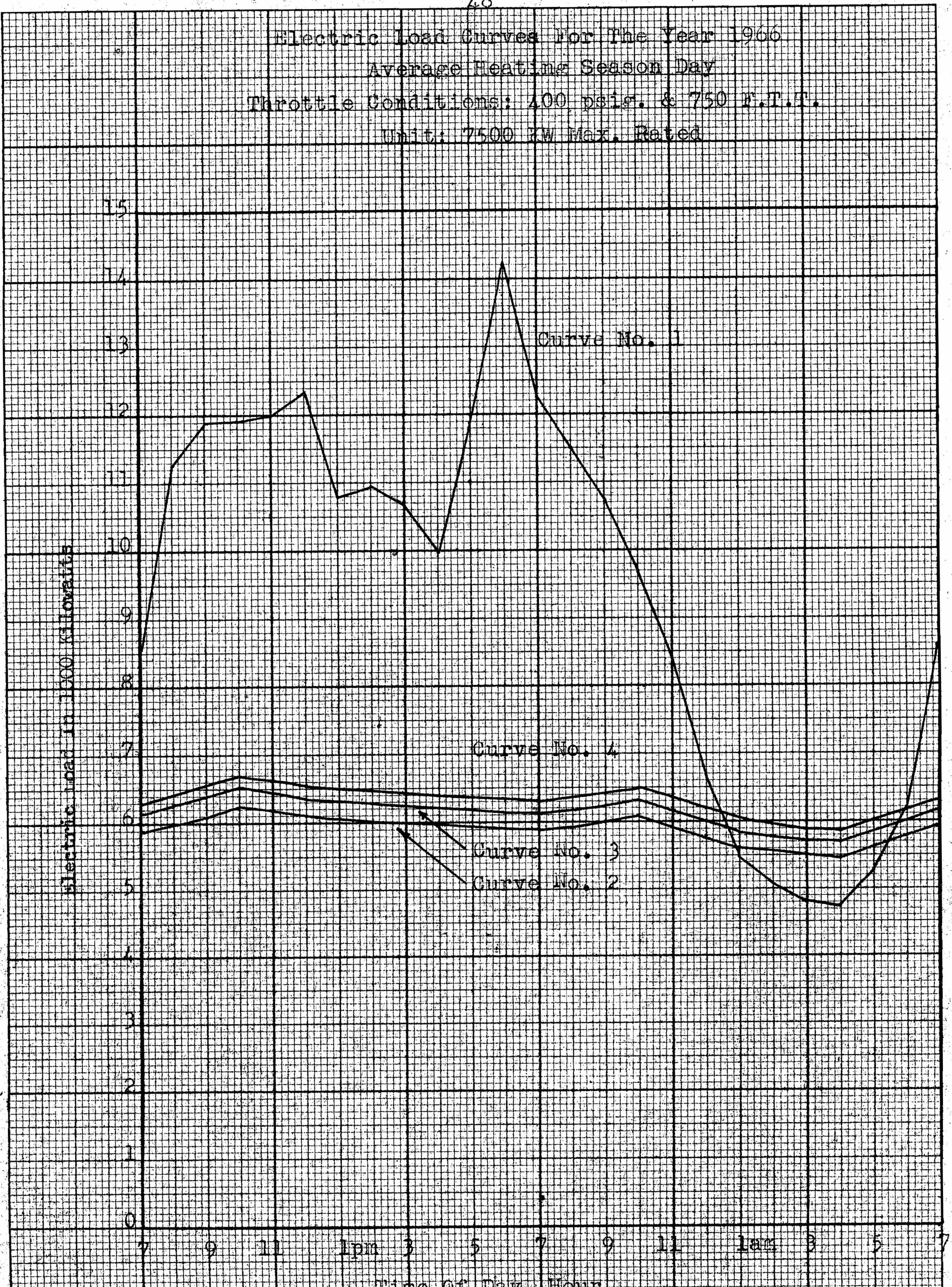


Fig. 28 Load Chart (Expansion Factor 2.14)

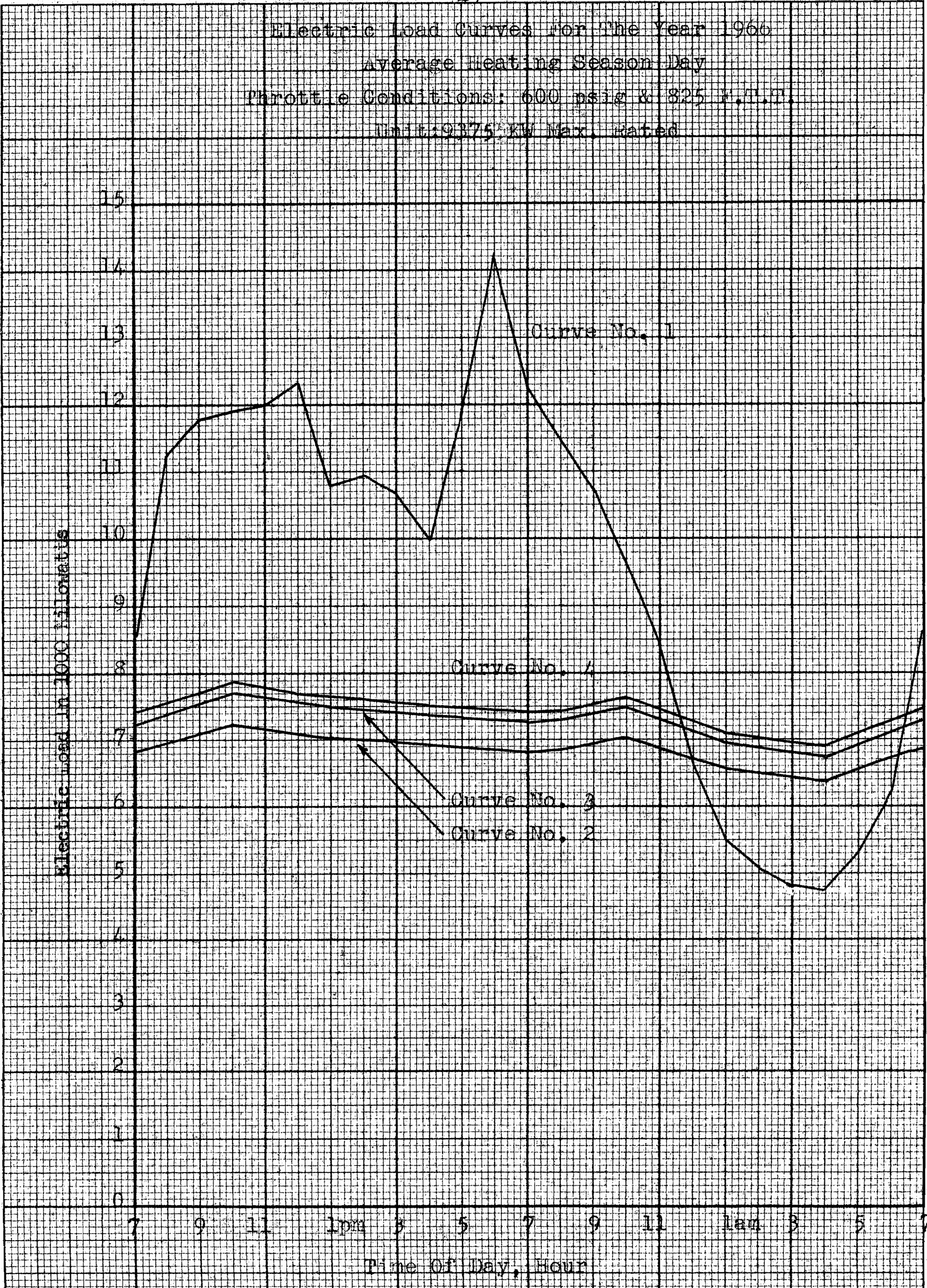


Fig. 29 Load Chart (Expansion factor 2.14)

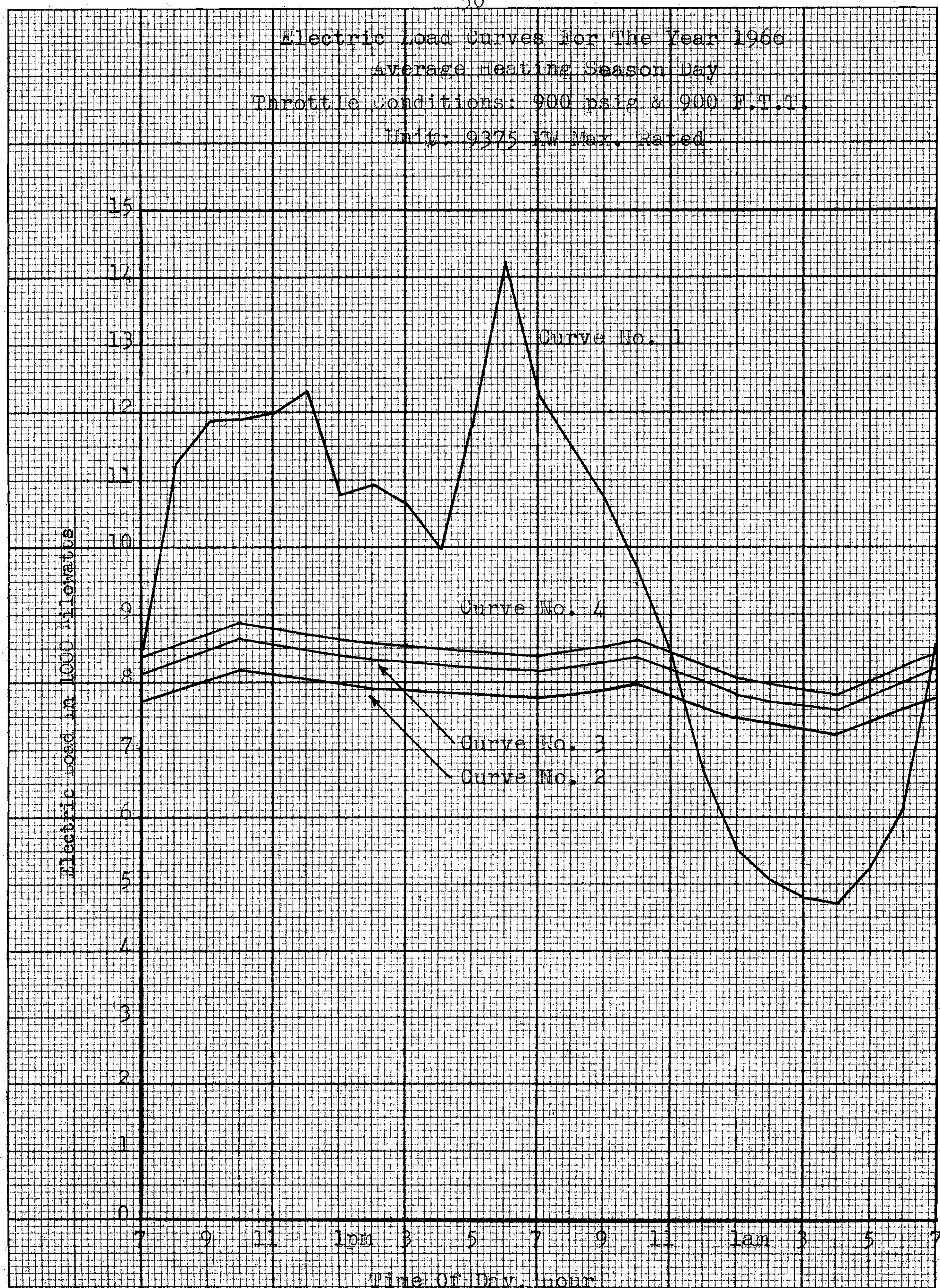


Fig. 30 Load Chart (Expansion Factor 2.14)

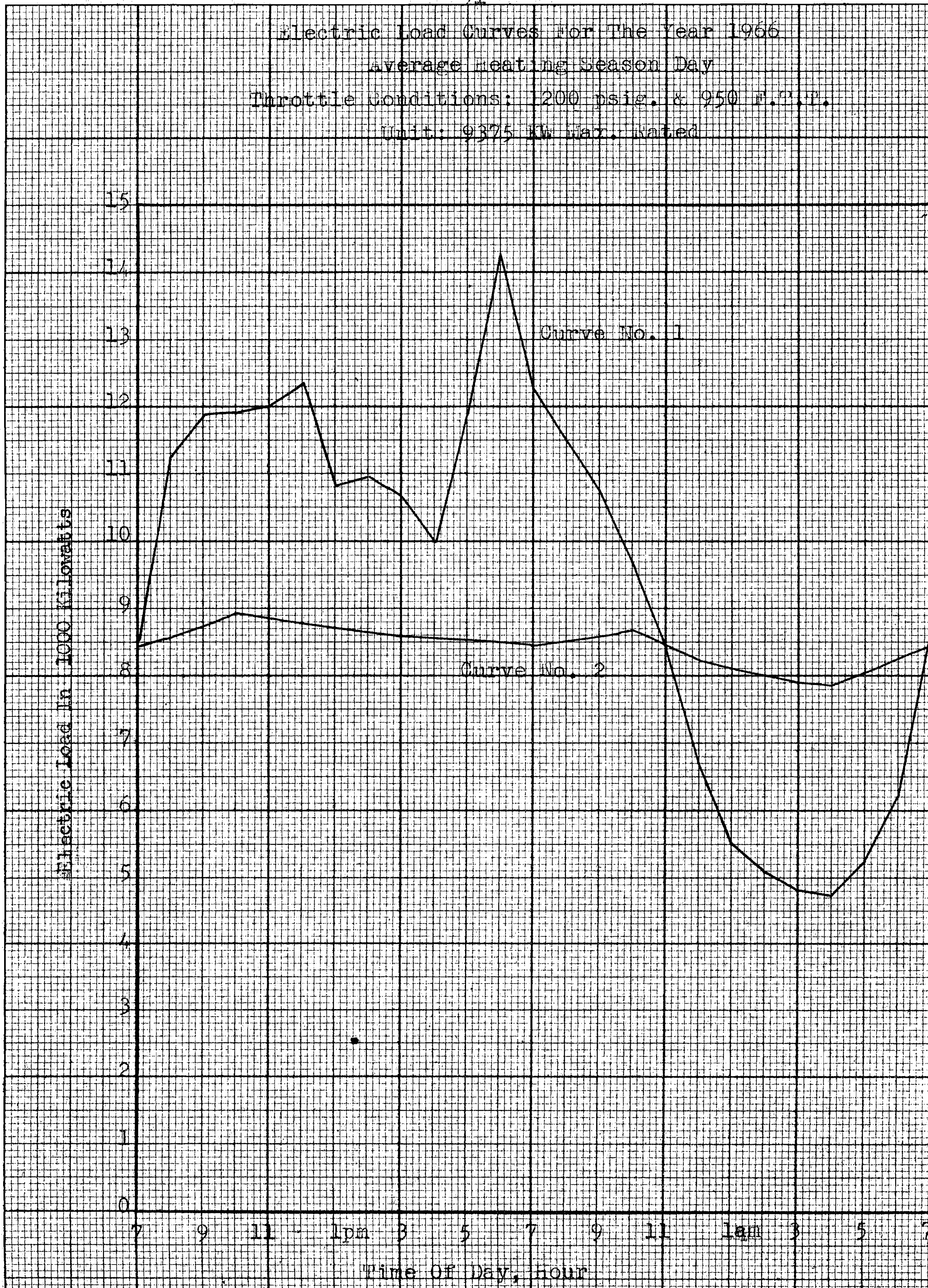


Fig. 31 Load Chart (Expansion Factor 2.14)

Fig. 32 Load Chart (Expansion Factor 1.50)

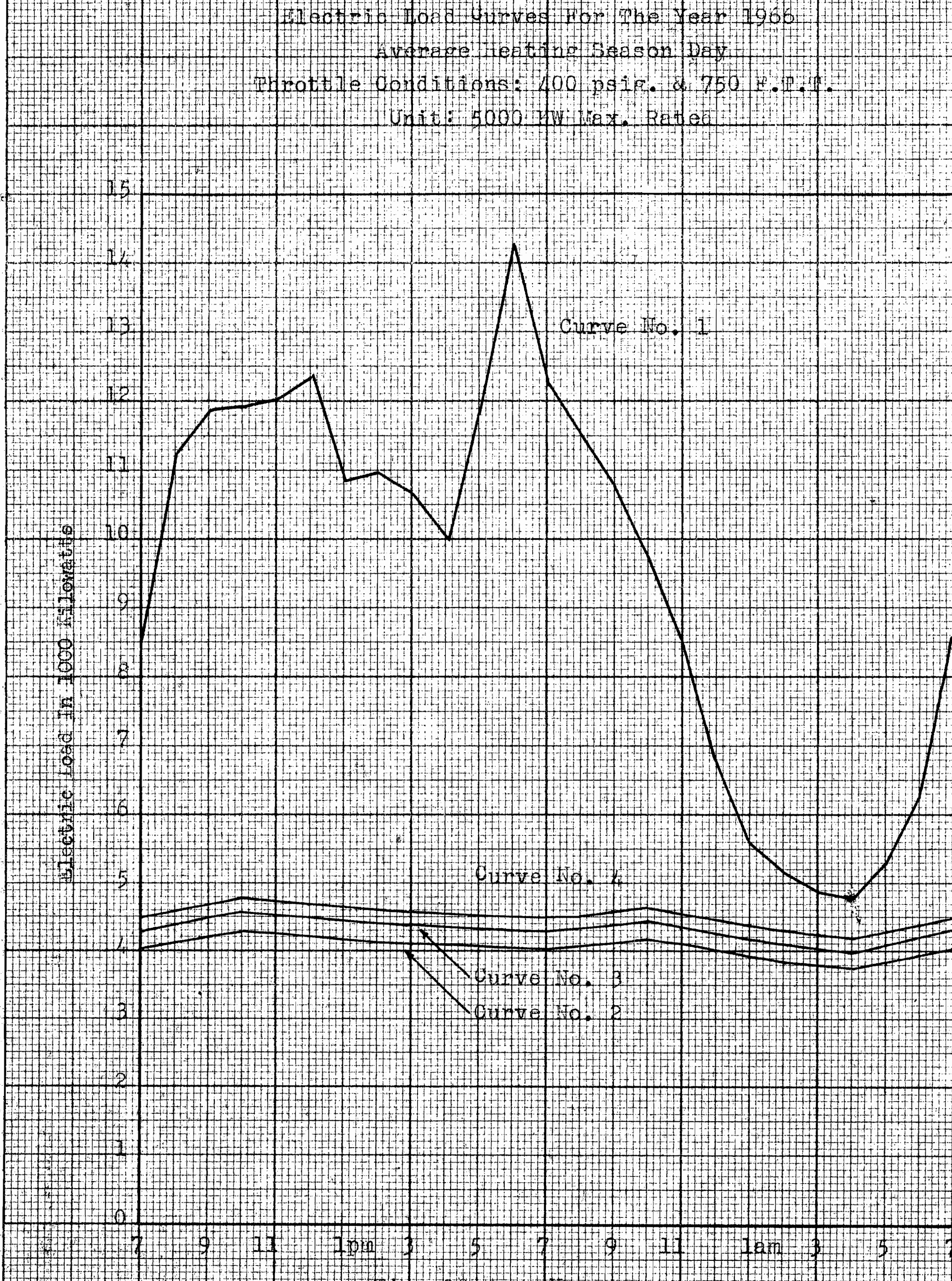


Fig. 33 Load Chart (Expansion Factor 1.50)

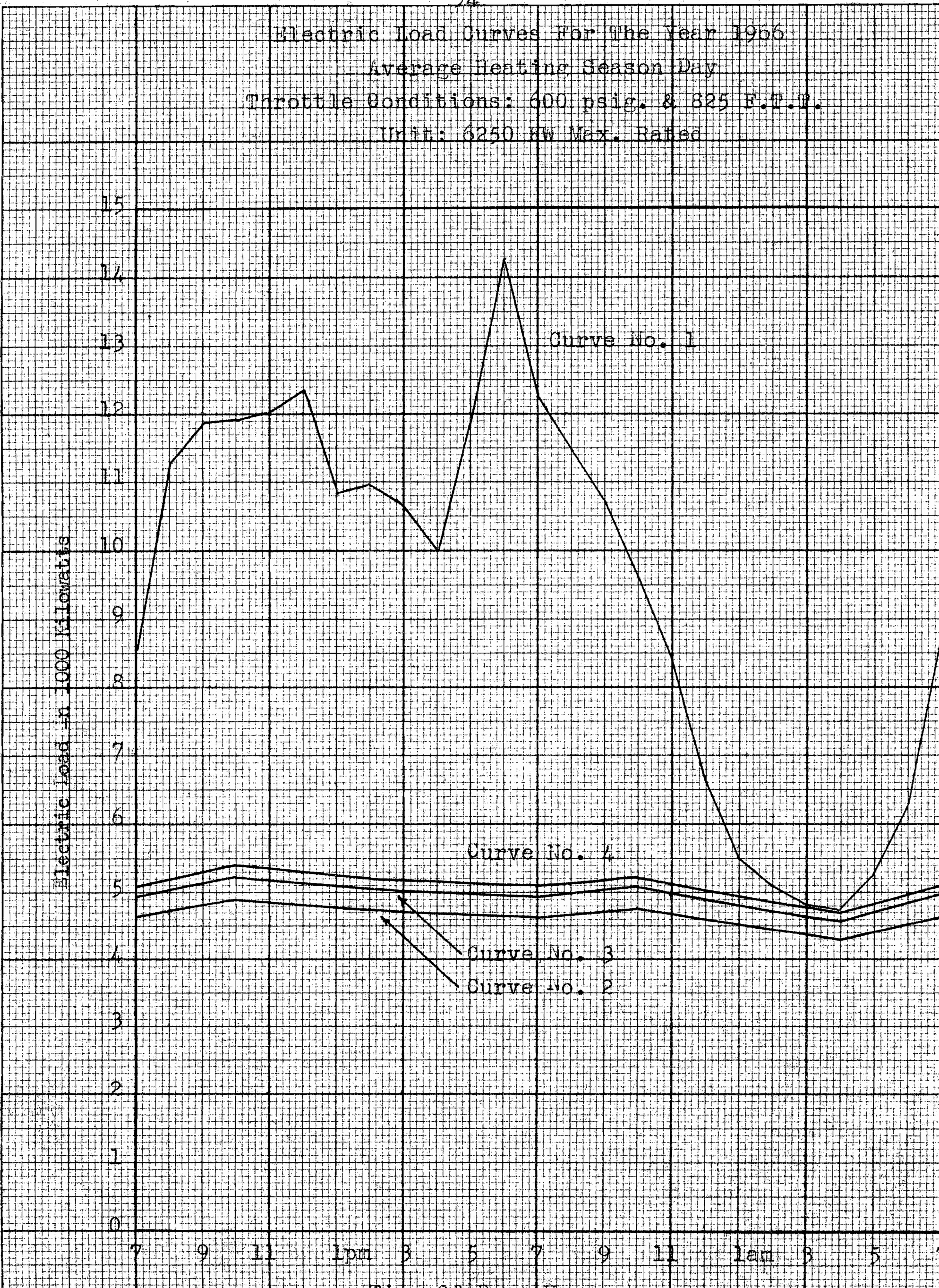


Fig. 34 Load Chart (Expansion Factor 1.50)

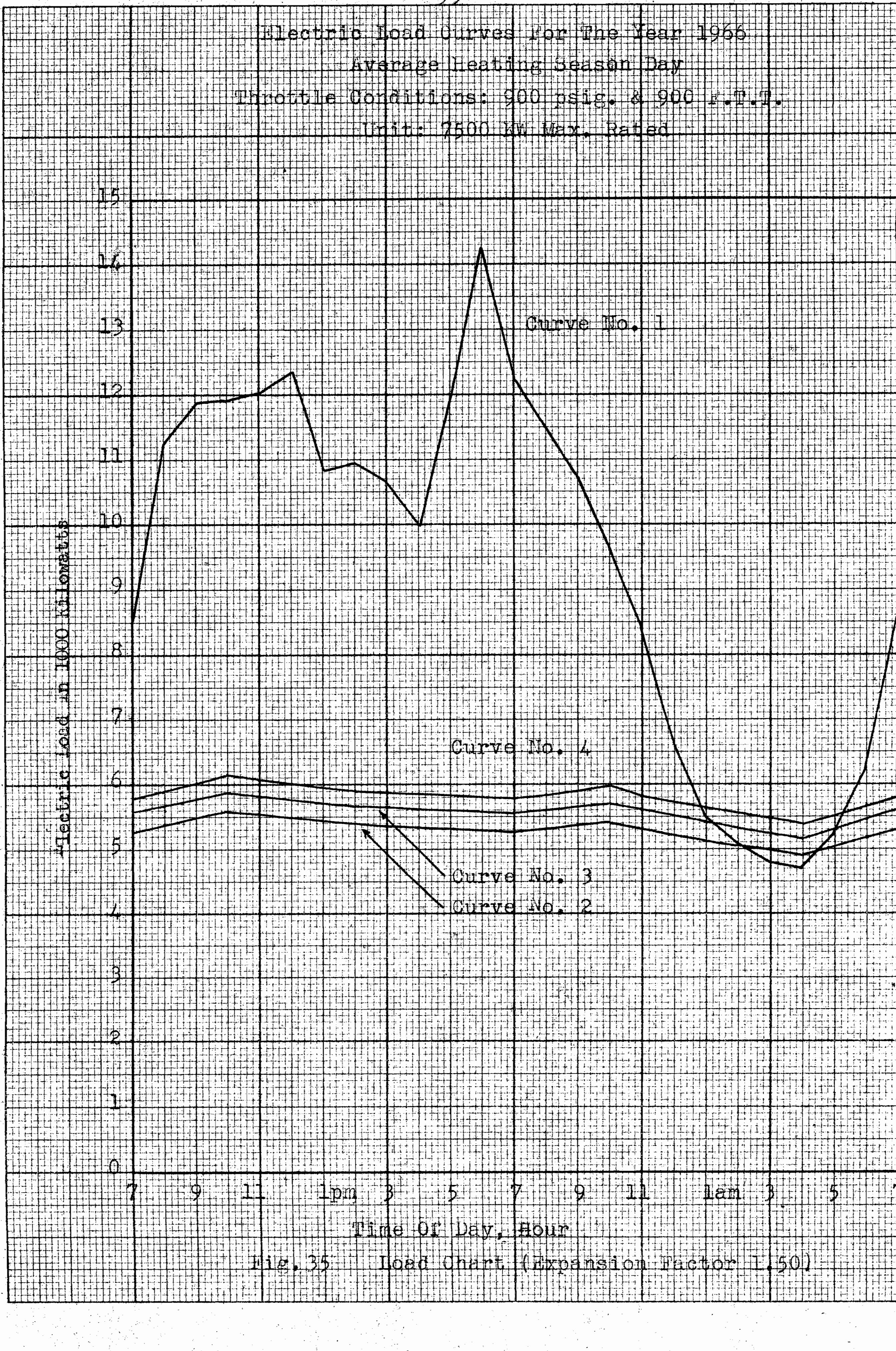


Fig. 35 Load Chart (Expansion Factor 1.50)

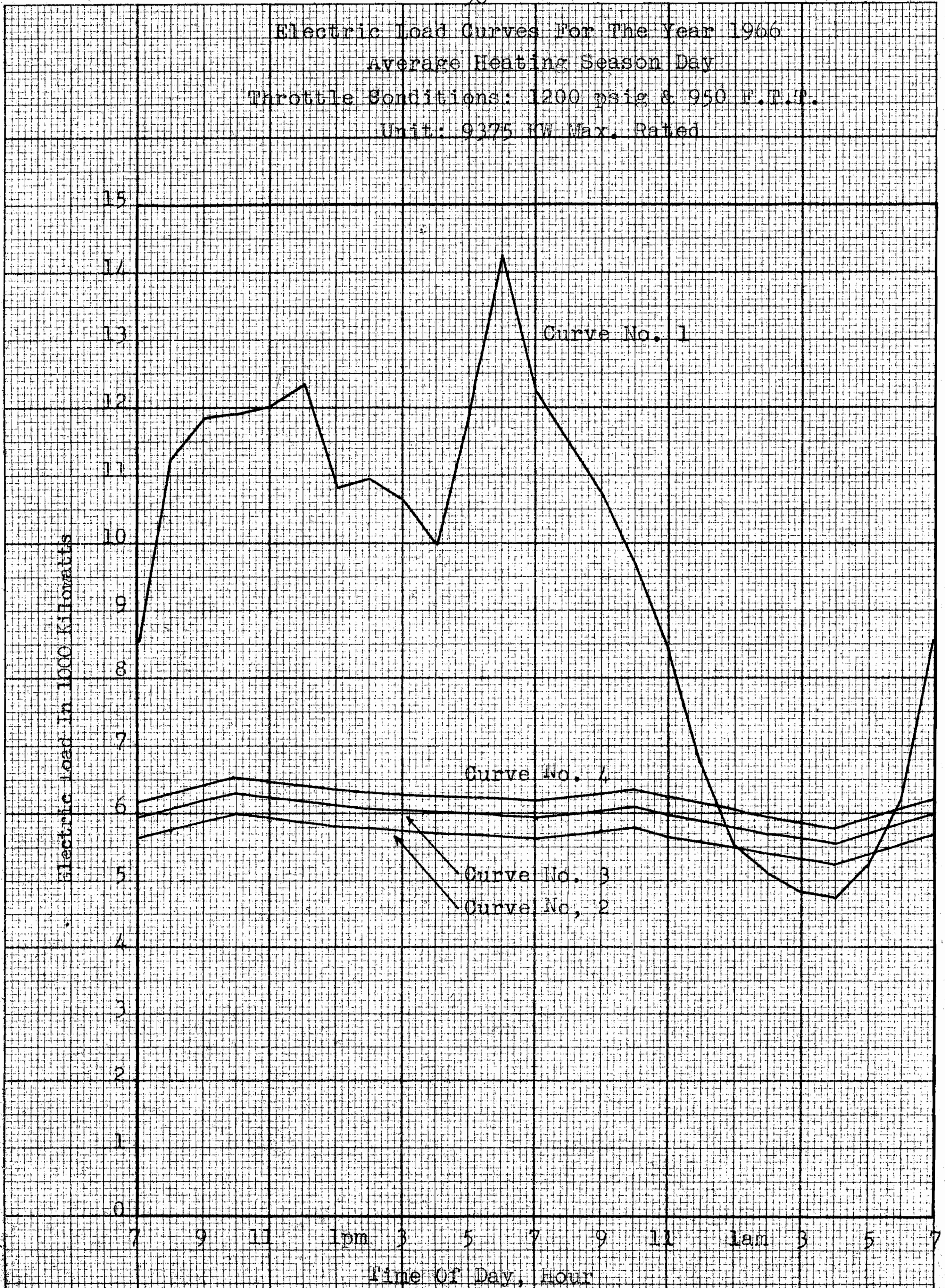
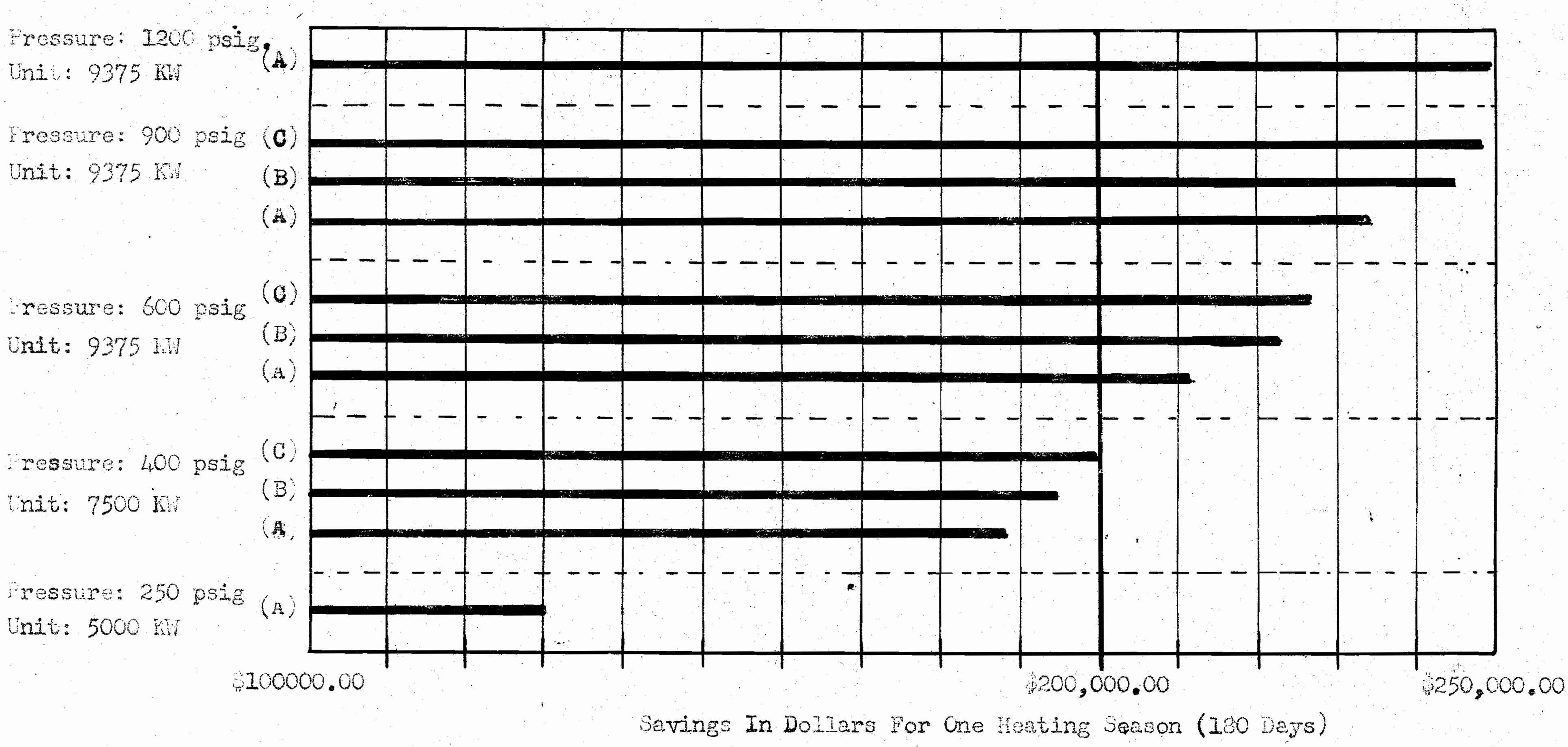


Fig. 36 Load Chart (Expansion Factor 1.50)

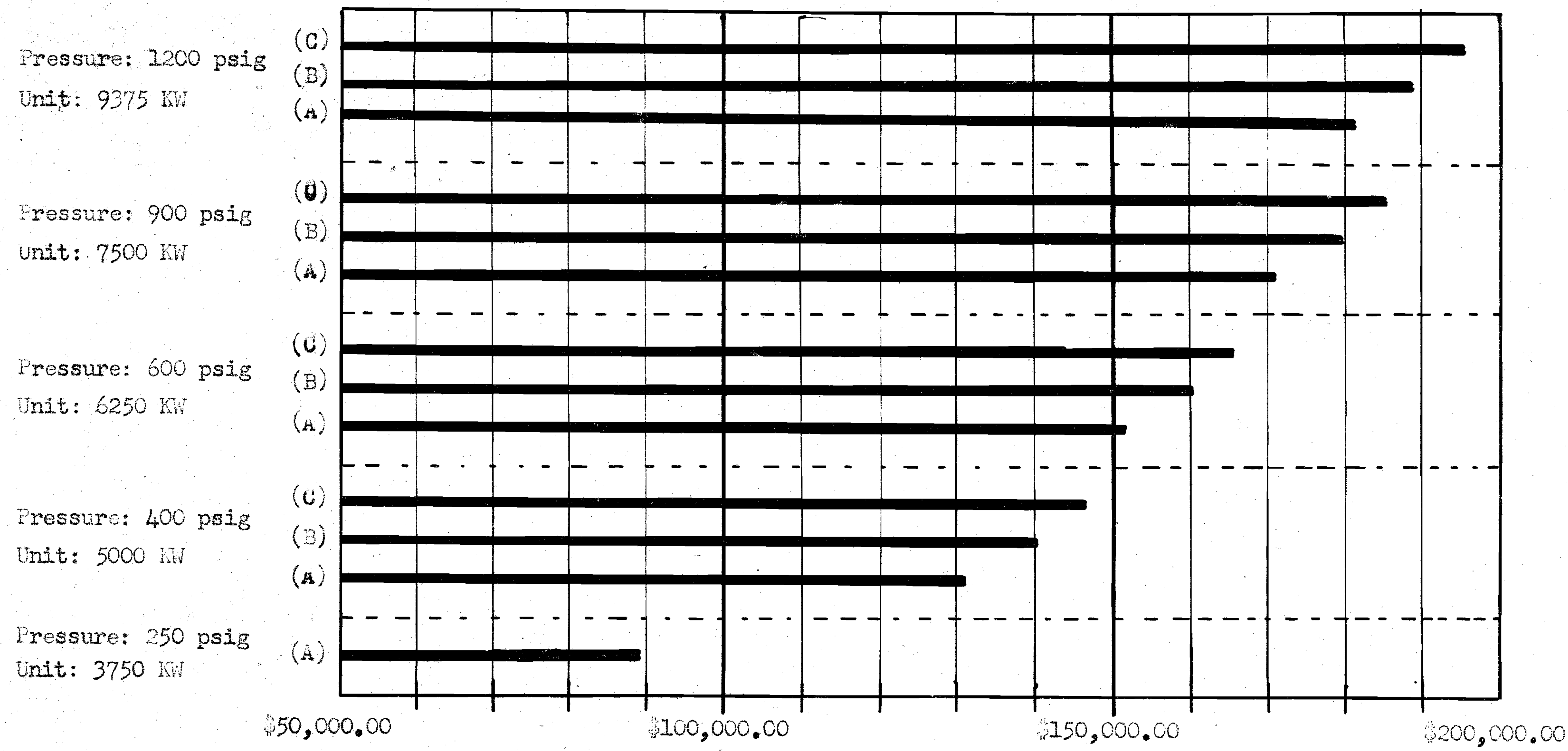


(A) One Stage Feedwater Heating

(B) Two Stage Feedwater Heating

(C) Three Stage Feedwater Heating

Fig. [illegible] Bar Chart (Expansion Factor 2.14)



Savings IN Dollars For One Heating Season (180 Days)

(A) One Stage Feedwater Heating

(B) Two Stage Feedwater Heating

(C) Three Stage Feedwater Heating

Fig. [illegible] Bar Chart (Expansion Factor 1.50)

VI

DISCUSSION OF RESULTS

The results of this investigation indicates that the installation of a back-pressure turbo-generator unit in the Virginia Polytechnic Institute Heating and Power Plant, operating under the specified conditions would show the following savings for the heating season of 1966-1967. Using a steam demand increase of 2.14 over the heating season of 1952-1953 the savings would be: for 250 psig. and 500 F and a 5000 KW unit with one stage feedwater heating, $130,757.00; for 400 psig. and 750 F and a 7500 KW unit with one stage heating, $188,440.00, the saving of two stage heating over one, $6,489.00, the saving of three stage heating over two, $4,997.00; for 600 psig. and 825 F and a 9375 KW unit with one stage heating, $212,095.00, the saving of two stage heating over one, $10,952.00, the saving of three stage heating over two, $3,816.00; for 900 psig. and 900 F and a 9375 KW unit with one stage heating, $234,580.00, the saving of two stage heating over one, $9,420.00, the saving of three stage heating over two, $4,731.00; for 1200 psig. and 950 F and a 9375 KW unit with one stage heating, $249,480.00.

Using a steam demand increase of 1.50 over the heating season of 1952-1953, the expected savings would be: for 250 psig. and 500 F and a 3750 KW unit with one stage feedwater heating, $89,100.00; for 400 psig. and 750 F and a 5000 KW unit with one stage heating, $131,299.00, the saving of two stage heating over one, $9,121.00, the saving of three stage heating over two, $6,311.00; for 600 psig. and 825 F and a 6250 KW unit with one stage heating, $151,738.00, the saving of two stage heating over one, $8,642.00, the saving of three stage heating over two, $5,130.00; for 900 psig. and 900 F and a 7500 KW unit with one stage heating, $171,180.00, the saving of two stage heating over one, $8,307.00, the saving of three stage heating over two, $6,068.00; for 1200 psig. and 950 F and a 9375 unit with one stage heating, $181,710.00, the saving of two stage heating over one, $7,440.00, the saving of three stage heating over two, $6,547.00.

These results are based on predictions of the electric load and steam demand placed on the plant during the heating season of 1966-1967. This year was chosen because it would represent in years approximately one fourth of the expected life of the turbo-generator unit provided it was installed by the year 1959. These predictions were used to determine the average daily load curves for the heating season of the year studied.

A study of these curves showed the following information:

1. The expected future low and high pressure steam load curves were used as the basis for this investigation. These curves showed the amount of steam available for electric energy production. The high pressure 75 psig. steam demand remained fairly constant for the 24 hour period with a maximum variation in load of 2000 lb. per hour. The actual variation in load came from the low pressure steam demand with peak loads at 10:00 A.M. and 10:00 P.M. and a load drop in the early morning hours.

2. The turbo-generator performance curves showed that with higher steam pressures and temperatures larger sized units could

be used for a given steam demand. The maximum size unit which could be installed is a 9375 KW turbo-generator because of the physical limitations of the Heating and Power Plant. Adding two and three stage feedwater heating increased the slope of the throttle flow curve while decreasing the slope of the exhaust flow curve. The effect of this was that for a given quantity of exhaust flow, a larger amount of electric energy could be produced by adding feedwater heating stages while also increasing the efficiency of the cycle. It was found that at the 250 psig. and 500 F throttle conditions for both the 5000 KW and 3750 KW units the exhaust flow for two and three stage feedwater heating was less than the amount of heating steam required. This same condition existed in the 1200 psig. class for a 9375 KW unit with an expansion factor of 2.14.

3. The generated electric load curves showed the advantages of higher steam pressures and temperatures along with feedwater heating. For each increase in steam pressure and temperature there is a definite increase in the amount of electric energy which can be produced. The increased electric energy production caused by using two and three stage feedwater heating were determined from these curves. In all cases the improvement in increased KW-hr production for two stage heating was greater than the improvement in production using the third stage. The difference in this improvement from one turbine to another was due to the per cent of rated loading at which each turbine was operating. The higher the loading the greater the improvement.

In the early morning hours in some cases the electric energy produced is higher than the total electric load. The extra power produced is considered dump power and is of no value and therefore creates an uneconomical situation. This situation could be remedied by lowering the steam demand during the early morning period or by selling the dump power to the nearby utility at a reduced rate.

Some assumptions were made in this investigation and must be kept in mind when interpreting the information presented in this thesis. Most of these assumptions have been mentioned previously, but it is felt that they should be repeated since the results of this investigation depend directly upon them. These assumptions were as follows:

1. Business and economical conditions remain constant, that there will be neither inflation nor depression. These conditions affect both the price of fuel and the price of electric power.

2. The total electric load for the college and the community of Blacksburg will increase at a rate of 12.5 per cent per year.

3. The college process and heating steam load will increase proportionally with the increase in student enrollment.

4. The same type fuel will continue to be used in steam generation. This fuel has a heating value of 11,500 B.T.U. per lb. The cost of this fuel will be $2.50 per ton.

5. A boiler with an efficiency of at least 80 per cent will be used to generate the steam.

6. The mixture of make up water and condensate drains will have an average temperature of 125 F.

7. The two surface type feedwater heaters will have a five

degree terminal difference. The deaerator will have a terminal difference of zero degrees.

8. There will be a 10 per cent drop in the steam pressure from the turbine to the feedwater heaters.

This investigation was based on the cost of fuel and the cost of electric power. The other operating costs and overhead costs were not considered to be within the scope of this thesis. As stated before, no attempt was made to select one definite turbo-generator unit as the most economical because of the many other factors involved.

VII

RECOMMENDATIONS

In conjunction with this investigation an economic study should be made concerning a low pressure condensing turbo-generator unit to operate off the exhaust flow of the proposed units in this thesis during the non-heating season. Also an investigation should be made concerning the physical problems of installing such units.

The next step in this problem would be an investigation to determine the type of steam generator needed to meet the future demands of the Heating and Power Plant. A study should also be made on the boiler water conditioning problems. An investigation should be made on the use of pulverized burners or cyclone furnaces as a more efficient method of burning the local fuel used in the Heating and Power Plant.

For more economical operation of the plant a study should be made to find a method of increasing the electric load during the early morning hours. This would reduce the amount of dump power generated and produce a larger saving.

VIII

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IX

VITA

David Henry Williams, Jr., was born in Portsmouth, Ohio, on March 29, 1931. One year later the family moved to Fort Thomas, Kentucky. The family is maintaining its residence in Fort Thomas at the present time. There Mr. Williams completed his secondary education by graduating from Highlands High School in June, 1949.

In the fall of 1949, he entered Virginia Polytechnic Institute as a student in Mechanical Engineering. He was elected to Pi Tau Sigma, the national honorary mechanical engineering fraternity, in the fall of 1952. He received his Bachelor of Science Degree in Mechanical Engineering in June, 1953, and the same month accepted an appointment as a graduate fellow in Power and Fuel Engineering at Virginia Polytechnic Institute.

Mr. Williams is a member of the Officer's Reserve Corps and has been ordered to active duty with the Engineer Corps, United States Army, upon completion of his graduate program.

David H. Williams Jr.

X

APPENDICES

TABLE NO. 1

AVERAGE ELECTRIC LOAD FOR THE HEATING SEASON IN KILOWATTS

Time Of Day	Year 1952-1953	Year 1966-1967
7:00 A.M.	1649	8583
8:00 A.M.	2161	11248
9:00 A.M.	2283	11883
10:00 A.M.	2293	11935
11:00 A.M.	2304	11992
12:00 A.M.	2371	12341
1:00 P.M.	2078	10815
2:00 P.M.	2104	10951
3:00 P.M.	2055	10696
4:00 P.M.	1917	9978
5:00 P.M.	2286	11898
6:00 P.M.	2736	14241
7:00 P.M.	2357	12268
8:00 P.M.	2213	11518
9:00 P.M.	2069	10769
10:00 P.M.	1864	9702
11:00 P.M.	1619	8427
12:00 P.M.	1281	6668
1:00 A.M.	1055	5491
2:00 A.M.	977	5085
3:00 A.M.	925	4815
4:00 A.M.	912	4747
5:00 A.M.	1009	5251
6:00 A.M.	1190	6194

TABLE NO. II

AVERAGE DAILY STEAM DEMAND FOR THE HEATING SEASON IN POUNDS PER HOUR

EXPANSION FACTOR 2.14

Time Of Day	Year 1952-1953		Year 1966-1967		
	Low Pressure Steam Demand	High Pressure Steam Demand	Low Pressure Steam Demand	High Pressure Steam Demand	Total Steam Demand
7:00 A.M.	35900	10070	76826	21746	98575
10:00 A.M.	38100	10404	81535	22264	103799
1:00 P.M.	37000	9623	79180	20593	99773
4:00 P.M.	36400	9743	77896	20850	98746
7:00 P.M.	35900	9747	76826	20859	97685
10:00 P.M.	36900	9563	78966	20464	99430
1:00 A.M.	34800	9450	73212	20233	93435
4:00 A.M.	33200	9468	71048	20261	91309

TABLE NO. III

AVERAGE DAILY STEAM DEMAND FOR THE HEATING SEASON IN POUNDS PER HOUR

EXPANSION FACTOR 1.50

Time Of Day	Year 1952-1953		Year 1966-1967		
	Low Pressure Steam Demand	High Pressure Steam Demand	Low Pressure Steam Demand	High Pressure Steam Demand	Total Steam Demand
7:00 A.M.	35900	10070	53850	15105	68955
10:00 A.M.	38100	10404	57150	15606	72756
1:00 P.M.	37000	9623	55500	14434	69934
4:00 P.M.	36400	9743	54600	14614	69214
7:00 P.M.	35900	9747	53850	14621	68471
10:00 P.M.	36900	9563	55350	14345	69695
1:00 A.M.	34800	9450	52200	14175	66375
4:00 A.M.	33200	9468	49800	14202	64002

TABLE NO. IV

EXPECTED LOADING OF 5000 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 250 psig. and 500 F

Expansion Factor: 2.14

Time Of Day	One Stage Feedwater Heating
7:00 A.M.	4000
8:00 A.M.	4080
9:00 A.M.	4180
10:00 A.M.	4280
11:00 A.M.	4220
12:00 A.M.	4190
1:00 P.M.	4150
2:00 P.M.	4120
3:00 P.M.	4100
4:00 P.M.	4070
5:00 P.M.	4040
6:00 P.M.	4020
7:00 P.M.	4010
8:00 P.M.	4050
9:00 P.M.	4100
10:00 P.M.	4170
11:00 P.M.	4030
12:00 P.M.	3920
1:00 A.M.	3820
2:00 A.M.	3790
3:00 A.M.	3750
4:00 A.M.	3710
5:00 A.M.	3820
6:00 A.M.	3920

TABLE NO. V

EXPECTED LOADING OF 7500 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 400 psig. and 750 F

Expansion Factor: 2.14

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	5900	6140	6300
8:00 A.M.	6000	6230	6410
9:00 A.M.	6090	6380	6570
10:00 A.M.	6260	6520	6690
11:00 A.M.	6190	6450	6620
12:00 A.M.	6100	6360	6560
1:00 P.M.	6080	6320	6500
2:00 P.M.	6020	6290	6490
3:00 P.M.	6000	6240	6430
4:00 P.M.	5980	6210	6400
5:00 P.M.	5940	6190	6390
6:00 P.M.	5910	6150	6360
7:00 P.M.	5900	6140	6310
8:00 P.M.	5920	6180	6380
9:00 P.M.	6000	6240	6420
10:00 P.M.	6080	6310	6500
11:00 P.M.	5910	6160	6370
12:00 P.M.	5780	6000	6190
1:00 A.M.	5620	5850	6050
2:00 A.M.	5580	5790	5980
3:00 A.M.	5510	5730	5900
4:00 A.M.	5480	5690	5880
5:00 A.M.	5630	5830	6020
6:00 A.M.	5780	6000	6190

TABLE NO. VI

EXPECTED LOADING OF 9375 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 600 psig. and 825 F

Expansion Factor: 2.14

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	6840	7280	7440
8:00 A.M.	6970	7400	7560
9:00 A.M.	7100	7560	7700
10:00 A.M.	7250	7720	7870
11:00 A.M.	7180	7650	7790
12:00 A.M.	7090	7570	7700
1:00 P.M.	7030	7500	7650
2:00 P.M.	7000	7470	7610
3:00 P.M.	6980	7410	7540
4:00 P.M.	6920	7370	7510
5:00 P.M.	6890	7340	7470
6:00 P.M.	6860	7300	7430
7:00 P.M.	6800	7260	7410
8:00 P.M.	6860	7300	7430
9:00 P.M.	6970	7400	7540
10:00 P.M.	7030	7480	7620
11:00 P.M.	6860	7300	7430
12:00 P.M.	6700	7130	7280
1:00 A.M.	6570	6960	7100
2:00 A.M.	6500	6880	7020
3:00 A.M.	6430	6810	6960
4:00 A.M.	6380	6760	6910
5:00 A.M.	6530	6920	7080
6:00 A.M.	6710	7130	7290

TABLE NO. VII

EXPECTED LOADING OF 9375 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 900 psig. and 900 F

Expansion Factor: 2.14

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	7740	8180	8400
8:00 A.M.	7880	8280	8520
9:00 A.M.	8020	8460	8700
10:00 A.M.	8190	8650	8900
11:00 A.M.	8110	8550	8800
12:00 A.M.	8040	8480	8700
1:00 P.M.	7990	8400	8650
2:00 P.M.	7920	8340	8600
3:00 P.M.	7900	8300	8550
4:00 P.M.	7840	8230	8500
5:00 P.M.	7830	8210	8470
6:00 P.M.	7800	8190	8430
7:00 P.M.	7770	8180	8400
8:00 P.M.	7800	8210	8470
9:00 P.M.	7890	8300	8510
10:00 P.M.	7990	8380	8620
11:00 P.M.	7800	8200	8420
12:00 P.M.	7610	8020	8250
1:00 A.M.	7490	7820	8050
2:00 A.M.	7410	7720	7980
3:00 A.M.	7300	7670	7900
4:00 A.M.	7260	7600	7820
5:00 A.M.	7420	7800	8020
6:00 A.M.	7610	8000	8240

TABLE NO. VIII

EXPECTED LOADING OF 9375 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 1200 psig. and 950 F

Expansion Factor: 2.14

Time Of Day	One Stage Feedwater Heating
7:00 A.M.	8450
8:00 A.M.	8550
9:00 A.M.	8740
10:00 A.M.	8950
11:00 A.M.	8850
12:00 A.M.	8780
1:00 P.M.	8700
2:00 P.M.	8650
3:00 P.M.	8590
4:00 P.M.	8550
5:00 P.M.	8530
6:00 P.M.	8500
7:00 P.M.	8440
8:00 P.M.	8500
9:00 P.M.	8580
10:00 P.M.	8700
11:00 P.M.	8470
12:00 P.M.	8230
1:00 A.M.	8100
2:00 A.M.	8000
3:00 A.M.	7900
4:00 A.M.	7870
5:00 A.M.	8050
6:00 A.M.	8270

TABLE NO. IX

EXPECTED LOADING OF 3750 KW UNIT FOR THE HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 250 psig. and 500 F

Expansion Factor: 1.50

Time Of Day	One Stage Feedwater Heating
7:00 A.M.	2740
8:00 A.M.	2800
9:00 A.M.	2875
10:00 A.M.	2920
11:00 A.M.	2900
12:00 A.M.	2870
1:00 P.M.	2840
2:00 P.M.	2820
3:00 P.M.	2800
4:00 P.M.	2780
5:00 P.M.	2760
6:00 P.M.	2750
7:00 P.M.	2740
8:00 P.M.	2770
9:00 P.M.	2800
10:00 P.M.	2820
11:00 P.M.	2780
12:00 P.M.	2710
1:00 A.M.	2670
2:00 A.M.	2620
3:00 A.M.	2570
4:00 A.M.	2530
5:00 A.M.	2610
6:00 A.M.	2690

TABLE NO. X

EXPECTED LOADING OF 5000 KW UNIT FOR THE HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 400 psig. and 750 F

Expansion Factor: 1.50

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	4030	4300	4500
8:00 A.M.	4100	4390	4590
9:00 A.M.	4200	4480	4680
10:00 A.M.	4280	4570	4780
11:00 A.M.	4240	4530	4730
12:00 A.M.	4200	4480	4680
1:00 P.M.	4150	4440	4640
2:00 P.M.	4130	4410	4610
3:00 P.M.	4100	4390	4590
4:00 P.M.	4090	4360	4560
5:00 P.M.	4050	4330	4530
6:00 P.M.	4040	4310	4510
7:00 P.M.	4030	4300	4500
8:00 P.M.	4045	4320	4520
9:00 P.M.	4100	4390	4590
10:00 P.M.	4170	4440	4640
11:00 P.M.	4100	4320	4520
12:00 P.M.	4000	4250	4450
1:00 A.M.	3920	4180	4380
2:00 A.M.	3840	4100	4300
3:00 A.M.	3800	4050	4250
4:00 A.M.	3750	3980	4180
5:00 A.M.	3820	4080	4280
6:00 A.M.	3950	4200	4400

TABLE NO. XI

EXPECTED LOADING OF 6250 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 600 psig. and 825 F

Expansion Factor: 1.50

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	4630	4940	5090
8:00 A.M.	4740	5020	5190
9:00 A.M.	4820	5130	5300
10:00 A.M.	4910	5220	5400
11:00 A.M.	4860	5180	5350
12:00 A.M.	4820	5120	5290
1:00 P.M.	4780	5100	5250
2:00 P.M.	4760	5060	5200
3:00 P.M.	4740	5020	5190
4:00 P.M.	4700	5000	5150
5:00 P.M.	4680	4980	5120
6:00 P.M.	4650	4960	5100
7:00 P.M.	4630	4940	5090
8:00 P.M.	4670	4970	5110
9:00 P.M.	4740	5020	5190
10:00 P.M.	4770	5080	5220
11:00 P.M.	4670	4970	5110
12:00 P.M.	4590	4890	5020
1:00 A.M.	4510	4800	4930
2:00 A.M.	4420	4720	4860
3:00 A.M.	4360	4660	4790
4:00 A.M.	4300	4580	4700
5:00 A.M.	4400	4700	4830
6:00 A.M.	4540	4840	4980

TABLE NO. XII

EXPECTED LOADING OF 7500 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 900 psig. and 900 F

Expansion Factor: 1.50

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	5280	5590	5800
8:00 A.M.	5380	5650	5900
9:00 A.M.	5480	5770	6010
10:00 A.M.	5570	5870	6170
11:00 A.M.	5540	5820	6070
12:00 A.M.	5480	5770	6020
1:00 P.M.	5450	5720	5990
2:00 P.M.	5400	5680	5930
3:00 P.M.	5380	5650	5900
4:00 P.M.	5350	5620	5880
5:00 P.M.	5300	5600	5840
6:00 P.M.	5290	5580	5820
7:00 P.M.	5280	5570	5800
8:00 P.M.	5300	5590	5830
9:00 P.M.	5380	5650	5900
10:00 P.M.	5420	5710	5980
11:00 P.M.	5300	5590	5830
12:00 P.M.	5220	5520	5730
1:00 A.M.	5120	5410	5630
2:00 A.M.	5040	5320	5550
3:00 A.M.	5000	5260	5470
4:00 A.M.	4900	5170	5380
5:00 A.M.	5010	5290	5520
6:00 A.M.	5160	5450	5670

TABLE NO. XIII

EXPECTED LOADING OF 9375 KW UNIT FOR THE

HEATING SEASON OF 1966-1967 IN KILOWATTS

Throttle Conditions: 1200 psig. and 950 F

Expansion Factor: 1.50

Time Of Day	One Stage Feedwater Heating	Two Stage Feedwater Heating	Three Stage Feedwater Heating
7:00 A.M.	5640	5940	6170
8:00 A.M.	5750	6050	6290
9:00 A.M.	5880	6170	6410
10:00 A.M.	6000	6290	6530
11:00 A.M.	5920	6220	6460
12:00 A.M.	5880	6180	6420
1:00 P.M.	5810	6110	6350
2:00 P.M.	5780	6080	6320
3:00 P.M.	5750	6050	6290
4:00 P.M.	5720	6020	6260
5:00 P.M.	5700	6000	6240
6:00 P.M.	5670	5970	6210
7:00 P.M.	5640	5940	6180
8:00 P.M.	5680	5980	6220
9:00 P.M.	5750	6050	6290
10:00 P.M.	5800	6100	6340
11:00 P.M.	5680	5980	6220
12:00 P.M.	5600	5900	6140
1:00 A.M.	5500	5800	6040
2:00 A.M.	5400	5700	5940
3:00 A.M.	5340	5640	5880
4:00 A.M.	5250	5550	5790
5:00 A.M.	5390	5690	5930
6:00 A.M.	5540	5840	6080

TABLE NO. XIV

INDICATED SAVINGS PER DAY FOR THE HEATING SEASON OF

1966-1967

Throttle Conditions	Turbine Size	One Stage Feedwater Heating	Additional Saving With Two Stages	Additional Saving With Three Stages
Expansion Factor 2.14				
250 psig.	5000 KW	$ 726.00		
400 psig.	7500 KW	1047.00	$36.00	$28.00
600 psig.	9375 KW	1178.00	61.00	21.00
900 psig.	9375 KW	1303.00	52.00	26.00
1200 psig.	9375 KW	1386.00		
Expansion Factor 1.50				
250 psig.	3750 KW	$ 495.00		
400 psig.	5000 KW	729.00	$51.00	$35.00
600 psig.	6250 KW	843.00	48.00	28.00
900 psig.	7500 KW	951.00	46.00	34.00
1200 psig.	9375 KW	1010.00	41.00	36.00

B. Sample Calculations

1. Future Expected Electrical Load:

$$EL_f = (1 + \frac{X}{100})^n \times EL_p$$

Where: EL_f is the future electrical load, kw.

X is the per cent increase expected per year.

EL_p is the present electric energy load, kw.

For heating season of 1966-1967, 7:00 A.M.

$$EL_f = (1 + \frac{12.5}{100})^{14} \times 1649 \text{ kw}$$

$$EL_f = \underline{8583 \text{ kw}}$$

2. Increase in steam demand:

It was assumed that the increase in the steam demand would be proportional to the increase in student enrollment.

$$\text{Expansion factor} = \frac{\text{enrollment 1966-1967}}{\text{enrollment 1952-1953}}$$

$$\text{Expansion factor} = \frac{7,500}{3,500} = 2.14$$

For the heating season of 1966-1967, 7:00 A.M.

$$SD_f = SD_p \times EF$$

Where: SD_f is the future steam demand, lb. per hr.

SD_p is the steam demand for 1952-1953, lb. per hr.

EF is the expansion factor.

$$SD_f = 35,900 \times 2.14$$

$$SD_f = \underline{76,826} \text{ lb. per hr.}$$

The following sample calculations will be for a throttle condition of 900 psig. and 900 F.T.T. and a 9375 KW unit with a loading of 80 per cent:

3. Theoretical steam rate:

$$T.S.R. = \frac{3413}{h_1 - h_e}$$

Where: T.S.R. is the theoretical steam rate, lb. per Kw-hr.

h_1 is the enthalpy of the steam at throttle conditions, B.T.U. per lb.

h_e is the enthalpy of the steam at exhaust conditions, B.T.U. per lb.

$$T.S.R. = \frac{3413}{1451.4 - 1109.0}$$

T.S.R. = <u>9.99</u> lb. per Kw-hr.

4. Actual Steam Rate At Rated Load:

$$S.R. = \frac{T.S.R.}{E_b \times S_f} \times LF$$

Where: E_b is the basic efficiency. *

S_f is the superheat factor. *

LF is the load factor at 80 per cent. *

$$S.R. = \frac{9.99}{0.747 \times 1.003} \times 1.006$$

S.R. = <u>13.42</u> lb. per Kw-hr.

5. Theoretical Steam Rate, Throttle To Extraction At 75 Psig:

$$T.S.R.' = \frac{3,413}{h_1 - h_2}$$

Where: T.S.R.' is the theoretical steam rate, throttle to extraction, lb. per Kw-hr.

h_2 is the enthalpy of the steam at the extraction point, B.T.U. per lb.

$$T.S.R.' = \frac{3413}{1451.4 - 1197.0}$$

T.S.R.' = 13.42 lb. per Kw-hr.

* Tables A, B, and C, pp. 251-275, <u>General Electric Apparatus</u>, General Electric, New York, 1953.

6. Basic Extraction Factor:

$$\text{T.S.R. Ratio} = \frac{\text{T.S.R.}}{\text{T.S.R.}'}$$

T.S.R. Ratio = 0.744

Correction to T.S.R. Ratio = 0.010 *

Corrected T.S.R. Ratio = 0.754

Basic Extraction Factor = 0.300 *

7. Throttle Flow With One Stage Extraction:

T = (Extraction Factor x Q) + S.R. x 0.80 x 9375 kw.

Where: T is the throttle flow, lb. per hr.

Q is the quantity of steam being extracted, lb. per hr.

T = (0.300 x 22,000) + 13.42 x 0.80 x 9375

T = 107,250 lb. per hr.

8. Actual Exhaust Enthalpy:

$$h_f = h_1 - \frac{F}{\text{S.R.}}$$

Where: h_f is the actual exhaust enthalpy B.T.U. per lb.

F is the final heat factor, 3600. [11]

S.R. is the steam rate at full load, lb. per kw-hr.

$$h_f = 1451.4 - \frac{3600}{13.34}$$

h_f = <u>1181.0</u> B.T.U. per lb.

9. Heat Balance For One Stage Feedwater Heating With a Deaerator:

$$Sb3\ (h_f) + (T - Sb3)\ (h_4) = T\ (h_5)$$

Where: Sb3 is the quantity of steam going to the deaerator, lb. per hr.

* Tables B and C, pp. 311-327, <u>General Electric Apparatus</u>, General Electric, New York, 1953.

h_4 is the enthalpy of the feedwater, B.T.U. per lb.

h_5 is the enthalpy of the feedwater leaving the deaerator, B.T.U. per lb.

Sb3 (1181.0) + (107,250 - Sb3) (93.0) = 107,250 (216.8)

Sb3 = <u>12,200</u> lb. per hr.

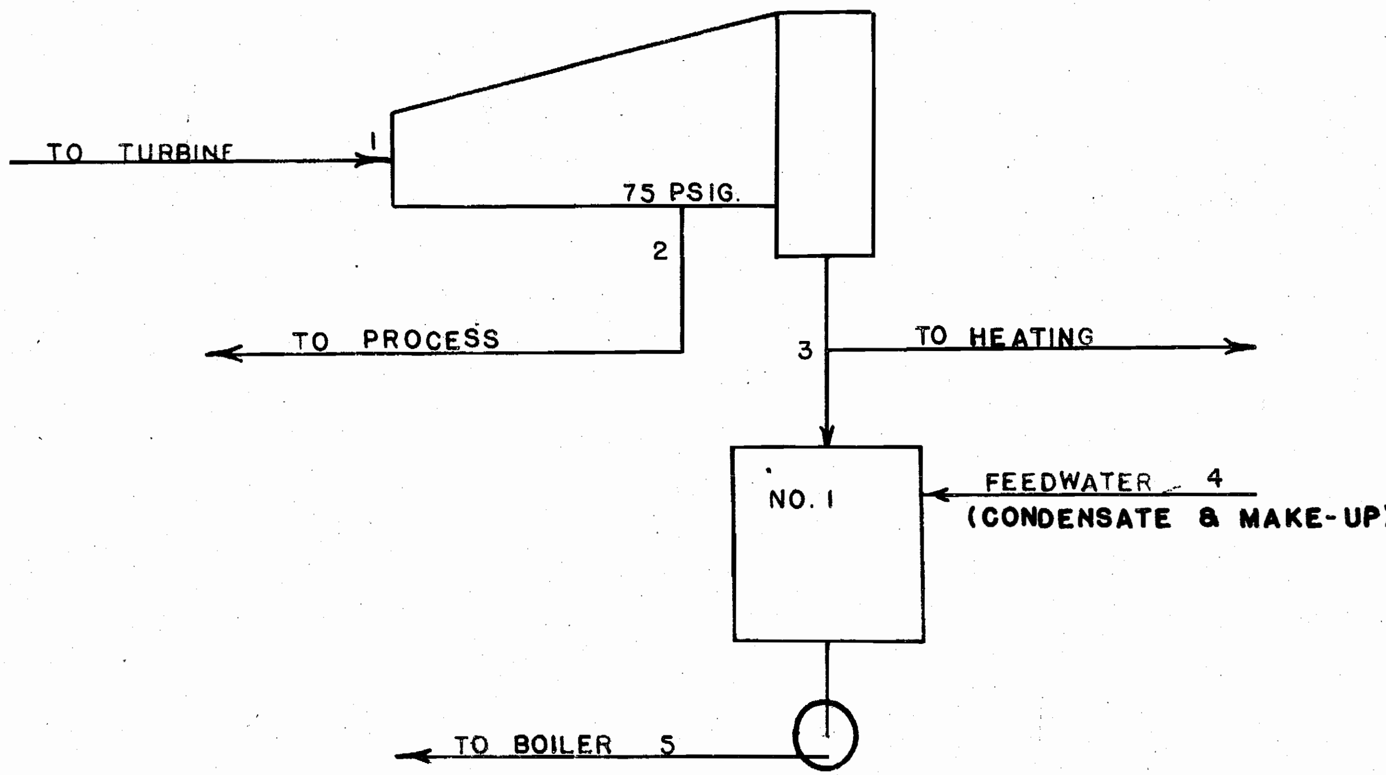

FIG. 39 ONE STAGE FEEDWATER HEATING CYCLE

HEAT CHART FOR DIAGRAM OF ONE STAGE FEEDWATER HEATING CYCLE

	B.T.U. per lb.				
Throttle Conditions	250 psig. 500 F	400 psig. 750 F	600 psig. 825 F	900 psig. 900 F	1200 psig. 950 F
Point 1	1262.0	1389.0	1421.0	1451.4	1466.0
Point 2	1187.0	1261.0	1257.0	1257.0	1252.0
Point 3	1118.0	1181.0	1179.5	1181.0	1178.0
Point 4	93.0	93.0	93.0	93.0	93.0
Point 5	216.8	216.8	216.8	216.8	216.8
Turbine KW	5000	7500	9375	9375	9375

Temperature Of Feedwater To Boiler (Point 5) Equals 248 F

EXPANSION FACTOR (2.14)

HEAT CHART FOR DIAGRAM OF ONE STAGE FEEDWATER HEATING CYCLE

	B.T.U. per lb.				
Throttle Conditions	250 psig. 500 F	400 psig. 750 F	600 psig. 825 F	900 psig. 900 F	1200 psig. 950 F
Point 1	1262.0	1389.0	1421.0	1451.4	1466.0
Point 2	1193.0	1265.0	1264.0	1264.0	1252.0
Point 3	1120.0	1186.0	1186.0	1188.0	1178.0
Point 4	93.0	93.0	93.0	93.0	93.0
Point 5	216.8	216.8	216.8	216.8	216.8
Turbine KW	3750	5000	6250	7500	9375

Temperature Of Feedwater To Boiler (Point 5) Equals 248 F

EXPANSION FACTOR (1.50)

10. Exhaust Flow With One Stage Feedwater Heating:

$Qe = T - Q - Sb3$

Where: Qe is the exhaust flow with one stage feedwater heating, lb. per hr.

$Qe = 107{,}250 - 22{,}000 - 12{,}200$

$Qe = \underline{73{,}050}$ lb. per hr.

11. Rankine Cycle Ratio:

$$R.C.R. = \frac{h_1 - h_f}{h_1 - h_e}$$

$$R.C.R. = \frac{1451.4 - 1161.0}{1451.4 - 1109.0}$$

$R.C.R. = 0.79$

12. Mechanical And Generator Efficiency:

$$Emg = \frac{S_f \times E_b}{R.C.R. \times LF} \times 100$$

Where: Emg = Mechanical and Generator Efficiency.

$$Emg = \frac{1.003 \times 0.747 \times 100}{0.79 \times 1.006}$$

$Emg = \underline{94}$ per cent

13. Internal Work:

$$Iw = \frac{3413 \times 0.80 \times 9375 \text{ kw}}{Emg}$$

Where: Iw is the internal work, B.T.U. per hr.

14. Heat Balance For Two Stage Feedwater Heating, Five Degree Terminal Difference For Surface Heater:

$Iw = T\ (h_1 - h_2) + (T - Q - Sb2)\ (h_2 - h_f)$

Surface Feedwater Heater:

$Sb2\ (h_2) + T\ (h_5) = T\ (h_6) + Sb2\ (h_7)$

Deaerator:

$$Sb3\ (h_f) + (T - Sb2 - Sb3)\ (h_4) + Sb2\ (h_7) = T\ (h_5)$$

Where: h_2 is the enthalpy of the steam at extraction, 75 psig., B.T.U. per lb.

h_f is the enthalpy of the steam at exhaust, B.T.U. per lb.

h_4 is the enthalpy of the feedwater entering the deaerator.

h_5 is the enthalpy of the feedwater leaving the deaerator.

h_6 is the enthalpy of the feedwater leaving the surface heater.

h_7 is the enthalpy of the drains leaving the surface heater.

Sb2 is the quantity of steam to the surface heater, lb. per hr.

Sb3 is the quantity of steam to the deaerator, lb. per hr.

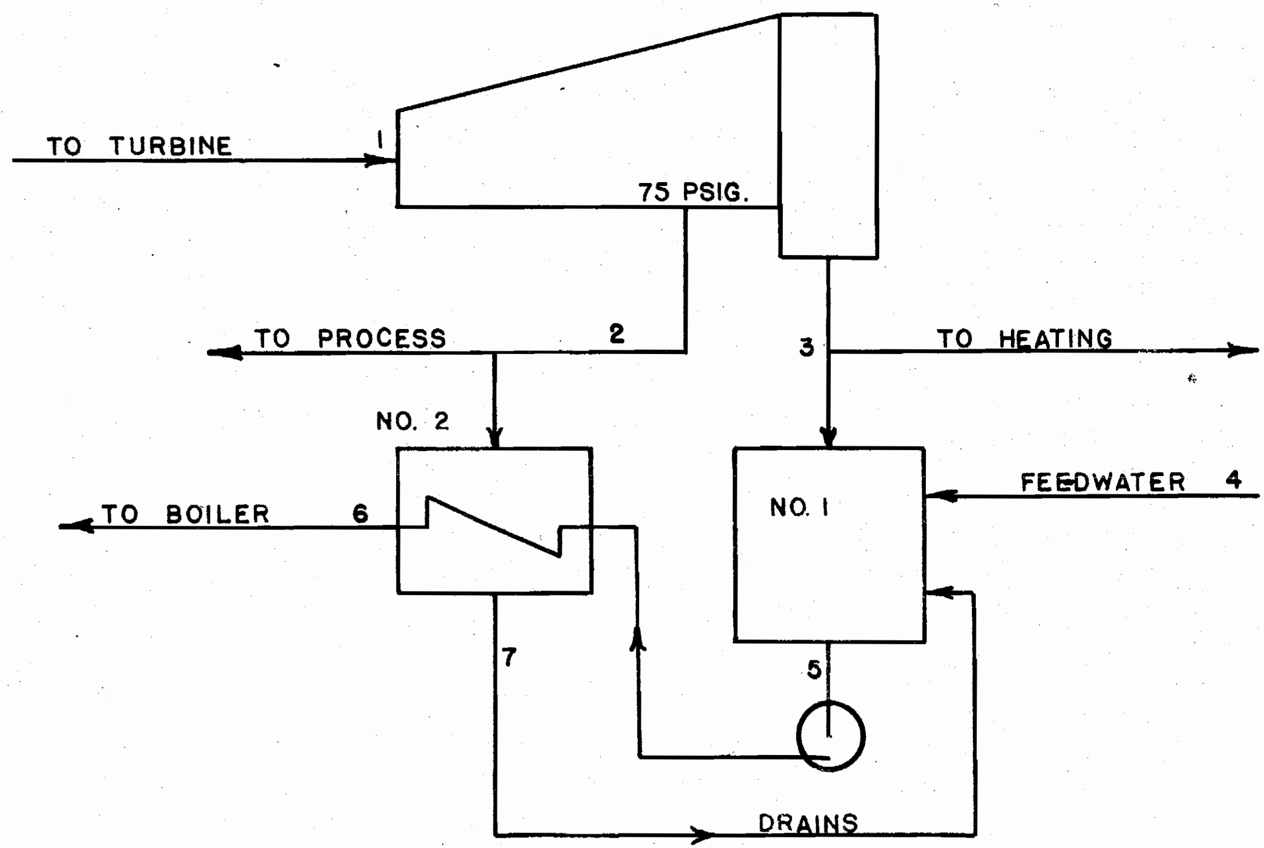

FIG. 40 TWO STAGE FEEDWATER HEATING CYCLE

HEAT CHART FOR DIAGRAM OF TWO STAGE FEEDWATER HEATING CYCLE

	B.T.U. per lb.				
Throttle Conditions	250 psig. 500 F	400 psig. 750 F	600 psig. 825 F	900 psig. 900 F	1200 psig. 950 F
Point 1	———	1389.0	1421.0	1451.4	1466.0
Point 2	———	1265.0	1264.0	1264.0	1252.0
Point 3	———	1186.0	1186.0	1188.0	1178.0
Point 4	———	93.0	93.0	93.0	93.0
Point 5	———	216.8	216.8	216.8	216.8
Point 6	———	277.8	277.8	277.8	277.8
Point 7	———	283.0	283.0	283.0	283.0
Turbine KW	———	5000	6250	7500	9375

Temperature Of Feedwater To Boiler (Point 6) Equals 308 F

EXPANSION FACTOR (1.50)

HEAT CHART FOR DIAGRAM OF TWO STAGE FEEDWATER HEATING CYCLE

B.T.U. per lb.					
Throttle Conditions	250 psig. 500 F	400 psig. 750 F	600 psig. 825 F	900 psig. 900 F	1200 psig. 950 F
Point 1	———	1389.0	1421.0	1451.4	1466.0
Point 2	———	1261.0	1257.0	1257.0	1252.0
Point 3	———	1181.0	1179.5	1181.0	1178.0
Point 4	———	93.0	93.0	93.0	93.0
Point 5	———	216.8	216.8	216.8	216.8
Point 6	———	277.8	277.8	277.8	277.8
Point 7	———	283.0	283.0	283.0	283.0
Turbine KW	———	5000	9375	9375	9375

Temperature Of Feedwater To Boiler (Point 6) Equals 308 F

EXPANSION FACTOR (2.14)

$27.5 \times 10^6 = T\ (1451.4 - 1257.0) + (T\text{-}22{,}000 - Sb2)\ (1257.0 - 1181.0)$

$Sb2\ (1257.0) + T\ (216.8) = T\ (277.8) + Sb2\ (283)$

$Sb3\ (1181.0) + (T\text{-}Sb2 - Sb3)\ (\ 93.0) + Sb2\ (283.0) = T\ (216.8)$

T = <u>110,000</u> lb. per hr

Sb2 = <u>6,900</u> lb. per hr

Sb3 = 11,300 lb. per hr

15. Exhaust Flow With Two Stage Feedwater Heating:

Qe = T - Q - Sb2 - Sb3

Where: Qe is the exhaust flow with two stage feedwater heating, lb. per hr.

Qe = 110,000 - 22,000 - 6,900 - 11,300

Qe = <u>69,800</u> lb. per hr

16. Heat Balance For Three Stage Feedwater Heating, Five Degree Terminal Difference For Both Surface Heaters:

$Iw = T\ (h1 - h_2) + (T - Sb2)\ (h_2 - h_3) + (T - Sb2 - Sb3 - Q)\ (h_3 - h_4)$

High Pressure Surface Heater:

$Sb2\ (h_2) + T\ (h_7) = Sb2\ (h_9) + T\ (h_8)$

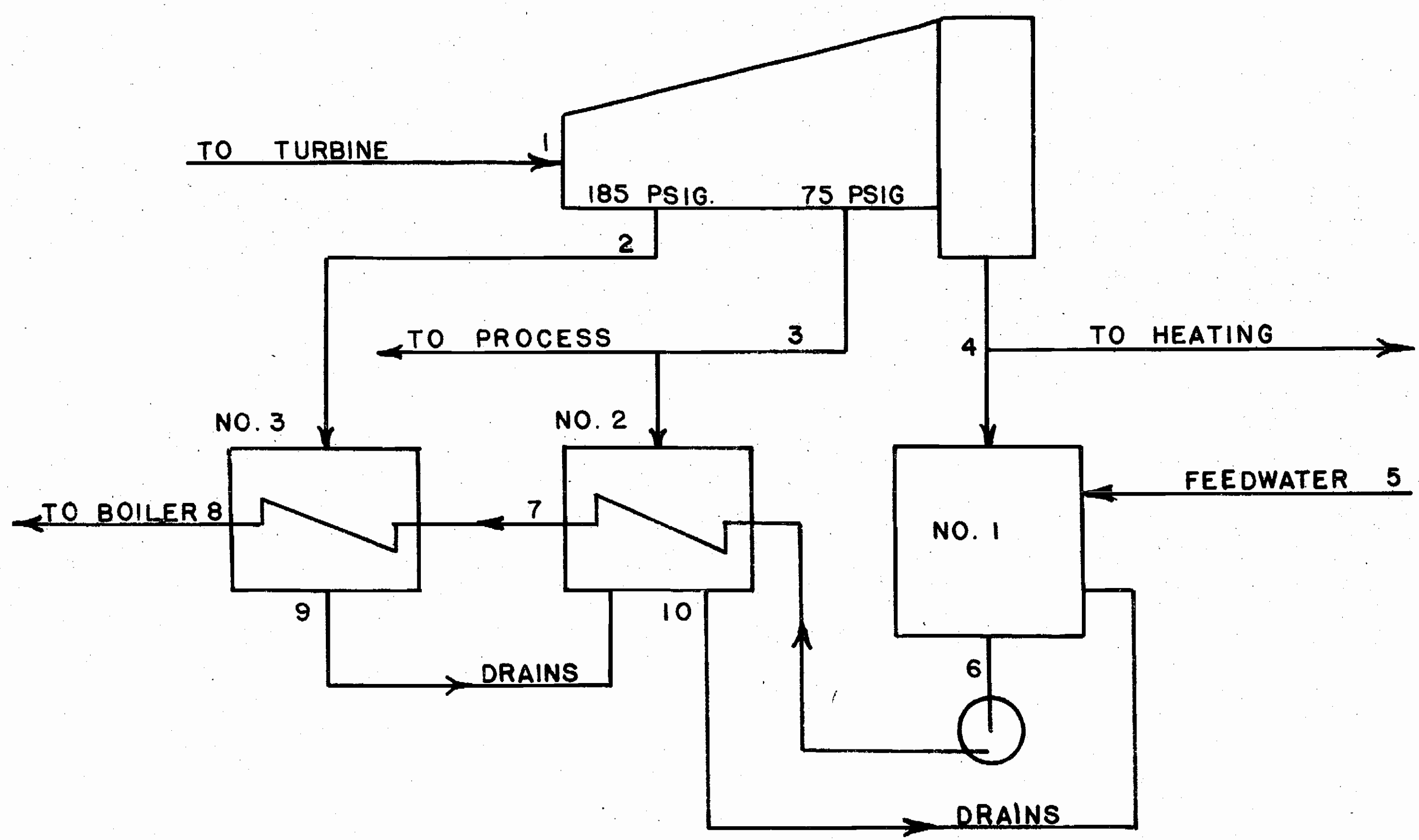

FIG. 41 THREE STAGE FEEDWATER HEATING CYCLE

HEAT CHART FOR DIAGRAM OF THREE STAGE FEEDWATER HEATING CYCLE

	B.T.U. per lb.				
Throttle Conditions	250 psig. 500 F	400 psig. 750 F	600 psig. 825 F	900 psig. 900 F	1200 psig. 950 F
Point 1	———	1389.0	1421.0	1451.4	1466.0
Point 2	———	1327.0	1321.0	1319.0	1312.0
Point 3	———	1261.0	1257.0	1257.0	1252.0
Point 4	———	1181.0	1179.5	1181.0	1178.0
Point 5	———	93.0	93.0	93.0	93.0
Point 6	———	216.8	216.8	216.8	216.8
Point 7	———	277.8	277.8	277.8	277.8
Point 8	———	340.6	340.6	340.6	340.6
Point 9	———	346.0	346.0	346.0	346.0
Point 10	———	283.0	283.0	283.0	283.0
Turbine KW	———	5000	9375	9375	9375

Temperature Of Feedwater To Boiler (Point 8) Equals 368 F

EXPANSION FACTOR (2.14)

HEAT CHART FOR DIAGRAM OF THREE STAGE FEEDWATER HEATING CYCLE

B.T.U. per lb.					
Throttle Conditions	250 psig. 500 F	400 psig. 750 F	600 psig. 825 F	900 psig. 900 F	1200 psig. 950 F
Point 1	-------	1389.0	1421.0	1451.4	1466.0
Point 2	-------	1329.0	1326.0	1325.0	1312.0
Point 3	-------	1265.0	1264.0	1264.0	1252.0
Point 4	-------	1186.0	1186.0	1188.0	1178.0
Point 5	-------	93.0	93.0	93.0	93.0
Point 6	-------	216.8	216.8	216.8	216.8
Point 7	-------	277.8	277.8	277.8	277.8
Point 8	-------	340.6	340.6	340.6	340.6
Point 9	-------	346.0	346.0	346.0	346.0
Point 10	-------	283.0	283.0	283.0	283.0
Turbine KW	-------	5000	6250	7500	9375

Temperature Of Feedwater To Boiler (Point 8) Equals 368 F

EXPANSION FACTOR (1.50)

Low Pressure Surface Heater:

$$Sb3\ (h_3) + T\ (h_6) + Sb2\ (h_9) = T\ (h_7) + (Sb3 - Sb2)\ (h_{10})$$

Deaerator:

$$Sb4\ (h_f) + (Sb3 - Sb2)\ (h_{10}) + (T - Sb2 - Sb3 - Sb4)\ (h_5) = T\ (h_6)$$

Where: h_2 is the enthalpy of the steam at the first extraction stage, 185 psig., B.T.U. per lb.

h_3 is the enthalpy of the steam at the second extraction stage, 75 psig., B.T.U. per lb.

h_f is the enthalpy of the steam at exhaust, B.T.U. per lb.

h_5 is the enthalpy of the feedwater entering the deaerator, B.T.U. per lb.

h_6 is the enthalpy of the feedwater leaving the deaerator, B.T.U. per lb.

h_7 is the enthalpy of the feedwater leaving the low pressure surface heater, B.T.U. per lb.

h_8 is the enthalpy of the feedwater leaving the high pressure surface heater, B.T.U. per lb.

h_9 is the enthalpy of the drains leaving the high pressure surface heater, B.T.U. per lb.

h_{10} is the enthalpy of the drains leaving the low pressure surface heater, B.T.U. per lb.

Sb2 is the quantity of steam to the high pressure surface heater, lb. per hr.

Sb3 is the quantity of steam to the low pressure surface heater, lb. per hr.

Sb4 is the quantity of steam to the deaerator, lb. per hr.

$$27.5 \times 10^6 = T\ (1451.4 - 1319.0) + (T - Sb2)\ (1319.0 - 1257.0)$$
$$+ (T - Sb2 - Sb3 - 22{,}000)\ (1257.0 - 1181.0)$$

$$Sb2\ (1319.0) + T\ (277.8) = Sb2\ (346.0) + T\ (340.6)$$

$$Sb3\ (1257.0) + T\ (216.8) + Sb2\ (346.0) = T\ (277.8) + (Sb2 - Sb3)$$
$$(283.0)$$

$$Sb4\ (1181.0) + (Sb3 - Sb2)\ (283) + (T - Sb2 - Sb3 - Sb4)\ (93) =$$
$$T\ (216.8)$$

T = <u>114,000</u> lb. per hr.

Sb2 = <u>7,350</u> lb. per hr.

Sb3 = <u>6,670</u> lb. per hr.

Sb4 = <u>10,500</u> lb. per hr.

17. Exhaust Flow With Three Stage Feedwater Heating:

$$Qe = T - Q - Sb2 - Sb3 - Sb4$$

$$Qe = 114{,}000 - 22{,}000 - 7{,}350 - 6{,}670 - 10{,}500$$

Qe = 67,480 lb. per hr.

This same procedure was followed for 40 per cent loading of the generator.

18. Fuel Cost Per Kw-hr:

(a) $H_1 = T\ (h_1 - h_w)$

(b) $H_2 = Qe\ (h_f - h_d) + Q\ (h_2 - h_d)$

(c) Turbine Heat Rate, B.T.U. per Kw-hr. $= \dfrac{H_1 - H_2}{Be\ (T.L.)}$

(d) Fuel Rate, lb. per Kw-hr. $= \dfrac{\text{Turbine Heat Rate}}{\text{H.V.F.}}$

(e) Fuel Cost per Kw-hr. $= \dfrac{\text{Fuel Rate} \times \$2.50}{2000}$

Where: H_1 is the heat added to the water in the boiler, B.T.U. per hr.

H_2 is the heat to heating and process, B.T.U. per hr.

h_1 is the enthalpy of super heated steam from boiler, B.T.U. per lb.

h_w is the enthalpy of the feedwater entering the boiler, B.T.U. per lb.

h_f is the enthalpy of the exhaust steam from the turbine, B.T.U. per lb.

h_2 is the enthalpy of the extracted steam for process, B.T.U. per lb.

h_d is the enthalpy of the condensate drains from heating and process steam, B.T.U. per lb.

Qe is the quantity of exhaust steam, lb. per hr.

Q is the quantity of extracted steam, lb. per hr.

Be is the assumed boiler efficiency, 80 per cent.

T.L. is the turbine load.

H.V.F. is the heating value of the fuel, 11,500 B.T.U. per lb.

The cost of the fuel is $2.50 per ton.

For throttle conditions of 1200 psig. and 950 F and a 9375 KW unit operating at an exhaust flow of 77,000 lb. per hr.

(a) $H_1 = 112{,}000\ (1466 - 216.8)$

$H_1 = \underline{140{,}000{,}000}$ B.T.U. per hr.

(b) $H_2 = 77{,}000\ (1178 - 93) + 22{,}000\ (1252 - 93)$

$H_2 = \underline{109{,}050{,}000}$ B.T.U. per hr.

(c) Turbine Heat Rate $= \dfrac{140{,}000{,}000 - 109{,}050{,}000}{0.80\ (8500\ \text{Kw})} = \underline{4550}$ B.T.U. per Kw-hr.

(d) Fuel Rate $= \frac{4550}{11,500} = \underline{0.396}$ lb. per Kw-hr.

(e) Fuel Cost $= \frac{0.396 \times \$2.50}{2000} = \$0.000495$ per Kw-hr.

19. Daily Saving:

D.S. = A (K) ($0.008 - F.C.)

Where: D.S. is the daily saving in dollars.

A is the area under the generated load curve, square inches.

K is the scale constant, 1000 Kw-hr. per square inch.

F.C. is the fuel cost per Kw-hr.

For throttle condition of 900 psig. and 900 F and a 9375 KW turbine with one stage feedwater heating.

D.S. = 173.3 (1000) ($0.008 - $0.00049)

D.S. = <u>$1303.22</u> per day

20. Saving For The Heating Season:

Sh = 180 x D.S.

Where: Sh is the saving for the heating season.

Sh = 180 x $1303.22

Sh = <u>$234,580.00</u>

21. Daily Saving Of Two Stage Feedwater Heating Over One Stage Heating:

S' = D.S.2 - D.S.1

Where: S' is the daily saving of two stage heating over one stage heating.

D.S.1 is the daily saving for one stage heating.

D.S.2 is the daily saving for two stage heating.

S' = $1,355.56 - $1,303.22

S' = <u>$52.34</u> per day.