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ASSISTANCE TO NITROGEN CHEMICALS

OF ZAMBIA LIMITED

DP/ZAM/82/025/11-01

Technical Report*

Prepared for the Government of the Republic of Zambia
by the United Nations Industrial Development Organization,
acting as executing agency for the United Nations Development Programme

Based on the work of Doo Nam, Oh
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ABSTRACT

As part of the United Nations Development Programme (UNDP) Project "Assistance to Nitrogen Chemicals of Zambia Limited", an expert was sent to the Republic of Zambia by the United Nations Industrial Development Organization (UNIDO), executing agency for the project, to assist in resolving problems in the ammonia storage refrigeration systems at the Kafue fertilizer plant of Nitrogen Chemicals of Zambia Limited, one of the subsidiary of Zambia Industry and Mining Corporation Limited (ZIMCO).

In the course of his one year mission (September, 1986 to August, 1987), he has analyzed the problems which cause the refrigeration system not in successful operation since it was installed in 1979, and provided proposals to the company for the successful operation of the system.

In his proposal, he recommended ;

1. To remove the excessive scale formed on the tubes of H.P. ammonia condenser to increase the heat transfer efficiency and thus to reduce overloaded refrigeration capacity in ammonia synthesis and storage unit.

2. To change the refrigeration system in ammonia storage unit with simple modification from the indirect freon refrigeration to the direct ammonia refrigeration system in order to increase refrigeration capacity and to have more smooth operation.

3. and others.

Half of his recommendeations has been already implemented with good effect and the remaining half are under way of implementation by the company.

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INTRODUCTION

Nitrogen Chemicals of Zambia Limited (NCZ), established in 1967, is wholly owned subsidiary of the Zambia Industrial and Mining Corporation Limited (ZIMCO) which is wholly owned by the Government of Zambia.

Nitrogen Chemicals first plant commissioned in 1970 is a coal based plant and produce 50,000 tons of Ammonium Nitrate annually.

The NCZ expansion project which commenced in 1975 and commissioned in 1981 have the capacity of 55.00 tons per annum of prilled Ammonium Nitrate and 142,000 tons per annum of NPK compound fertilizers.

Due to aging of the old plant and various technical problems on the new expansion plant, the overall utilization of the integrated facility is below 40% of the design capacity.

NCZ has established a plan for the rehabilitation of old and expanion plant to achieve a minimum of 85% capacity utilization.
The rehabilitation of the old plant was completed by July 1986 by the Kobe Steel Limited of Japan and rehabilitation of expansion plant is now underway by Klockner Ina of West Germany.

In the interim period, NCZ is directing its effort to improve the production and material efficiencies of the plants. At the moment, one of the major factors contributing to poor material efficiency is the losses of ammonia from the system and storage tanks due to poor performance of the refrigeration units.
This can certainly be improved by proper preventive measures and skilful operation of the refrigeration system.

UNDP/UNIDO assistance was therefore being requested to help NCZ to develop a programme for the successful operation of the refrigeration system.

As part of UNDP project, an expert was sent to the company by UNIDO, executing agency for the project, to assist in resolving problems in refrigeration system.

In the course of his one year mission (September, 1986 - August 1987), he has analyzed the problems which cause the refrigeration system in storage not in successful operation since it was installed in 1979, and provided proposals to the company for the successful operation of the system.

Half of his recommendations has been already implemented with good effect and the remaining half are underway of implementation by the company.

Since it was analyzed the reasons of ammonia losses from the system are mainly due to

1. Poor performance of refrigeration system in ammonia synthesis unit and

2. Its won problems in storage refrigeration system.

The problems in above two area would be analyzed hereafter separately.

1. REFRIGERATION SYSTEM IN AMMONIA SYNTHESIS UNIT

1-1 OBSERVATION

The temperature of ammonia leaving ammonia synthesis unit to storage was observed to be in the range of 12°C - 19°C at the plant operating capacity of 45 - 60%, instead of design temperature of 1°C.

Liquid ammonia of this higher temperature at 16 BAR.G would generate several times of vapor ammonia than design, when the pressure is reduced to the storage pressure of 3.5 BAR.G.

The capacity of refrigeration unit at the storage is not sufficient to handle this big amount of heat.

Log data sheet was prepared, as showing in Annex - 3, for the year of 1981, 1982, 1983 and 1986, and comparison was made against design data.

1-2 FINDING

a) EXCESSIVE SCALE ON THE TUBES OF H.P. CONDENSER

The temperature of ammonia leaving H.P. condenser was 49°C (October, 1986) instead of 39°C in the year of 1982 at the same operating capacity of 43% (Design is 37°C at full load).

Therefore, it was recommended by the expert to open the cover during scheduled overhaul maintenance period (November, 1986) and remove the scale thoroughly.

It was found out that excessive scale was formed on the shell (Cooling water side). The thickness of the scale was about 2 - 4mm.
This thick layer of scale had been playing a role of an insulation material to give high heat transfer resistance between the cooling water and the product ammonia.

The fouling resistance was calculated to be 12 times bigger than fouling factor given by the designer.

It was also found out that the tubes were heavily corroded in some spot, where the thickness was reduced to 2.2-2.4mm from the original thickness of 6mm.

b) IMPROPER FUNCTION OF COMPRESSOR UNLOADING SYSTEM

The refrigeration compressor has three cylinders with double acting function, so there are six suction and discharge valves. The capacity control is achieved by lifting the suction valves by the force of compressed air.

It was found out that one capacity controller has reverse action, while another controller was out of order.
Therefore, only four controllers were in action against six in design.

c) HIGHER SUCTION AND DISCHARGE PRESSURE OF REFRIGERATION COMPRESSOR

Suction pressure of the compressor show higher from initial operation at 2.9-5.0 BAR.G instead of 1.9 BAR.G of design. And discharge pressure was also higher than the corresponding ammonia equilibrium pressure at the condenser outlet liquid ammonia temperature.
This means that some kind of inert gases are existing in the ammonia vapour.

Therefore, inert gas purging was recommended by utilizing ammonia pressure vs temperature equilibrium curve.

1-3 REASONS

a) EXCESSIVE SCALE

The excessive sclae has been formed due to excessive low velocity of cooling water

This condenser is high pressure duty (450 BAR). Due to special consideration of high pressure, it seems to be almost impossible to give higher water velocity in this kind of special configuration of H.P. condenser.

Anyhow, the design water velocity is 0.24 ft/sec.

As general rule, in cooler design, water velocity below 3 ft/sec are never recommended and even there is increasing incentive to go as high as 8 to 10ft/sec. (Perry Handbook 10-38).
Very few waters are acceptable for cooling without excessive scaling when the velocities falls below 1 ft/sec. (Ludwig: Applied Process Design for chemical and petrochemical plant vol 3, Page 87).

b) COMPRESSOR UNLOADING SYSTEM

During precommissioning stage of the plant, the system seems to be not properly adjusted and precommissioned.

c) INERT GAS IN THE SYSTEM

As mentioned before, capacity control of refrigeration compressor is done by lifting suction valve with compressed air.
It was found out that compressed air was leaking contineously to the suction side of compressor through the suction valve lifting device.

This air behave as inert gas in the ammonia system.

1-4 RECOMMENDATION

a) CLEANING TUBES OF H.P. CONDENSER

It was recommended to remove the excessive scale formed on the tubes during scheduled overhaul maintenance period (November, 1986).

After having chemical cleaning on the tubes of H.P. condenser, the temperature of ammonia leaving synthesis to storage system dropped to 1°C of design temperature from previous 12°C - 19°C (at plant operating load of 60%).

Even if there are improvements in operating conditions, it is calculated that only 56% of scale was removed. The reason was due to the finding of severe corrosion on some spot where the thickness had been reduced to 2,2-2,4mm from 6mm of original thickness.
To avoid further corrosion on this weak spot, thorough chemical cleaning was avoided.

b) NITROGEN FLUSHING

For the protection of further excessive scale formation in comparatively short period, it was recommended to flush contineousily with nitrogen to the shell side where cooling water flows.

In order to protect from cooling water back flow into the nitrogen system, when the nitrogen supply is stopped, it is suggested that nitrogen line should be extended high enough to give suffiecient static head to balance the cooling water pressure.

c) FURTHER CLEANING OF H.P. CONDENSER

Since the scale was not completely removed, it is calaculated that H.P. condenser outlet product temperature will go up to 50°C, when the plant is full loade..
This will be excessive heat duty to the next step refrigeration system in storage unit.

Therefore, it is recommended to have further cleaning of tubes with sypecial care to the weak spot, if the plant is to be operated at 100% operating rate.

d) REPAIR OF COMPRESOR UNLOADING DEVICE

Compressor unloading device should be repaired when the plant is shut down, so that the compressor could operate at a full load.

e) PURGE INERT GAS FROM AMMONIA SYSTEM

By utilizing ammonia pressure vs temperature equilibrium curve, inert gas, whenever it exist in excess in the system, should be purged out.

1-5 IMPLEMENTATION

After having implemented most of recommendation on above, the temperature of ammonia leaving synthesis to storage system dropped to 1°C of design temperature from previous temperature of 12°C - 19°C which had caused the overload of storage refrigeration system.

2 REFRIGERATION SYSTEM AT AMMONIA STORAGE UNIT

2-1 INTRODUCTION

It was learnt that FREON refrigeration system in ammonia storage has not yet been in successful operation since it was installed in 1980.
It was also told that trial operation, using ammonia as a refrigerant, was carried out, but it was also failed.

After having observation of the working conditions of the plant and extensive study and analysis of the failure of both cases, the conclusion was made that the direct ammonia refrigeration, with some modification of existing system, could be most appropriate for the company to have successful operation of the system.

2-2 FINDING

Reasons why the FREON refrigeration system in storage unit has not yet been successfully operated are analyzed and considered to be as follows.

2-2-1 POOR PERFORMANCE OF REFRIGERATION SYSTEM IN AMMONIA SYNTHESIS UNIT

The temperature of ammonia leaving synthesis section to storage were much higher than design, and thus force overload of the refrigeration system in storage unit.

Pleaae refer previous chapter for the reasons and countermeasures on this problem.

2-2-2 IT'S OWN PROBLEMS IN STORAGE REFRIGERATION

a) BAD INSULATION ON STORAGE TANK

Cold losses at design conditions due to heat transfer from ambient temperature to cold ammonia through insulation was calculated to be about 37,000 kcal/hr, while design refrigeration capacity of FREON system is only 41,000 Kcal/hr.

Therefore, the system has almost no reserve rooms (41,000 - 37,000 = 4,000 Kcal/hr) for the increased incoming heat from the ammonia synthesis unit.
In case the ammonia temperature is just $1^{\circ}C$ above design temperature (9051 kg/h x 1.14 Kcal/Kg$^{\circ}$C = 10318 Kcal/h) the refrigeration system in storage become overloaded.

In fact, the ammonia from the synthesis unit has been always about $10^{\circ}C$ above design temperature untill the plant overhaul in November, 1986.

The insulation material is specified as polystyrol basis of which conductivity is 0.022 - 0.028 Kcal/h m^2c, while average conductivity of insulation actually measured is about 0.07 Kcal/h $m^{2\circ}C$ due to bad conditions of insulation.

b) COMPLICATED OPERATION OF FREON SYSTEM

Freon system require more careful operational attention than ammonia refrigeration system which the operating staff in the plant are familiar with.

The following are the possible operational problems most probably taking place during initial operation.

b-1 WATER CONTENTS

Water vapor does not affect ammonia, except to modify the pressure temperature relationship, while little amounts of water contents in FREON could cause bigh operational problems.

FREON has a very small dissolving capacity for water, non soluble parts of water is separated in the form of snows and finally in the form of ice, which especially clog the control organs.
The water contents in the FREON, at which this appearance begins, lies in the range of 20 to 30 mg of water in one kg of Freon.

Further to this, acid condition is established when water comes in contact with FREON, and thus corroded the tubes of condenser etc.

b-2 LOSSES OF REFRIGERANT

Ammonia could be easily recharged (just by opening one valve), whenever there are losses due to operational problems.
However, in FREON system it is not so easy as in ammonia system.

In case air exist in FREON system more than allowable contents, then there are possibilities to have safety valve on the condenser burst and cause FREON losses.
In this case, compressor discharge temperature goes high due to high compression ratio and cause machine to stop.

In case spare charge of FREON was all spent due to similar problems as stated above, the plant could not be recommissioned until new spare cahrge arrive from abroad.

In order to avoid this difficulty, FREON system require complicated evacuation procedure, which ammonia refrigeration normally does not require and thus operators often have tempetation to skip it.

Also, there are possibility to skip this procedure due to non availability of vacuum pumps etc.

b-3 MECHANICAL AND INSTRUMENT PROBLEMS

I was told that there were some mechanical problem of compressors and also instrument problems during initial operations.

I could not as yet find out any written detail history on these.

2-3 REVIEW

2-3-1 GENERAL REVIEW

It is considered that there exist two ways to make the storage refrigeration system operationable.

One way is to repair or recondition all the items which are, at present, out of order and go back to the original FREON system. The otherway is to modify the system to the direct ammonia refrigeration system, similar to the old storage liquifier unit.

2-3-2 RECOMMENDATION

Taking into consideration of plant operating conditions and factors listed below, it is recommended to modity the system to the direct ammor refrigeration system.

a) The refrigeration capacity of AMMONIA system is minimum 70,000 Kcal/hr, while FREON system has only 41,000 Kcal/hr.
Therefore, there are more reserve room in operation with ammonia system.

b) Easier in operation with ammonia system, which plant operating staff are well familiar with.

c) Ammonia is always available at plant, while FEON has to be purchased from abroad

d) Modification is simple.
Additional equipments required are essentially new bigger condenser and suction separator, both of which are available at plant.

2-3-3 ANALYSIS

The following are analysis of each equipment to find possible problems due to change of the refrigeration system from FREON to direct AMMONIA system

a) COMPRESSOR 316 K 1201 A/B

CAPACITY

Compressor refrigeration capacity could increase up to 140,000 Kcal/h with ammonia system.

This could be achievable due to the big difference of suction pressure in two different system. The suction pressure in FREON system is 1.94 BAR-A, while 4.56 BAR.A with AMMONIA

However, the capacity of existing motor become bottleneck of the system (only 31.4 kw).

Therefore, the compressor should be operated at half load and thus system could handle only 70,000 Kcal/h in line with existing motor capacity.

DISCHARGE TEMPERATURE

The discharge temperatures with ammonia are expected in the range of 97^{o}C - 125^{o}C.

SETTING OF SAFETY VALVES PSV 1210 AND 1211

It is recommended to set above safety valves at 16 BAR.G

b) CONDENSER 309 E 10

Condenser E1202 is small in size for the refrigeration system with the capacity of 70,000 Kcal/h and above.

Fortunately, there is one spare condenser in storage which KOBE STEEL CO. had supplied as spare unit for the old storage liquefier system

CAPACITY OF 309 E 10

The capacity of condenser 309 E 10 is calculated to be able to handle 400 kg/h of ammonia at 130°C and 14 BAR.G. This is enough to condense ammonia gases from compressor with 75% operation.

C) MOTORS 316 MK 1201 A/B

Motor power requirement at 75% operation of the compressor is calculated about 34Kw, while the capacity of existing motor is 31.4 Kw.

It is recommended to run the compressor at 50% load to protect it's motor from damage.

REMARKS

Compressor capacity control is achievable only at 25%, 50%, 75% and 100%.

2-4 ARRANGEMENT

a) PLAN-1 (REFRIGERATION CAPACITY 70,000 Kcal/h)

Storage (316V1201) - Separator (New) - Compressor (316 K1201 A/B at 50% operation) - Condenser (309 E10)- Receiver (309 V21) - Old ammonia storage.

This plan is the simplest. Additional equipment required are only condenser and separator, both of which are available at plant.

b) PLAN-2 (REFRIGERATION CAPACITY 70,000 Kcal/h

Same as PLAN-1, except liquefied ammonia goes to new storage tank.
Additional equipments required are one liquid ammonia receiver, level control system and automatic inert gas vent system plus equipments required for PLAN-1

c) PLAN-3 (REFRIGERATION CAPACITY 105,000 Kcal/h

Same as PLAN-2, except compressor operation at 75% load. Additional equipments required are two new motors (48 kw each) plus equipments required for PLAN-2

2-5 SUGGESTION

Considering long equipment delivery of new purchase order, PLAN-1 is recommended for immediate implementation. Afterwards, PLAN-2 OR PLAN-3 could be implemented for better service.

3 TRAINING

The expert has explained to the operator the reasons and countermeasures of the problems of refrigeration system.

Thereby, the operators know at present the way how to avoid these kinds of problems. The following are major items:

a) The reasons of excessive scale formation on the tubes of H. P. condenser

The excessive scale has been formed due to excessive low velocity of cooling water. The condenser is high pressure duty (450 Bar).

Due to special consideration of high pressure, it seems to be almost impossible to give higher water velocity in this kind of special configuration of H. P. condenser (the design water velocity is 0.24 ft/sec).

As a general rule, in cooler design, water velocity below 3 ft/sec are never recommended and even there is increasing incentive to go as high as 8 to 10 ft/sec (Perry Handbook 10 - 38).

b) The countermeasures on above

The expert recommended the company to have chemical cleaning of tubes and to have continuous nitrogen flushing into the water to make turbulance so that rapid scale formation could be protected.

The company has implemented the recommendations and the operators currently check the operating condition of refrigeration system on a regular basis, since they know wholly the reasons why this checking is required for safe and effective operation through the explanation of the expert.

c) Purging of inert gas system

The expert has explained how the inert gas has been introduced through the compressor suction control valve lifting device and explained the way how to purge effectively by using pressure vs. temperature equilibrium curve on ammonia. Since then, inert gas has been effectively purged out and thereby the burden of refrigeration compressor was reduced.

d) Process calculation

The expert has explained how to provide heat balance and material balance of refrigeration system. The calculation sheets are attached to the report as an Annex.

e) Complicated operation of freon refrigeration system

The expert has explained the several points of attention for safe operation of freon system. Please refer 2-2-2-b, on page 15 and 16.

ANNEX - 1, JOB DESCRIPTION

POST TITLE: EXPERT in Operation and Maintenace of Refrigeration System

DURATION: 12 MONTHS

DUTY STATION: KAFUE, ZAMBIA.

PURPOSE OF PROJECT" The project is to assist the Nitrogen Chemicals of Zambia Limited in providing a proposal for the successful operation of refrigeration system in the plant.

DUTIES:

Specifically, the expert will be expected to

1. Evaluating presently applied system of operating and maintaining refrigeration system.

2. Advising the national counterpart personnel in their day-to-day practice of refrigerating equipment operation.

3. Working out a proposal for the successful operation of refrigeration system.

4. Training a number of counterpart personnel

5. Preparation of progress and terminal report in accordnace with UNDP policies and UNIDO requirements.

QUALIFICATION:

University Degree in Mechanics or equivalent experience not less than 20 years practical experience at fertilizer plants and not less than 5 years of the recent experience in operation and maintenance of refrigeration system at fertilizer plants.

ANNEX – 2

PERFORMANCE EVALUATION ON THE EFFECT OF CHEMCIAL CLEANING OF H.P. CONDENSER

1. DESIGN DATA OF H.P. CONDENSER

- Heat duty : 5739000 Kcal/h
  - Desuperheating duty: 2624000 Kcal/h
  - Condensing duty : 31150.0 Kcal/h
- Heat transfer surface Area : 720 M^2
- Weighted ΔT : 28.7°C
  - Product temp : 148°C → 37°C
  - C.W. temp : 37°C ← 28°C
  - LMTD : 40°C, Weight Factor = $\frac{28.7}{40}$ = 0.7175
- Fouling factor on shell (C.W.) : 0.0004 M^2 h°C/Kcal
- Fouling factor on tube (Product) : 0.0002 M^2h°C/Kcal
- C.W. flow Rate : 638000 kg/h
- Sectional Area of shell : 1.225m x 2.260 m
  = 2.7685 m^2
- Area holding by the Tubes : (24mm O.D x 800 each)
  = 3.14 $(0.012)^2$ x 800
  = 0.36 m^2
- Area for C.W. pass : 2.7685 – 0.36 = 2.4085 m^2
- C.W. velocity = 638 m^3 ÷ 2.4085 m^2
  = 265 m/h = 4.4 m/min.
  = 0.0735 m/sec
  = 0.24 ft/sec

2. REVIEW AND ANALYSIS

2.1 DESIGN OVERALL HEAT TRANSFER Coeffecient (Ud)

$Q = UA\Delta T$

From Design Data A= 5739000 Kcal/H

A= 720 m^2

ΔT= 28.7°C

: Ud = 278 Kcal/hm²°C

- Heat trnasfer coefficient of clean tube (U d.c.)

$$\frac{1}{Ud} = \frac{1}{Ud.c} + fi + fo$$

From Design data fi = 0.002 hm²°C/Kcal

$$\frac{1}{278} = \frac{1}{Ud.c} + 0.0002 + 0.0004$$

$$\therefore \frac{1}{Ud.c} = 0.003 \qquad Ud.c = 333\ Kcal/hm^2{}^\circ C$$

2.2 Fouling Resistance before cleaning

a) Operating Data (86.9.27) collected by Mr. Loiacono

- Plant operating capacity : 50% of design
- Product temp : $148^\circ C \rightarrow 37.5^\circ C$
- C.W. Temp : $26.5^\circ C \leftarrow 24^\circ C$
- LMTD = $49^\circ C$
- Weighted $\Delta T = 49^\circ C \times 0.7175 = 35^\circ C$
- C.W. flow = $5739000\ Kcal/h \times 0.5 \div (26.5 - 24)^\circ C$
  $= 1148000\ kg/h$
- C.W velocity $= 1148\ m^3/h \div 2.4085\ m^2$
  $= 0.132\ m/sec = 0.43\ ft/sec$

b) Overall heat transfer coefficient before cleaning (Ub.c) at 50% operating Rate

$Q = UA\Delta T$

$5739000\ Kcal/h \times 0.5 = Ub.c\ 720m^2 \times 35^\circ C$

$\therefore Ub.c = 114\ Kcal/hm^2{}^\circ C$

c). Heat transfer coefficient of C.W side (ho) at 100% operating Rate

- Assuming same amount of C.W used at 100% operating Rate
  $\therefore 1148M^3/h$
- C.W. flow rate in $lb/sec\ Ft^2$

  $= 1148000 \times 2.2\ lb/h \div (2.4085 \times 10.76\ ft^2)$
  $= 97455\ lb/ft^2\ h = 27\ lb/sec\ ft^2$
- Equivalent Tube dia(De)

$$De = 4Rh = 4\ \frac{\text{Flow Area}}{\text{wetted perimeter}}$$

$$= 4\ \frac{2.4085}{\pi \times 0.024 \times 800} = 0.16\ m = 6.3\ inch$$

Using fig 5.3 with De = 6.3 inch

$G = 27\ lb/sec\ ft^2$

ho at $200\ ^\circ F \fallingdotseq 160\ BT^u/h\ ft^2\ {}^\circ F$

Using Fig 5.4 for temperature at $80^\circ F$

heat at 80^{o}F = 160 x 0.6 = 96 BT^{u}/h ft^{2o}F
= 468 Kcal/h m^{2o}C

d) Heat transfer Coefficient of product side (hi) at 100% operating Rate

$$\frac{1}{Ud.c} = \frac{1}{hi} + \frac{1}{ho} \qquad \frac{1}{331} = \frac{1}{hi} + \frac{1}{468}$$

: hi = 1163 Kcal/hm^{2o}C

e) Heat transfer Coefficient of product side (hi) at 50% operating Rate

Using Fig 10-47

Hi at 50% rate = 660 Kcal/hm^{2o}C

f) Fouling Resistance Before cleaning operated at 50% Rate (fo.b.c)

$$\frac{1}{Ub.c} = \frac{1}{hi \text{ at } 50\%} + \frac{1}{h_o} + fi + fo.b.c$$

$$\frac{1}{114} = \frac{1}{660} + \frac{1}{468} + 0.0002 + f\,o.b.c$$

: fo.b.c = 0.00492

This is about 12 times higher than design figure of 0.0004 m^{2} h^{o}C/Kcal

2-3 Fouling Resistance after cleaning

a) Operating data

- plant operating Rate : 54% of design
- Product Temp : 148^{o}C → 32^{o}C
- C.W. Temp : 29^{o}C ← 27^{o}C
- LMTD : 36^{o}C
- Weighted ΔT = 36^{o}C x 0.7175 = 25.8^{o}C
- C.W. flow = 5739000 x 0.54 ÷ (29 - 27)
  = 1550000 Kg/h
- C.W. velocity = 1550 m^{3}/h ÷ 2.4085 m^{2}
  = 0.179 m/sec = 0.586 ft/sec

b) Overall heat transfer Coefficient after cleaning

$$U_{a.c} = 5739000\ \text{Kcal/h} \times 0.54 \div (720 \times 25.8)$$
$$= 167\ \text{Kcal/h m}^2\,{}^{o}C$$

c) Fouling Resistance after chemical cleaning

assumption was made hi at 54% same as
hi at 50% of 660 Kcal/hm^2 ^{o}c

$$\frac{1}{U_{a.c}} = \frac{1}{\ }\text{ at 54\%} + \frac{1}{ho} + fi + fo.a.c.$$

$$\frac{1}{167} = \frac{1}{660} + \frac{1}{468} + 0.0002 + fo.a.c$$

: fo.a.c = 0.00214

Above data shows that fouling resistance after cleaning is still 5 times higher than design resistance.

2 -4 Improvement by chemical cleaning

$$\frac{fo.b.c - fo.a.c}{fo.b.c} = \frac{0.00492 - 0.00214}{0.00492} = 56.5\%$$

2 - 5 Expected condenser outlet product temp at 100% operating rate with current fouling resistnance

a) Overall heat transfer Coefficient at 100% operating Rate with current fouling resistance

$$\frac{1}{U}\text{ at 100\%} = \frac{1}{hi}\text{ at 100\%} + \frac{1}{ho} + fi + fo.a.c.$$

$$= \frac{1}{1163} + \frac{1}{468}\quad 0.0002 + 0.00214$$

$$= 0.00534$$

: U at 100% = 187 Kcal/h m^2 ^{o}C

b) Expected Condenser outlet product Temperature

- Assumption was made C.W. temperature of 26^{o}C will rise up to 30^{o}C
- product temp : 148^{o}C → X^{o}C

C.W. temp : 30^{o}C ← 26^{o}C

118^{o}C y^{o}C

$Q = UA\Delta T$

$5739000\ Kcal/h = 187\ Kcal/hm^2{}^oC \times 72\text{?}\ m^2 \times \Delta T$

$\therefore \Delta T = 42.6^oC$

$LMTD = \frac{\text{weighted } \Delta T}{0.7175} = 59.4^oC$

FROM LMTD 59.4^oC, GTTP 118^oC $\rightarrow y = 23.5^oC$

$\therefore X = 23.5 + 26 = 49.5^oC$

Condenser outlet product temperature of 50^oC is too high for refrigeration system. Therefore, it is recommended to have further cleaning of tubes before the plant is to be loaded at 100% operating Rate.

3. EVALUATION

3.1 C.W. Velocity

Design cooling water velocity is 0.23 ft/sec.

a) In view of scale formation

- The excessive scale has been formed due to excessive low velocity of cooling water.
- This condenser is high pressure duty (450 bar)
  Due to special consideration of high pressure, it is almost impossible to give higher water velocity in this kind of configuration.
- As a general rule, in cooler design, water velocities below 3 ft/sec are never recommended and there is increasing incentive to go as high 8 to 10 ft/sec (Perry : Hand Book 10-38)
- Very few waters are acceptable for cooling without excessive scaling when the velocities falls below 1 ft/sec.
  (Ludwig: Applied process design for chemical and petrochemical plants Vl. 3 P. 87)

b) in view of heat transfer resistance

Heat transfer resistance analysis table

	h	$r = \frac{1}{h}$	%
Product side heat transfer	660	0.00151	25%
C.W. side heat transfer	468	0.00214	36%
Porduct side fouling		0.0002	3%
C.W. side fouling		0.00214	36%
Total		0.00599	100%

As shown in above table heat transfer resistance in this condenser are mostly from cooling water side: namely 36% from cooling water heat transfer resistance mainly due to low velocity and 36% from cooling water fouling effect due to low velocity.

If the cooling water velocity could increase to 8 ft/sec, then heat transfer coeffecient become 1500 Btu/h ft^{2}°F (7320 Kcal/h m^{2}°C.)

Provision for the plant operation of 100% Rate

As it was shown in above calculation, condeser outlet product temperature is expected to go up to 50°C with current scale condition. Therefore it is recommended to have another cleaning before plant be loaded at full capacity.

LOG SHEET

after chemical cleaning

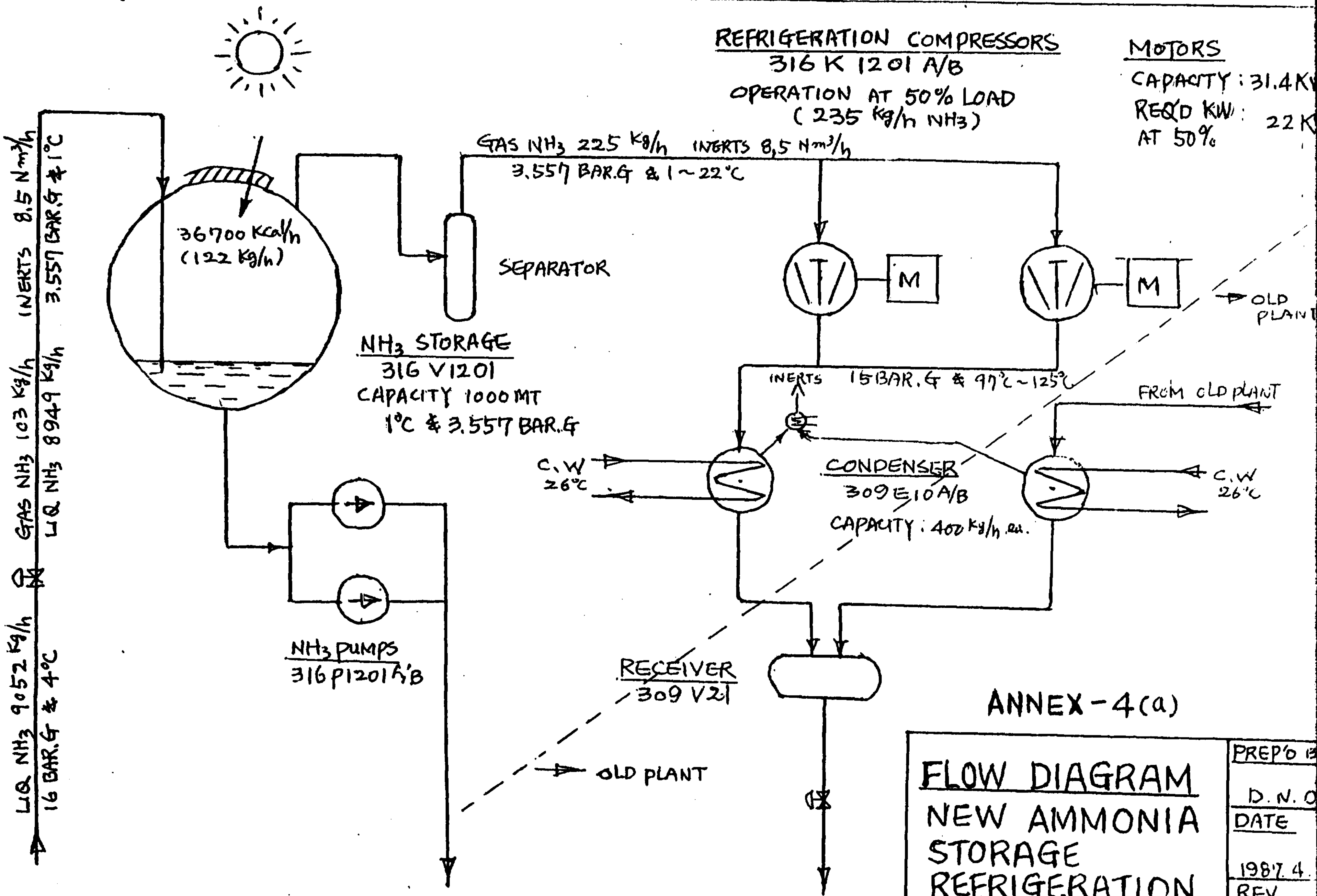
DATE	PLANT CAP.	TEMP (°C) H.P. COND OUT TR1225	TEMP (°C) NH_3 TO STORAGE TR1226	TEMP (°C) NH_3 AT REC TR1227	TEMP (°C) COMP SUC TR1235	PRESS (kg/m^2) Suction PIC1215	PRESS (kg/m^2) Suction Pl 1225	PRESS (kg/m^2) Suction Pl 1226	PRESS (kg/m^2) Dis-charge Pl 1227	PRESS (kg/m^2) Receiver Pl 1231	TEMP AT REC (°C) Tl 1243	COMP LOAD	C.W. TEMP (°C) In	C.W. TEMP (°C) Out
13.1.87	50%	30.5	-2	-4	-2	2.75	2.9	2.6	15	15.5	38.5	66%	26	28
21.1.87	54%	32	1	-1	0	3.35	3.5	3.3	16	16.5	41	66%	27	29

ANNEX - 3 LOGSHEET

OPERATING CONDITIONS OF SYNTHESIS

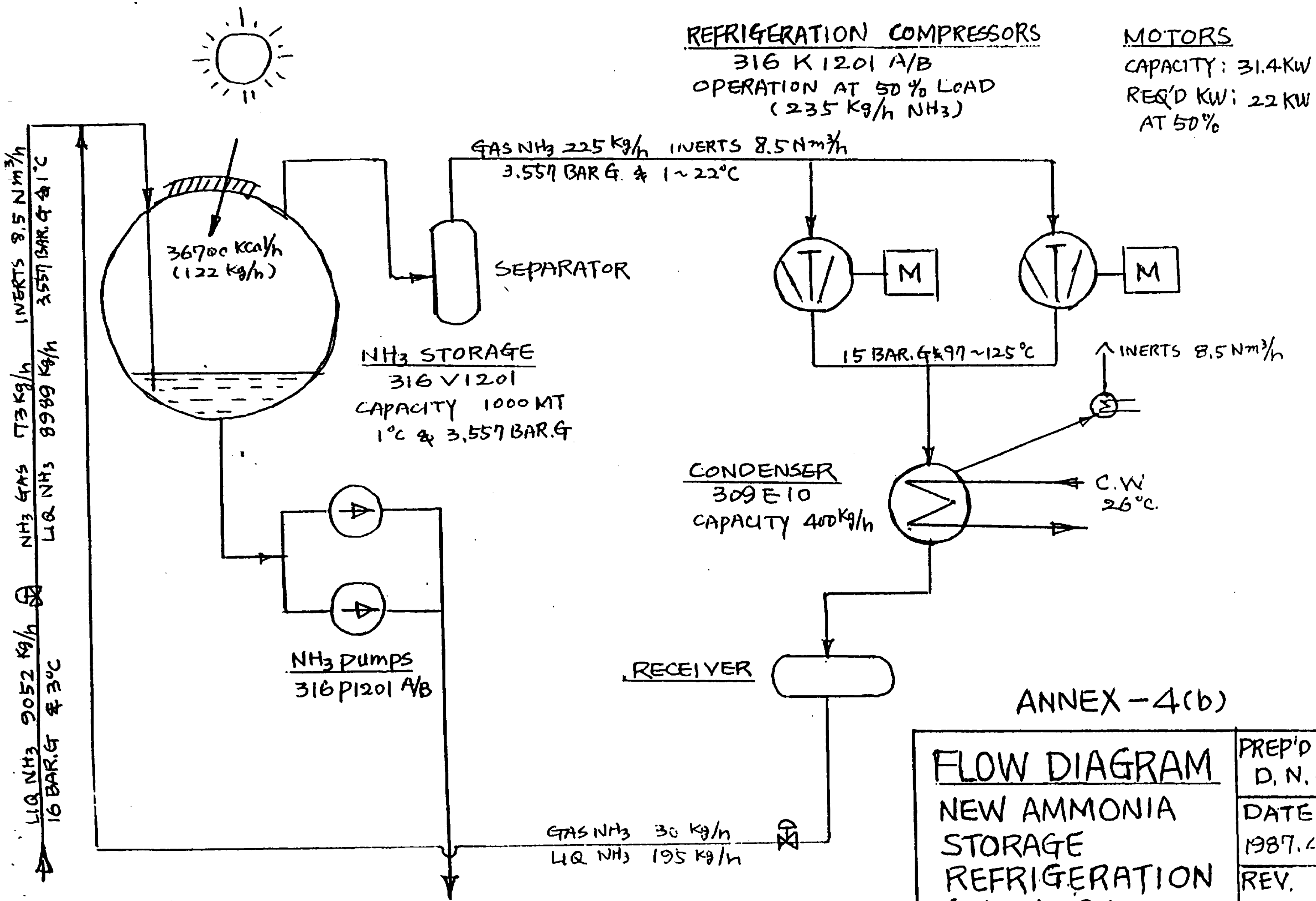
REFRIGERATION SYSTEM

DATE	PLANT ROAD (%) FR1203	TEMP (°C) HP COND OUT TR1225	NH3 TO SPER T TR1226	NH3 AT LP REC TR1227	COMP SUCTION TI 12235	PRESS AT REG COMP (KG/M^2 g) SUCTION PIC 1215	PI 1225	PI 1226	DISCHG PI 1227	RECEIVER PI 1231	TEMP (oC) AT REC TI 1243	COMPR CAPACITY (%)	REMARKS
Design	100	37	1		-10		1.96		14	14	35		
81.12.24 15.00	46	39	8.2	6.5	0	2.7	2.7	2.7	13	-	35		
81.12.30 15.00	49	39	9	11	2	3.5	3.0	3.0	15	14.5	32		
82.6.1 15.00	61	41	8	5.5	-1.5	2.9	2.7	2.7	11.2	10.6	25		
82.9.30 15.00	43	39	11	10	4	3.75	3.5	3.5	11.8	11.3	30.5		
82.6.20 15.00	44	32.5	5.5	4	-1.5	2.7	2.6	2.6	9.4	7.0	23.5		
83.6.21 15.00	46	38	10	8	1.0	3.1	3.1	3.2	14.5	12	37		
83.9.21 15.00	44	34	13	11	5.0	3.5	3.5	3.5	14.7	14.0	42		
86.6.15 15.00	73	47	22	19.5	9	74	4.9	5.5	14.2	13.5	29	100	
86.5.26 16.00	61	45	19	17	7.0	4	4.4	4.9	14	11.5	29	100	
86.9.20 15.00	45	49	17	15	7	4	4.2	5.0	14.6	14.3	29	50	
86.9.27 15.00	52	46	12	10	1	3.3	3.5	4	16.5	16	29	100	



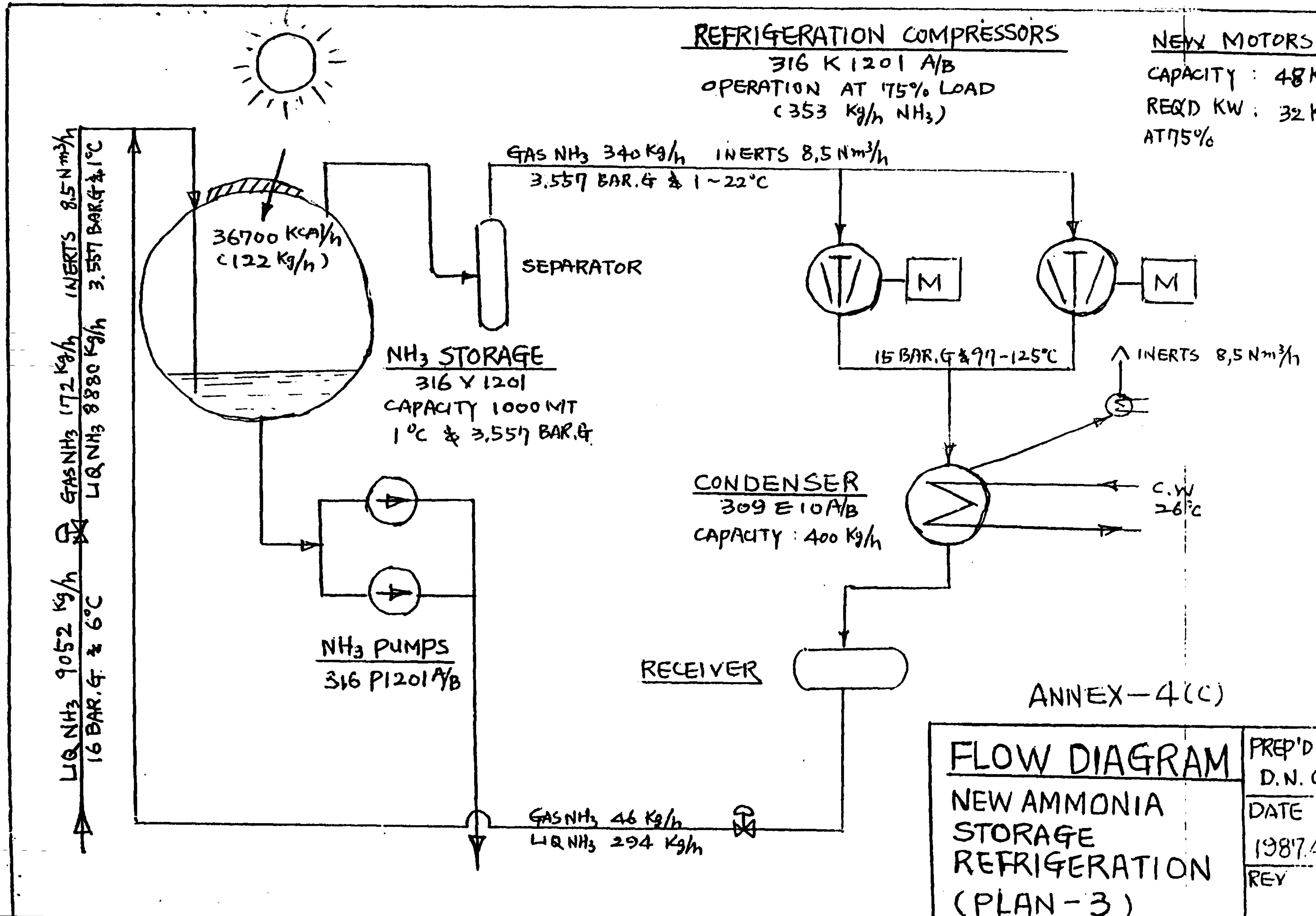
ANNEX-4(a)

FLOW DIAGRAM	PREP'D BY
NEW AMMONIA STORAGE REFRIGERATION (PLAN-1)	D. N. O
	DATE
	1987.4.
	REV



ANNEX-4(b)

FLOW DIAGRAM	PREP'D BY D. N. OH
NEW AMMONIA STORAGE REFRIGERATION (PLAN-2)	DATE 1987.4.
	REV.



ANNEX-4(C)

FLOW DIAGRAM	PREP'D D.N.C
NEW AMMONIA STORAGE REFRIGERATION (PLAN-3)	DATE 1987.4
	REV

ANNEX – 5

CALCULATION SHEET

REFRIGERATION SYSTEM AT STORAGE

A. Evaluation of Heat losses through storage insulation

1. Measurement

a) February 6th

TIME	PRESS	TEMP	REMARKS
11:19	3.43 Kg/cm^{2}g	Top 22°C Bottom 1°C	Storage level:18% Weather clear until 13.00 (34°C)
16:19	3.62	"	Rain 13.00 - 15.000(28°C)

Average increase: 0.038 Kg/cm^2/n.

b) February 9th

TIME	PRESS	TEMP	REMARKS
08:50	3.43 Kg/cm^{2}g	Top 22°C Bottom 0°C	Level : 11% Ambient Temp:34°C
10:50	3.57 "	"	Weather was clear
12:50	3.61 "	"	
15:40	3.66 "	Bottom 1°C	
16:40	3.73 "	Bottom 2°C	

Average increase : 0.0375 Kg/cm^2/h.

c) MARCH 3RD

TIME	PRESS	TEMP	REMARKS
10:20	3.55Kg/cm^{2}g	Top 22°C Bottom 1°C	Storage Level: 11%
13:25	3.65 "	Bottom 1.$^{\circ}$C	Ambient Temp: 34°C
16:50	3.72 "	Bottom 2°C	

Average increase : 0.0319 Kg/cm^2/h.

2. Evaluation of Heat less (based on measurement on February 9th)

a) Data

Storage volume : 1710 m^3

Volume of liquid at 11% : $66m^3$
Insulation thickness : 5cm
Pressure increase : 0.0375 $Kg/cm^2/h$.

b) Conductivity of insulation material

* Gas volume at 4.73 Kg/cm^2 a. and 12^oC

$$1644m^3 \times \frac{4.73}{1} \times \frac{273}{273+12} = 7449\ Nm^3$$

* Gas volume at 4.43 Kg/cm^2.a. and 11^oC

$$1644m^3 \times \frac{4.43}{1} \times \frac{273}{273+11} = 7001\ Nm^3$$

* Volume increase = 7449 - 7001

$$= 448\ Nm^3$$

$$= 448 \times \frac{17}{25.8} = 295\ Kg$$

REMARKS:
Molecular volume of ammonia : 25.8
(See perry Hand book 3-273)

* Assuming that average surface area temperature to be the arithmetic mean value of the top and bottom temperature.

$$tm = \frac{22+1}{2} = 11.5^oC$$

* Heat losses through insulation

Q= 295 kg/8hr x 301 Kcal/kg
= 11099 Kcal/h

* Average conductivity of the insulation

$$Q=KA\frac{\Delta T}{\Delta X} \qquad A= 3.14\ (14.82)^2 = 629\ m^2$$

$$11.099\ Kcal/h = K.692\ m^2\ \frac{11.5^oC}{0.05m}$$

$$\therefore\ K= 0.06973\ Kcal/h\ m^oC$$

c) <u>Heat losses at design condition</u>

inside temp : 1^oC

outside temp: 39^oC

$$Q = KA.\frac{\Delta T}{\Delta X} = 0.06973 \times 692 \times \frac{39-1}{0.05}$$

$$= 36672\ Kcal/h$$

We see that heat losses through insulation is very big and almost equivalent to the refrigeration capacity using FREON-22 as a refrigerent (41.000 Kcal/n).

B. <u>Refrigeration compressor capacity (316 K1201)</u>

1) <u>Design Data</u>

Inlet volume	:	$157\ m^3/h$
Compressor speed	:	1450 r.p.m.
Power reqirement	:	25.3 KW
Belt losses	:	1.2 KW
Total power reirement	:	26.5 KW
Motor power	:	31.4 KW
Maximum allowable cylinder press	:	$21\ Kg/m^2.g$
Maximum allowable temperature	:	140^oC.

<u>Compressor capacity at 4.557 Kg/m^2.a and 1^oC</u>

2) $157 \times \frac{4.557}{1} \times \frac{273}{273+1} = 715\ Nm^3/h$

= 27.7 Kg-mol/h

= 471 Kg/h of ammonia

C. Motor capacity

1. Data to be used

P1 = 4.9 Kg/cm^2.a. (PRSAHH = 4.85 Kg/cm^2.a)

P2 = 17.0 Kg/cm^2.a

Cp/cv = 1.317

2. Power Requirement

$$BHP = \frac{144}{33.000} \left(\frac{K}{K-1}\right) P_1 \cdot V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}} - 1\right] (Lo) \quad (F_2)$$

$$= \frac{144}{33000} \left(\frac{1.317}{1.317-1}\right) (4.9)\ (14.223)\ (92.4) \left[\left(\frac{17}{4.9}\right)^{\frac{0.317}{1.317}} - 1\right] (1.21)\ (1.05)$$

= 148.3 (1.349-1) = 51.757 Horsepower

= 38.6 KW

Assuming Belt loss : 1.9 KW

Motor eff : 0.90

$$\text{Motor capacity} = \frac{38.6+1.9}{0.90} = 45 \text{ Kw}$$

3. Motor load

a) Motor load at 75% operation of compressor

45 KW x 0.75 = 33.75 Kw

b) Motor load at 50% operation of compressor

45 Kw x 0.5 = 22.5 Kw

Above analysis shows that existing motor (31.4 Kw) is small in size for the 75% operation of compressor, but, enough for the 50% operation of compressor.

D. Condenser capacity (309E10)

1. Design Data

	SHELL	TUBE
Medium	NH_3	Water
Quantities	290 Kg/h	22 Ton/h
Noncondensable	8.1 Nm^3/h	
Gravity	8.7 Kg/m^3	1.000
Latent Heat	255 Kcal/kg	80
Temp in	75 ^0C	28 ^0C
Temp out	35 ^0C	32.8 ^0C
Operating press	15 Kg/cm^2g	2
Fouling resistance		0.0001 $m^2h^0C/Kcal$
Heat Echonged	106.000 Kcal/h	
M.T.D. (Corrected)	7.34 ^0C	
Transfer rate	340 $Kcal/m^2h\ ^0C$	
Surface area	42.3 m^2	

2. Review

a) Maximum expected compressor oulet temp

$$T_2 = (273 + 22) \left(\frac{17}{4.45}\right)^{\frac{0.317}{1.317}} = 295 \times 1.3807$$
$$= 407\ ^0K$$
$$= 134\ ^0C$$

b) Desuperheating heat at worst condition

290 Kg/h x 0.52 x (134 - 35) = 14929 Kcal/h

c) Latent heat
290 Kg/h x 268 Kcal/kg = 77720 Kcal/h

d) Total heat
14929 + 77720 = 92649 Kcal/h

e) Area required for desuperheating

* C.W outlet temp

$14929 \text{ Kcal/h} = 22000 \text{ Kg/h} \times 1 \text{ Kcal/Kg} \times \Delta T$

$\Delta T = 0.68 = 0.7\ ^{o}C$

NH_3 134 °C → 35°C

C.W 28.7°C ← 28°C

105.3°C   7°C

$LMTD = 36.5^{o}C$

Assuming $U = 115 \text{ Kcal/h } m^2\ ^{o}C$

$$A_1 = \frac{Q}{UOT} = \frac{14929}{(115)\ (36.5)} = 3.55\ m^2$$

f) Area required for condensing

* C.W. outlet temperature

$92649 \text{ Kcal/h} = 22000 \text{ Kg/h} \times 1 \text{ Kcal/Kg} \times \Delta T$

$\Delta T = 4.2^{o}C$

$\therefore T = 28 + 4.2 = 32.2^{o}C$

NH_3 35 → 35

C.W. 32.2°C ← 28.7°C

2.8°C   6.3°C

$$LMTD = \frac{3.5}{0.8109} = 4.3^{o}C$$

Assuming $U = 680 \text{ Kcal/h } m^{2o}C$

$$A_2 = \frac{77720}{680 \times 4.3} = 26.5\ m^2$$

g) Total Area required

$A = A_1 + A_2 = 3.55 + 26.5 = 30.05\ m^2$

Since Actual condenser surface area is $42.3\ m^2$, condenser seems to be overdesigned.

h) Condenser capacity

* oversize factor = $\frac{42.3}{30.05} = 1.40$

* amount of ammonia which could be condensed

290 Kg/h x 1.40 = 406 Kg/h

Above Review shows that condenser capacity is sufficient for 75% operation of compressor (471 Kg/h x 0.75 = 353 Kg/h NH_3)

E. Expected highest temperature of ammonia feed to the storage for plan -1

1. Case I (ammonia plant at 100% operation)

* Ammonia vapour due to heat through insulation; 36700 Kcal/301 Kcal/Kg = 122 Kg/h.

* Maximum allowable flashing vapour from plant ammonia feed to storage: 471 Kg/h x 0.5 - 122 = 135.5 Kg/h

* Expected temperature

$135.5\ Kg/h \times 301\ Kcal/kg = 9051\ Kg/h \times 1.14\ Kcal/kg\ ^oC \times \Delta t$

$\Delta t = 3.95^oC$

$\therefore\ t = 1^o + \Delta t = 4.95^oC = 5^oC$

2. Case 2. (ammonia production 125 Ton/day)

$135.5\ Kg/h \times 301\ Kcal/Kg = 125000/24 \times 1.14 \times \Delta t$

$\therefore \Delta t = 6.87^{o}C$

$\therefore t = 1 + 6.87^{o}C = 7.87^{o}C = 8^{o}C$

F. <u>Expected compressor discharge Temperature</u>

1. Case one (storage gas temp $1^{o}C$)

$T_1 = 273 + 1 = 274\ ^{o}K$

$P_1 = 4.557$ BAR.A

$P_2 = 16$ BAR.A

$$T_2 = 274 \left(\frac{16}{4.557}\right)^{\frac{0.317}{1.317}} = 274 \times 1{,}35$$

$$= 370\ ^{o}K$$

$$\therefore t = 97^{o}C$$

2. Case two (storage top gas temp $22^{o}C$)

$T_1 = 293 + 22 = 295^{o}K$

$$T_2 = 295 \left(\frac{16}{4.557}\right)^{\frac{0.317}{1.317}} = 398^{o}K$$

$$\therefore T = 125^{o}C$$