“Steady State and Dynamic Simulation of Cryogenic system for Superconducting Linear Accelerator (LINAC) at TIFR”

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The Outline of the Presentation

- Introduction of Superconducting LINAC and its Cryogenic system
- Steady State Simulation of Linde TCF-50S refrigerator plant using ASPEN HYSYS
- Characterization of the Expander, Heat exchanger, Valve & Dewar
- Dynamic Simulation of Linde TCF-50S using Hysys
- Pressure Drop and Heat load calculation of Cryogenic Distribution system.
- Upgradation of TCF 50S at TIFR
- Conclusion

Introduction

Particle Accelerator is essential tool in Nuclear Physics, Condensed matter physics, Radiation Physics, Material science, Bio-Physics, Bio-medical and Energy generation.

High accelerated beam of Heavy ions is possible with *Superconducting LINAC.*

Why superconducting ?

$Q=f/\Delta f=\omega U/P$; Quality factor

$P \sim \frac{1}{2} R \int I^2\, ds$ and $U = \int E^2\, dv \sim \frac{1}{2} CV^2$ or $\frac{1}{2} L \int I^2\, ds$, hence $Q=L / R$

LINAC QWR: f=150MHz, $\omega \sim 10^9$

E~3 MV/m, U (stored energy) ~0.5 Joules

Q(Cu, 300K)~10^4; P~50 kWatts, Q(Pb, 4.2K) ~10^8; P~5 Watts

The LINAC QWR are operated at liquid helium temperature (~4.2 K)

Superconducting LINAC smaller & efficient

The Objective…

For stable and uninterrupted operation of LINAC an efficient cryogenic system consisting of refrigerator and cryogen distribution system is required.

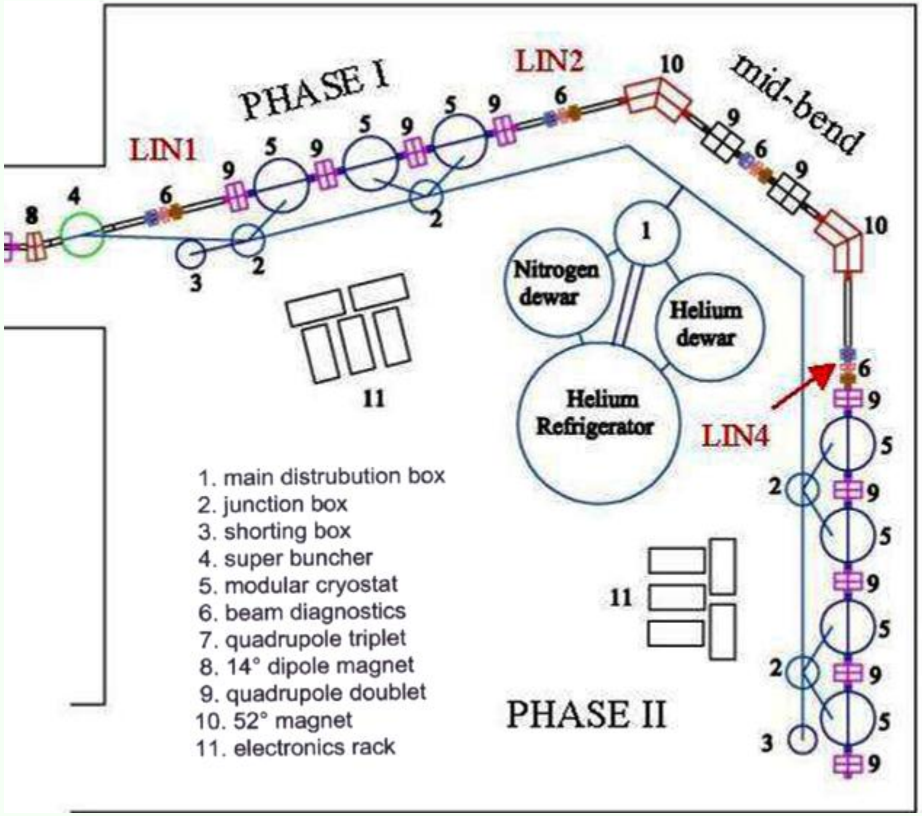
The project objective is to evaluate Cryogenic system using Process simulation.

- *Generate the cool down strategy for Helium system and Optimized Control parameters for automisation with PLC system*
- *Evaluate of the system considering the non-homogeneous transport phenomena, superfluidity and supercritical zone is necessary to validate the performance with respect to design parameters*
- *Suggest the rectification in system for uninterrupted performance*
- *Upgradation to higher capacities to extract high accelerating field*
- *Generate process parameters for operator training*

Introduction of LINAC

Specifications

Heavy ions upto A~80
E / Z~5-12 MeV
Energy gain 14MV/q
Acceptance ~β= 0.1
Module 7 nos
Resonators 28 nos
Bunch width ~200 ps
Beam Intensity 0.1-10 pnA

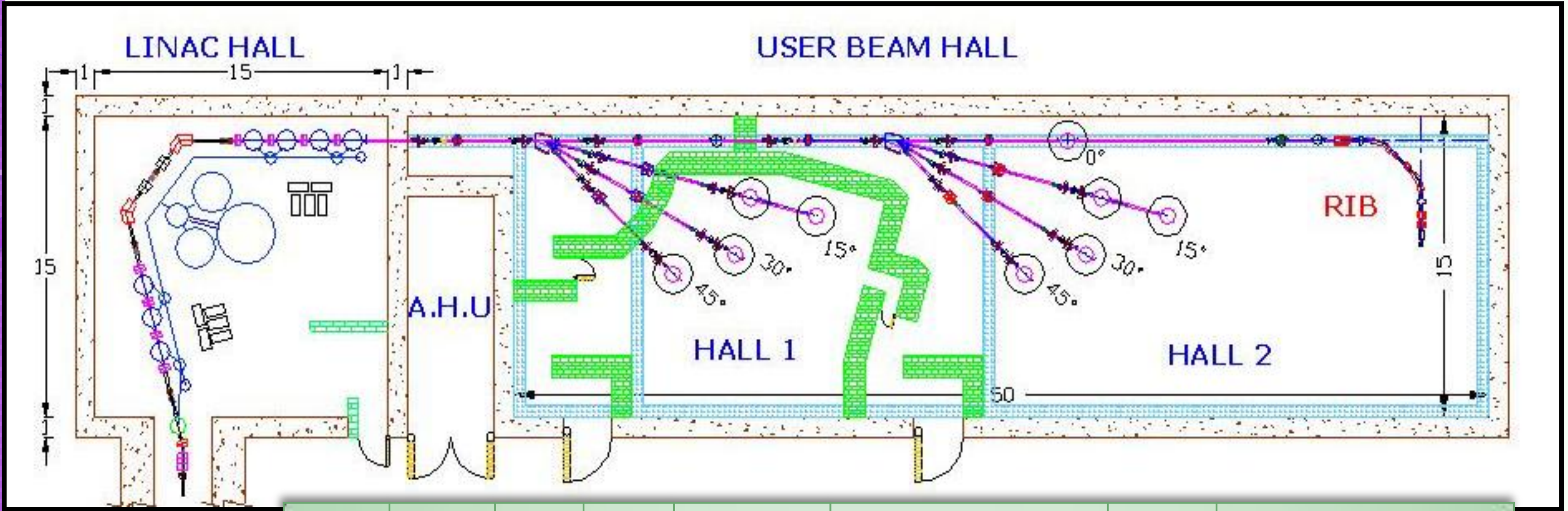


- The Superconducting Linear Accelerator has been indigenously developed to boost the energy of heavy ion beams delivered by the Pelletron accelerator.
- Development of the superconducting LINAC is a major milestone in the accelerator technology in INDIA. It was commissioned in JULY 2007.

View of LINAC

View of LINAC Cryostats and its associated RF and Cryogenic System

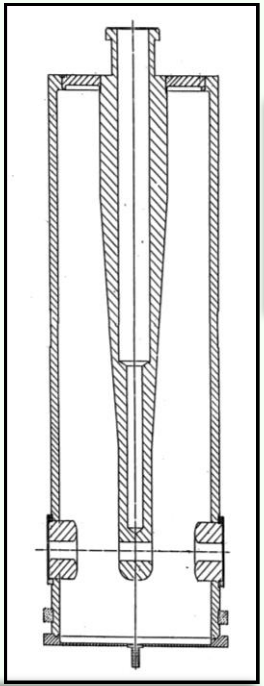
New user beam hall and experimental area



	Z	A	Q_s	β	E_{pell}(MeV)	Q_{s2}	E_{linac} (MeV)
O	8	16	6	0.106	84	8	150
F	9	19	6	0.097	84	8	150
Si	14	28	8	0.091	108	12	210
S	16	32	8	0.085	108	14	230
Cl	17	35	9	0.086	120	15	250

Q_s: Most probable charge state at terminal foil stripper
E_{pell} (β_{pell}): Energy (velocity) at Pelletron exit
Q_{s2}: Most probable charge state after post tandem foil stripper

Superconducting RF cavities for LINAC



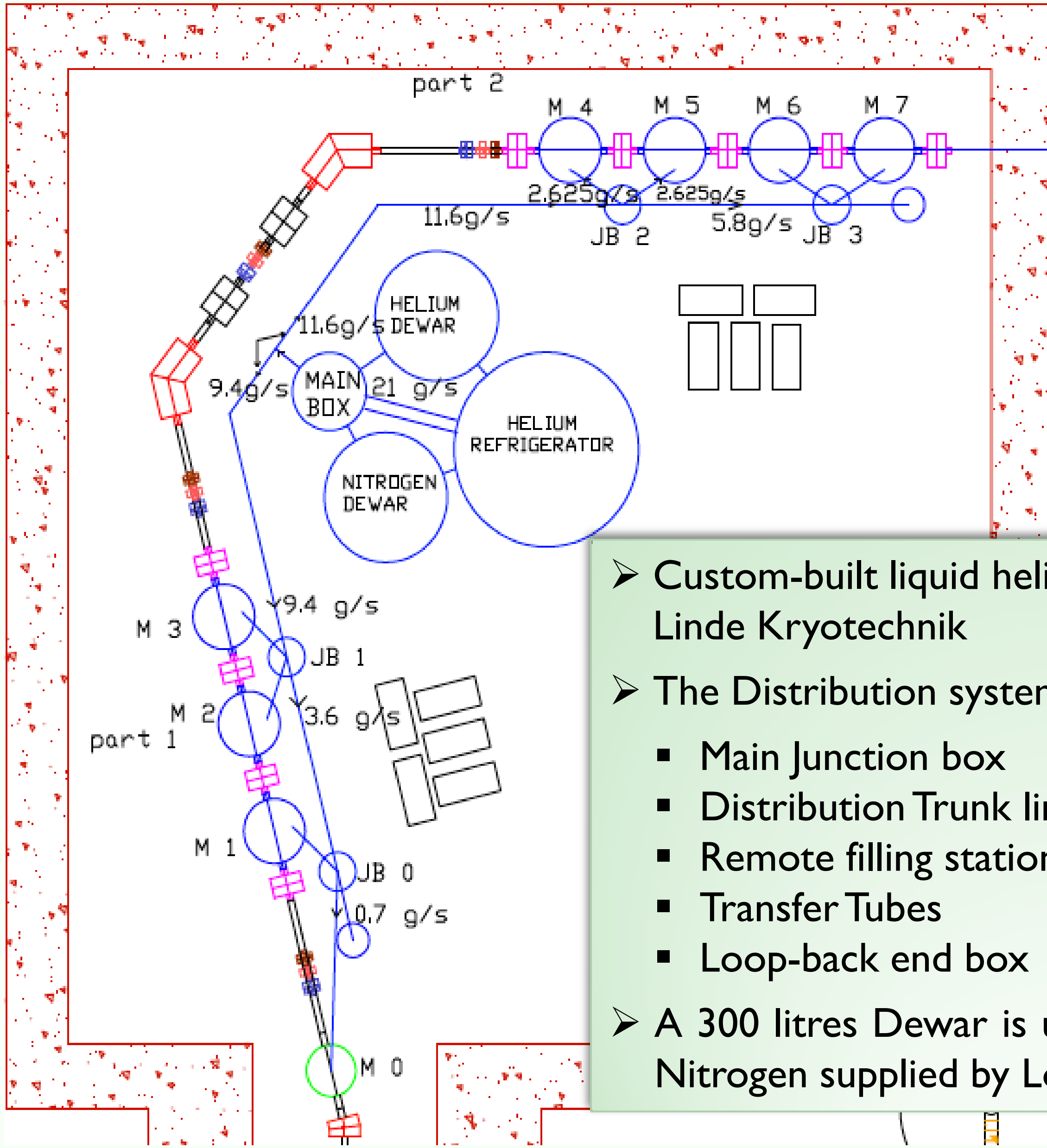
Cryostat receives two phase flow of He and liquid N_2 via tri-axial Transfer Tube

Internal view of Modular Cryostat

Quarter Wave Resonators

Material OFHC	Cu
Superconducting surface	2 μm thick. Pb
Frequency	150 MHz
Cavity Length	64 cm
Cavity Diameter	20 cm
Optimum velocity	β=0.1
Design goal	2.5 to 3 MV/m @ 6 to 9 Watts

Schematic of Cryogenic system for LINAC



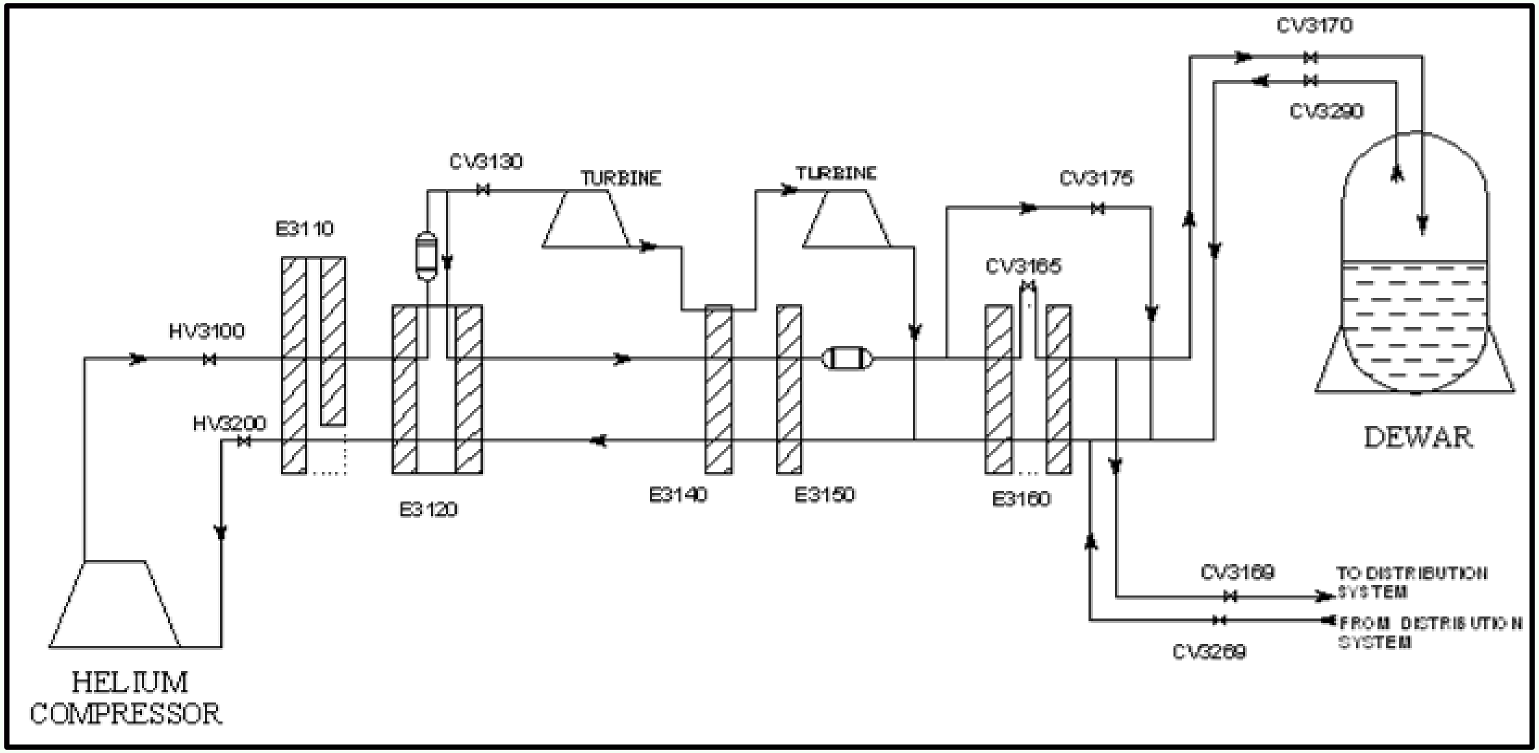
- Custom-built liquid helium refrigerator **TCF 50-S** Linde Kryotechnik
- The Distribution system consists of
  - Main Junction box
  - Distribution Trunk lines
  - Remote filling stations
  - Transfer Tubes
  - Loop-back end box
- A 300 litres Dewar is used as a source of the liquid Nitrogen supplied by Low Temp. Facility of TIFR.

Helium Refrigerator LINDE TCF- 50s

- Based on modified Claude cycle.
- Refrigeration, Liquefaction capacity

Old Configuration: Without LN_2 300 W @ 4.5 K, 50 l/hr

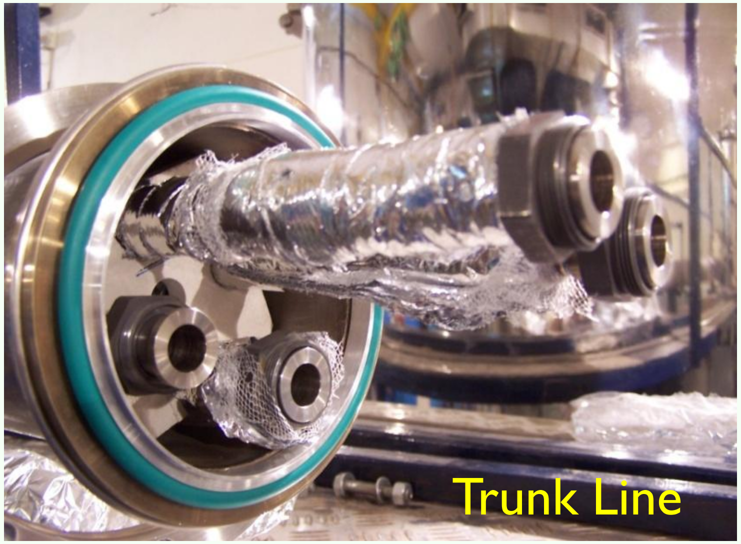
Upgraded Configuration: Without LN_2 410 W @ 4.5 K, 75 l/hr



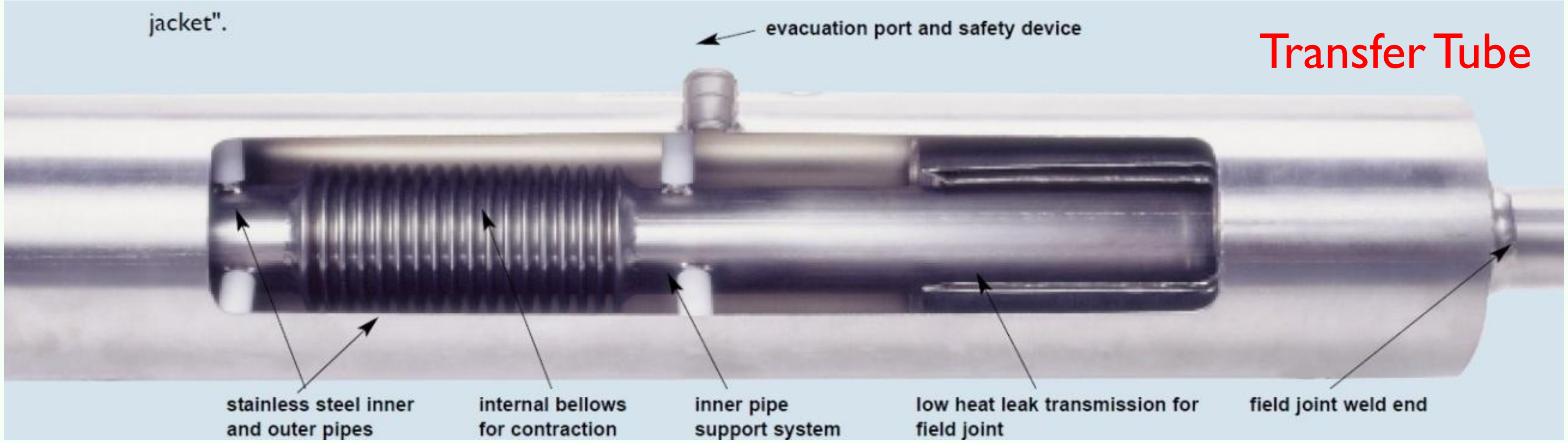
The two-phase helium at 4.5 K produced at the JT stage in the refrigerator is delivered to the cryostats through a Cryogen Distribution system.

View of Helium Refrigerator LINDE TCF- 50s and LN_2 storage

Cryogen distribution system for LINAC



- Vacuum insulated trunk line, 100mm in diameter has four tubes
- Made in separate sections with Kenol fittings supported by Teflon spacer



Individual triaxial transfer tubes of the cryostat serve as a final heat exchanger and a remote JT

Part I: Refrigerator Process simulation

- Steady state simulation
  - For rating and design of the refrigerator.
  - The upgradation of refrigerator to higher capacity
  - Generate process parameters for operator training
  - Parametric study of major components such as Turbine, Heat Exchanger, Valve and Dewar
  - Loss in refrigeration capacity due to off-design operation
- Dynamic simulation
  - Cool-down behavior of the plant in refrigeration mode
  - Operation of the plant at elevated discharge pressure.
  - Heat load fluctuations, pulsed load and disturbances to plant.
  - Control parameter values for the PLC's

Part 11: Evaluation of Cryogenic distribution system

Evaluation of the cryogenic system considering the non-homogeneous transport properties and Heat inleak.

A. The frictional pressure drop calculation:
   - To estimate the loss in the refrigeration capacity.
   - Calculation of the control parameters like Pressure, and Valve opening to achieve equal flow in each cryostats.

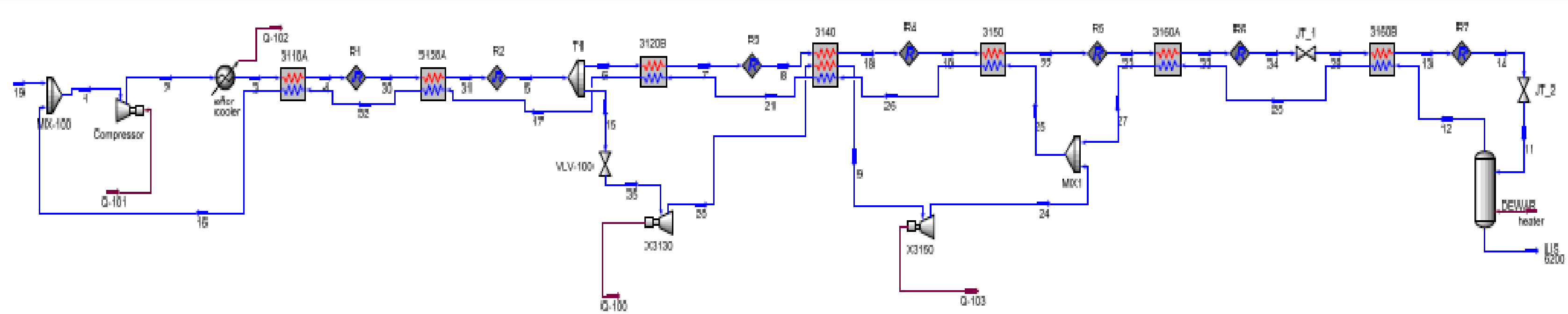
B. Calculation of the standalone heat load on system.
   - The loss in refrigeration due to conduction and radiation heat transfer.

Commercial simulator Aspen HYSYS®

- General purpose modular-sequential process simulator
- Easy Process Flow-sheet Generation with inbuilt blocks for Compressor, Expander, Heat Exchanger (LNG), Phase Separator
- Aspen Muse® software makes it possible to simulate complex plate-fin heat exchangers
- Easy to Switch from Steady State to Dynamics
- Dynamic Assistant
- Availability of Advanced Control Algorithms
- Logical Operations and Spread Sheet
- Easy Customization of Components
- Accuracy of Helium Property Data up to 2.2 K

Part 1: Steady state Simulation of TCF-50S refrigerator using Aspen HYSYS

Process flow diagram (PFD) is generated as shown.



The input conditions for the upgraded plant are as follows

- Temperature of feed=310 K
- HP=12.70 bara and LP=1.2 bara in the modified Claude cycle
- Mass flow rate of the feed stream = 79.5 g/s
- Expander Flow rate ratio=0.63(optimum)
- Two turbines X3130 is TED16-18RQ (adiabatic efficiency=0.81) and X3150 is TGL22 18L/N (adiabatic efficiency=0.70)
- The Pressure ratio for First turbine X3130 is 2.33

The Pressure ratio for second turbine X 3150 is 4.22

Loss in refrigeration capacity due to off-design operation

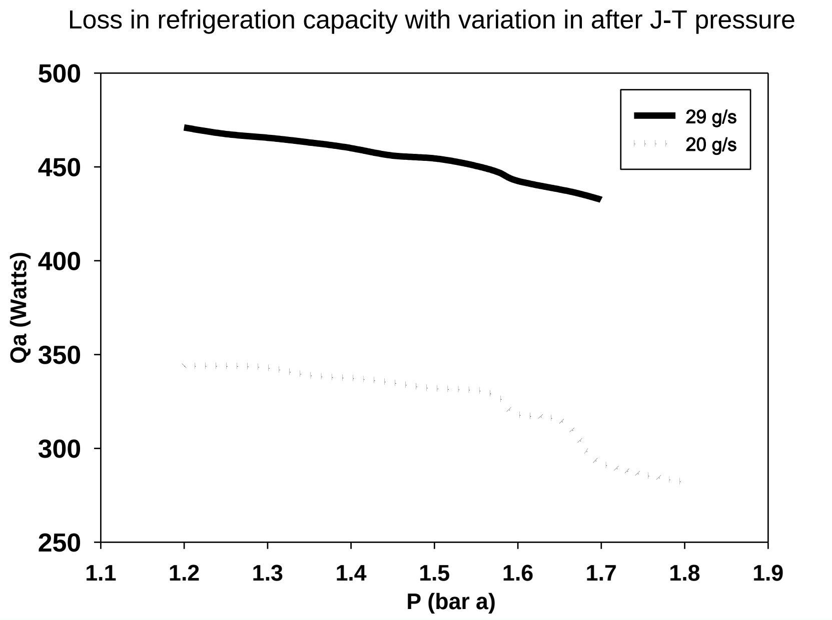
Using Aspen Hysys® the refrigeration capacity is calculated for different J-T outlet pressure

The mass energy balance across Dewar

$$\dot{Q}_a = (\dot{m} - \dot{m}_e)(h_{12} - h_{11})$$

$$(\dot{m} - \dot{m}_e) = 29\, g/s \; and \; 20\, g/s$$

The loss is around ~16 W in steady state for operating at 1.5 bara for 29 g/s and about 10 W for 20 g/s .



Characterization study of components: Turbine

Cryogenic turbines are Inward Flow Radial (IFR) turbines. The performance of turbine is defined by

1. The non dimensional mass flow parameter $\theta = f\left(\frac{u}{c_s}, p_r\right)$

The maximum mass flux, G_{max}, of the stream through turbine is given by choke flow condition. At Mach=1. For Helium as a polytropic gas, the flow becomes sonic if the pressure ratio is above a critical pressure ratio given by $pr_{crit} = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{(\gamma-1)}} = 0.4867$ and $G_{max} = \sqrt{p_0 . \rho_0 . \gamma}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma-1)}}$;Helium γ =5/3

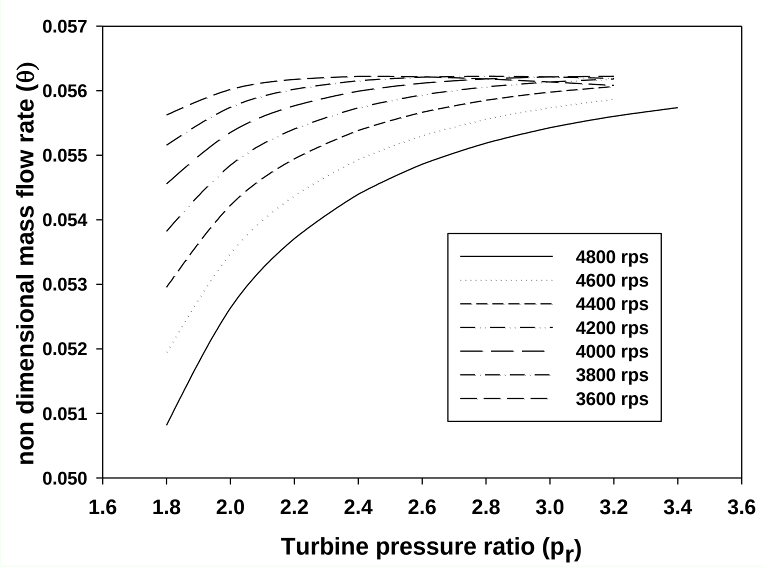
And Non dimensional mass parameter θ is defined as $\dot{m}_d = \theta\,\dot{m}_{max}$ and $\theta = \frac{\dot{m}\sqrt{RT_{01}/\gamma}}{A\,p_{01}}$

Where $\dot{m}_d$ is actual mass flow rate.

Linde TGL 16 X3130 performance map T_{01}=59 K, P_{01}=21 bara and d=16 mm.

behavior of θ is $\theta = a_0 + a_1(u/c_s) + a_2\left(u/c_s\right)^2$

$$u = \pi.\mathrm{N.d} \quad \text{and} \quad c_s = \sqrt{2(\Delta h)_s}$$



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2. Radial turbine efficiency

$$\eta_T = f\left(\frac{u}{c_s}, p_r\right)$$

Turbine stages exhaust in a closed space hence the kinetic energy is of the outgoing jet is lost because it is not used after the turbine Therefore total to static efficiency is

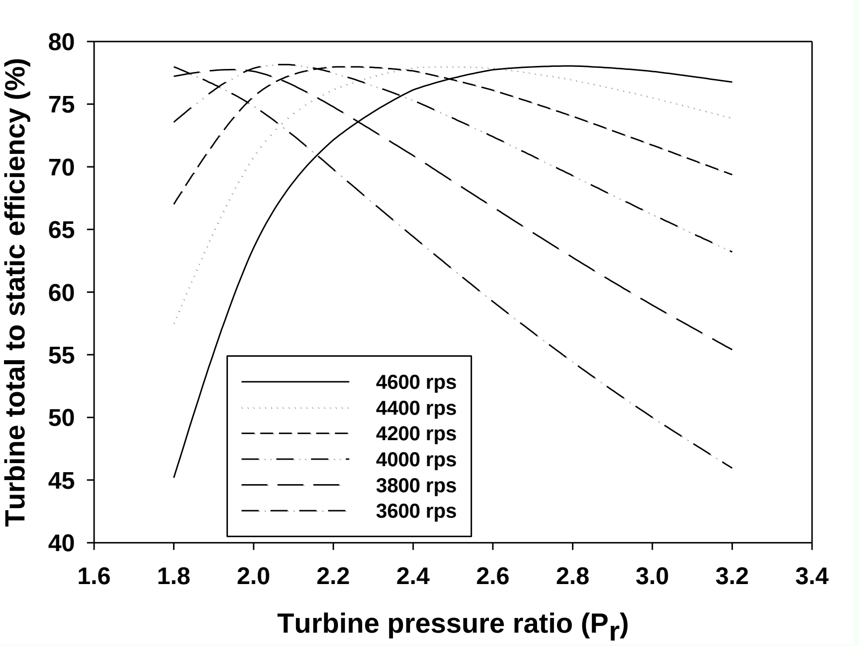
$$\eta_s = \frac{h_{01} - h_{02}}{h_{01} - h_{2s}} = \frac{(T_{01} - T_{02})}{(T_{01} - T_{2s})}$$

Turbine characteristics curve for small capacity expander (θ < 0.055) are retrieved from empirical performance curves.

The Linde TGL 16 X3130
N=4600 rps, d=16 mm,
The entrance velocity ratio
u/c_s= 0.625; optimal is 0.7
Mach number M=0.863
From performance curves
@ Pr=2.4 ; η_s=76%, θ= 0.055

$$\dot{m}_{max} = G_{max}.A = 884\,g/s$$

Hence $\dot{m}_d = \theta\,\dot{m}_{max} = 47.5 g/s$



Heat Exchanger

The rating of Heat exchanger determines the following performance parameters of Heat Exchanger.

Overall Heat Transfer Coefficient: UA
$$\frac{1}{UA}=\frac{1}{(\eta_0 hA)_c}+\frac{R'_{f,c}}{(\eta_0 A)_c}+R_w+\frac{R'_{f,h}}{(\eta_0 A)_h}+\frac{1}{(\eta_0 hA)_h}$$

The Colburn factor j_h ;
$$J_H=\frac{h_c\,\mathrm{Pr}^{2/3}}{c_p\left(\dot{m}/A_{ff}\right)}$$

The total heat transfer is $q=UA\Delta T_{\mathrm{lm}}$; ΔT_{lm} is LMTD

Density, viscosity, specific heat, etc., in the duct were variable, and calculated using HEPAK® at mean fluid temperature. The fin material was assigned a variable thermal conductivity and specific heat. Aspen MUSE was used to simulate the existing HEAT exchanger. The data is

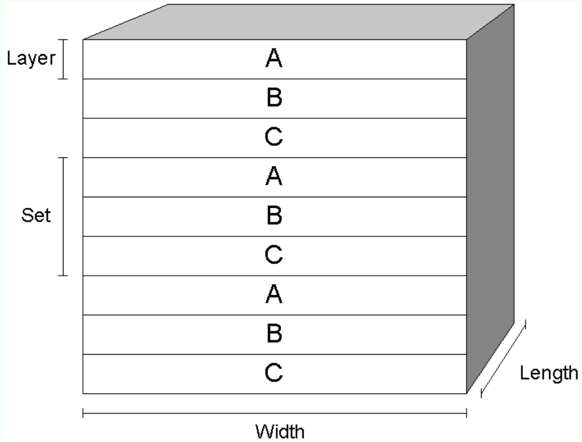
Main Fins Geometry:

Height 8.90 mm, Frequency 787 fins/m, thickness 0.203 mm Fin type: serrated fins

Heat exchanger	3110A	3120A	3120B	3140, Three stream	3150	3160A	3160B
Mass flow rate Hot	79 g/s	79 g/s	29 g/s	29 g/s 50 g/s	29 g/s	29 g/s	29 g/s
Mass flow rate cold	79 g/s	79 g/s	79 g/s	79 g/s	79 g/s	29 g/s	29 g/s
HX dimension (LxHxW) (mm)	966x504x449	1136x306x271	364x118x122	1061x365x318	1228x148x133	1140x207x183	798x128x132
Total surface area (m^2)	211.03	107.55	4.54	123.10	26.80	49.44	13.99
UA (kW/K)	12.252	9.339	0.753	2.39	3.058	2.63	1.218
Heat Duty (kW)	65	46.5	1.5	7.9	1.0	0.7	0.3

Heat Exchanger continue…

The dynamic LNG model is a rating model, which means the outlet streams are determined by the physical layout of the exchanger. The ordering of streams inside layers in each zone is an important consideration.



LNG exchanger block

For two stream heat exchanger the scheme is **C-A-C**

For three stream heat exchanger the scheme is **C-B-C-C-A-C-C-B-C**

Hysys scales the U values based on the flow condition equation. The Dynamic UA

$$UA_{dynamic} = F \times UA_{steadystate}$$

$$F = (mass\ flowrate / reference\ flowrate)^{0.8}$$

Hysys uses weighted method, the heating curves are broken into intervals, which then exchange energy individually. With its immediate layers.

An LMTD and UA are calculated for each interval Which helps in nullifying the errors due to variable heat capacity

Heat exchanger	3110A	3120A	3120B	3140, Three stream	3150	3160A	3160B
No. of Layer Hot	17	10	4	Stream1: 5 Stream2: 8	5	7	4
No. of Layers cold	33	20	7	23	9	13	8
Repeating set	17	10	4		5	7	4

Valve

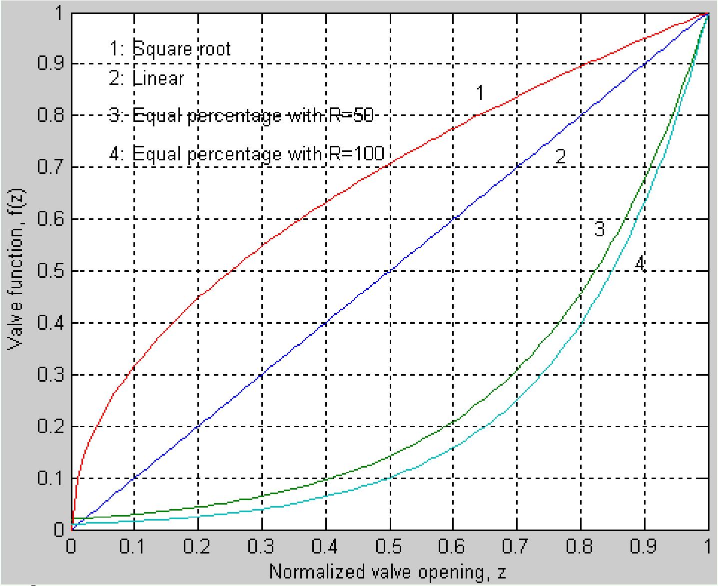
The general valve equation

$$q = C_v f(z)\sqrt{\Delta p / G}$$

Linear: $f(z) = Z$

Quick Opening: $f(z) = \sqrt{z}$

Equal Percentage: $f(z) = R^{1-z}$, R *is rangeable*

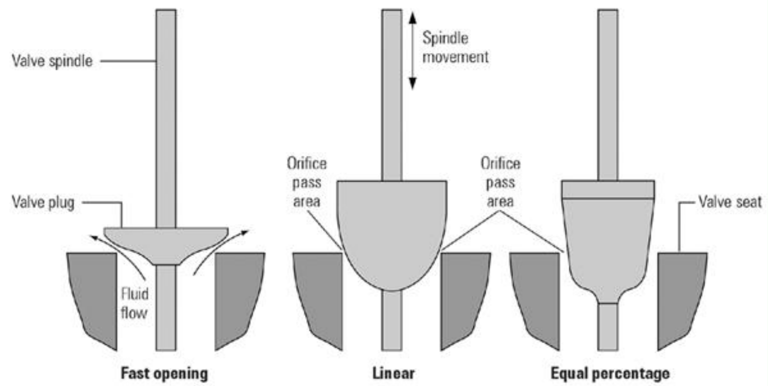


Hysys allows to select type of valve depend upon the requirements

Types of plugs and valve seats which are changed in upgraded plant

In our calculation we are using the Simple Resistance Equation to size the valve

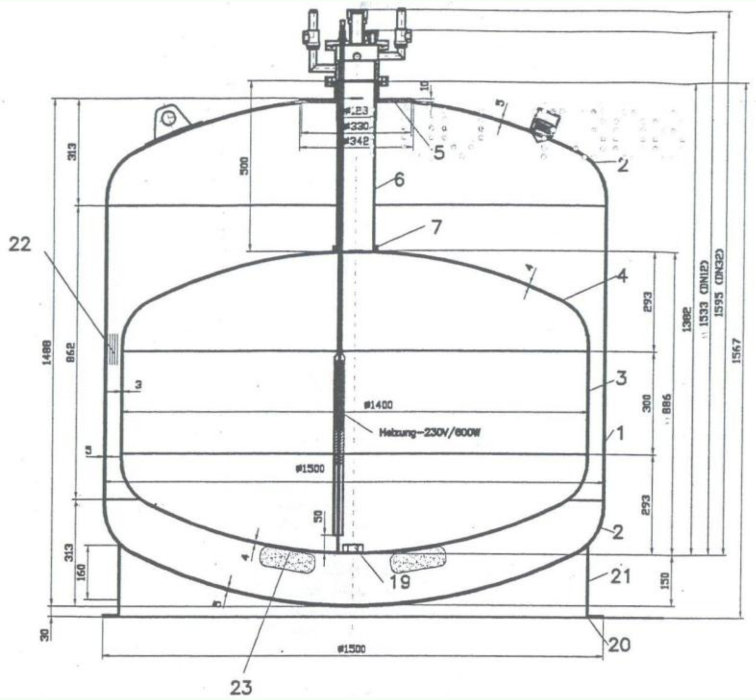
$$\dot{m} = \sqrt{density \times valve\,opening \times \Delta p}$$



Where k value is specified to get the desired pressure and Flowrate.
Care needs to take that the flow should not be choked.
Auto sizing in Hysys helps to size the valve.

Dewar

Aspen Hysys® Phase separator module can be used to accurately model and simulate a Helium Dewar in a process environment.



$$\dot{Q}_{total} = \dot{Q}_{rad} + \dot{Q}_{cond} + \dot{Q}_{g} = 2.41W + 3.20W + 0.277W = 5.887W \sim 6W$$

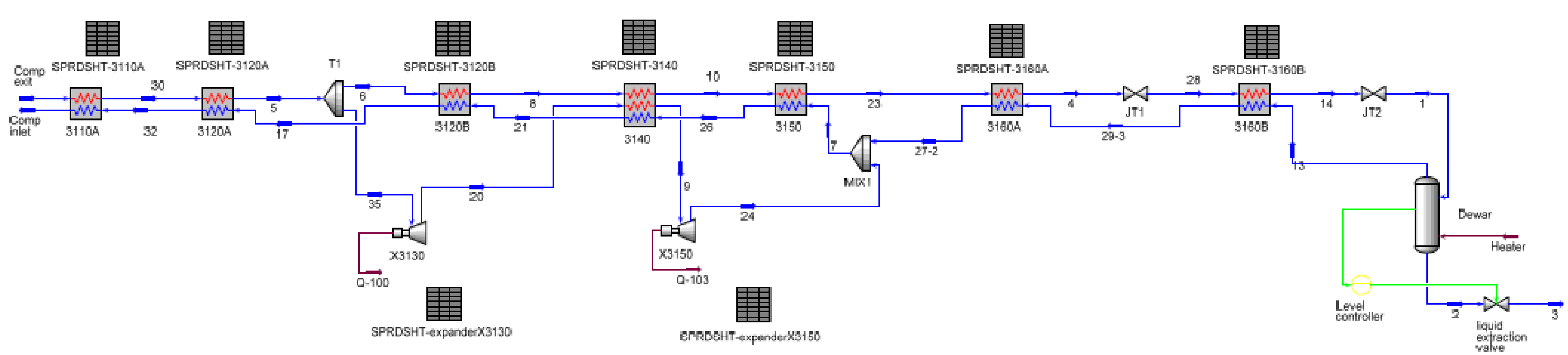
The vessel is equipped with a Helium level sensor and electrical heater(230V/600 W) Heater is used to balance the cold box. The refrigeration capacity is measured using this heater.

Part II: Dynamic Simulation of TCF-50S refrigerator

The cool down of Dewar and the refrigeration capacity of the plant in dynamics is simulated. Following Assumption are made

- The mass flow to the plant is constant and at constant pressure since buffer is used in plant. Compressor unit is not included in simulation
- The turbine adiabatic efficiency and outlet pressure is fixed. This can be done rigorously by using characteristic curves and speed of the turbine.
- The standing heat load of Dewar is considered as ~6 W.
- For refrigeration calculations built-in heater (i.e. direct Q) to Dewar is used.
- pipes connecting all the components are not considered for any hold up calculations

The PFD is generated as follows:



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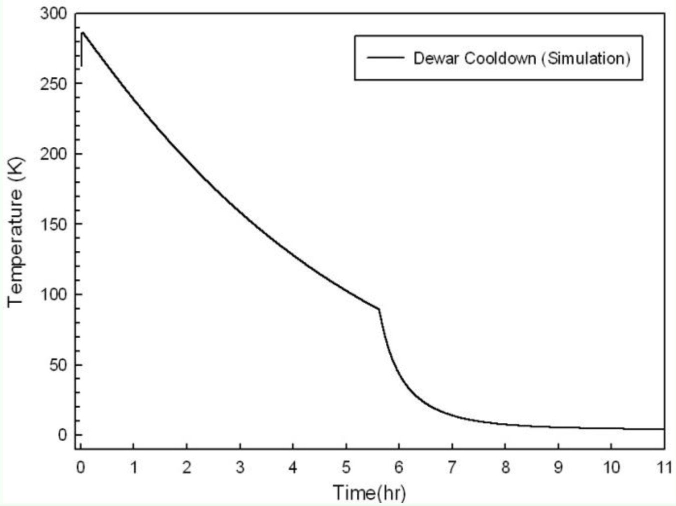
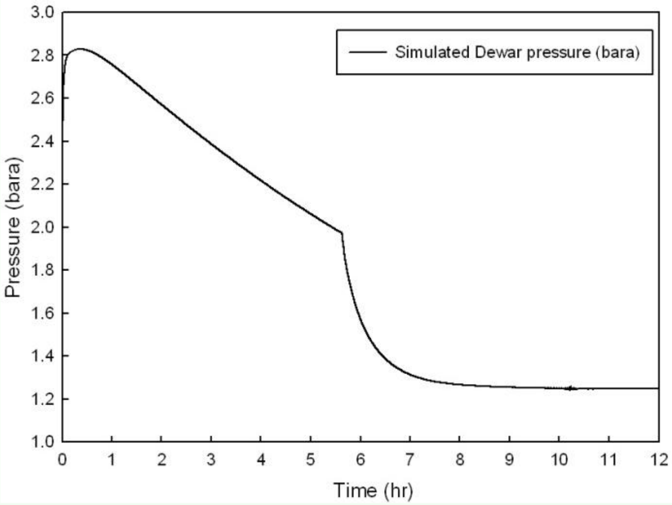
- The simulation is performed on the Intel® core® 2 Quad CPU at 2.66 GHz with 4 GB of RAM.
- Simulated cool down time was 10 hrs. Observed cool down for Dewar is around 12 hrs.
- The time taken for simulation was 30 minutes of computation time. The real time factor was 10.0 times and with acceleration of 2.0. Hence Simulator ran 20 times faster than the real process.
- The integration step size used is 0.5 seconds.
- The total equation summary of model is

Number of equations	276
Number of variables	276
User Spec Equations	9
User Spec Variables	9
Internal Spec Equation	3
Internal spec variables	3

Cooldown of Dewar

The complete cool down of the Helium Dewar from 300 K until 4.8 K is simulated





Liquefaction capacity of TCF 50S

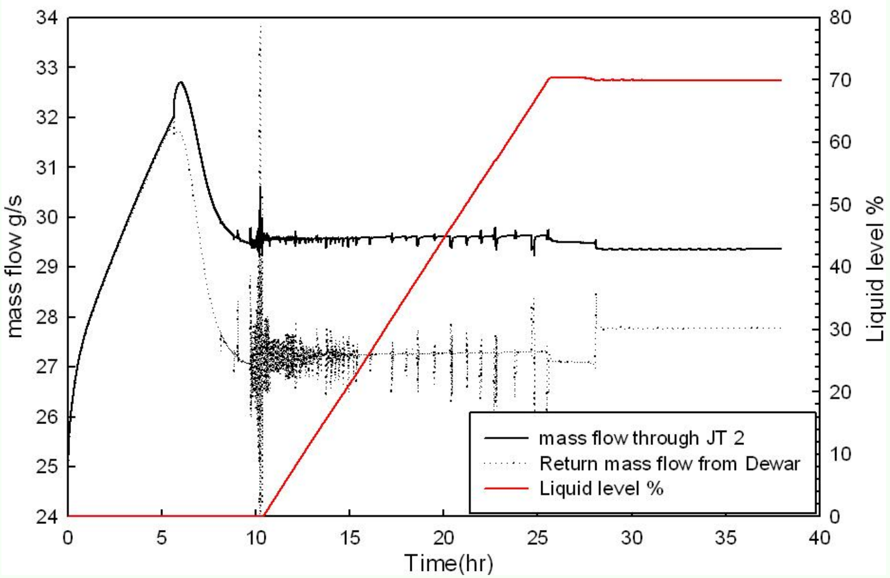
The time taken by vessel to fill:

upto 70 % was 17 hrs

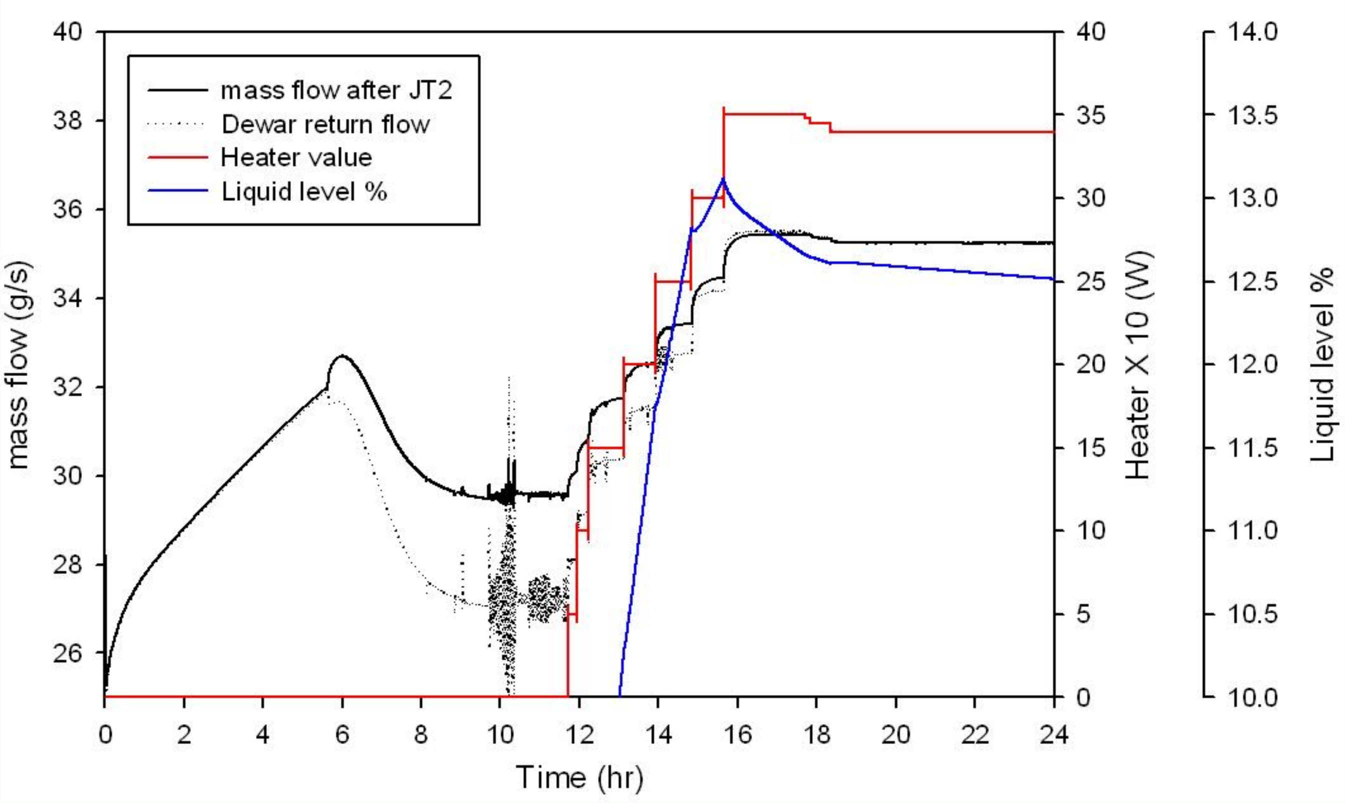
(Volume of Dewar ~1800 L)

The liquefaction rate: ~75 L/h.

Observed capacity is 80 L/h



Refrigeration capacity of TCF 50S



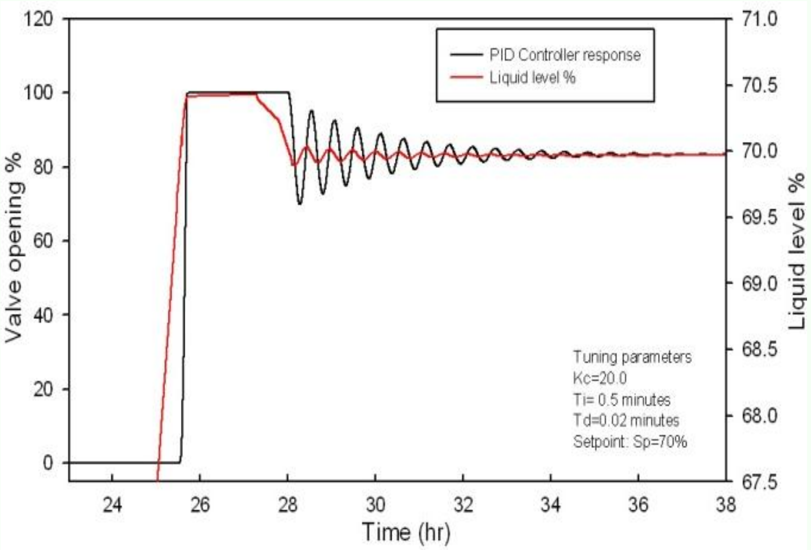
The heater balances at ~340 W@1.25bara. Hence the refrigeration capacity of plant is 340 W@1.25 bara. (Dynamic Simulation)

Actual observed capacity is ~410 W@1.25bara,

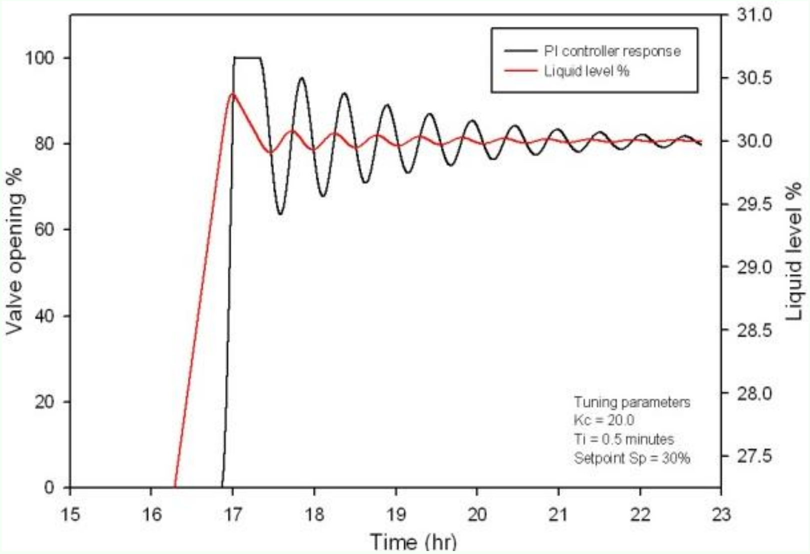
steady state value obtained is 440 W@1.25bara.

The inertia of the heater is well observed while balancing the plant.

The level controller in Dewar with both PID and PI controller types



PID controller response



PI controller response

$$OP(t) = K_c E(t) + \frac{K_c}{T_i} \int E(t) + K_c T_d \frac{dE(t)}{dt}$$

The tuning parameters for controller:

PID controller: gain Kc=20, Integration time Ti=0.5 minutes and Td=0.02 minutes.

PI controller: gain Kc=20, Integration time Ti=0.5 minutes

The controller stability is much better in PID controllers as expected

Pressure Drop calculation:

Pressure drop per unit length is

$$\frac{\Delta P}{\Delta l} = \frac{fG^2}{2\rho D}$$

G is the mass flux

Helium is in better agreement with the homogeneous model hence to predict the two phase flow this model was used over the separated flow model of Lockhart and Martinelli (1949)

mean density is related to fluid quality "x" by

$$\frac{1}{\rho_m} = \frac{(1-x)}{\rho_L} + \frac{x}{\rho_G}$$

The viscosity used in the Reynolds number is a mean viscosity, defined by McAdams (1942)

$$\frac{1}{\mu_m} = \frac{(1-x)}{\mu_L} + \frac{x}{\mu_G}$$

The simultaneous equation obtained are solved, the following output was obtained for

$$V_7 = 1.0$$

$$P_R = 1.2\,bara$$

$$P_s = 1.51\,bara$$

Pressure in mbar							
P_0	P_1	P_2	P_3	P_4	P_5	P_6	P_7
1361.4	1365.1	1365.4	1365.4	1367.2	1367.2	1367.8	1367.8
Valve mass attenuation factor							
V_0	V_1	V_2	V_3	V_4	V_5	V_6	V_7
0.04	0.10	0.10	0.10	0.22	0.22	1.00	1.00

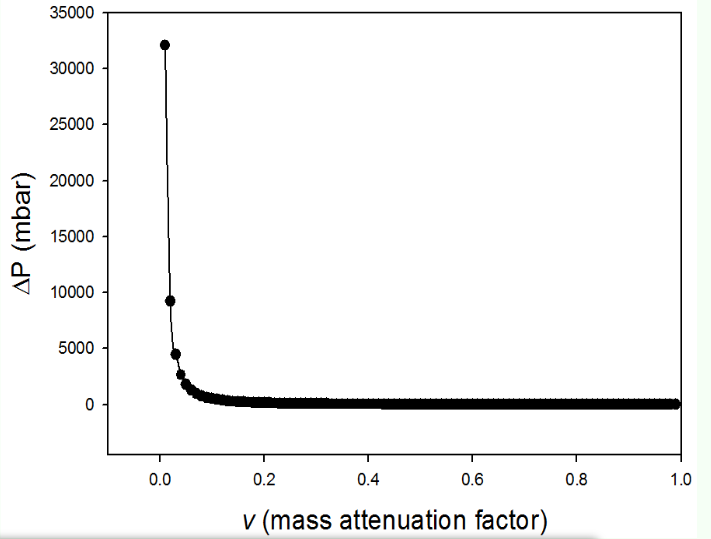
For equal flow in all module cryostats, the valve openings vary over a wide range (0.1 to 1.0). This can be attributed to the asymmetry of distribution between the two parts of the LINAC. Very small pressures changes (~10 mbar) in the supply lines, significantly affect the valve openings.

Continue…

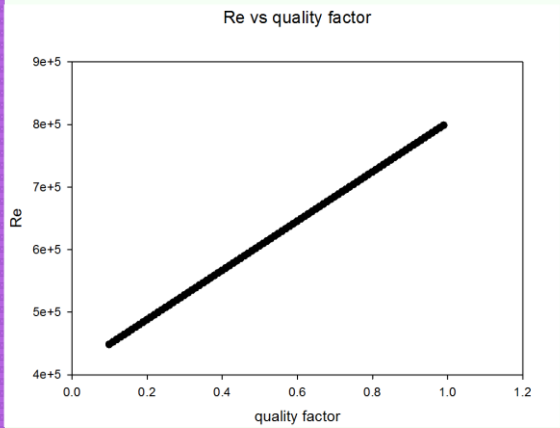
The pressure drop across the valve as a function of valve opening in terms of mass attenuation factor '*v*' is

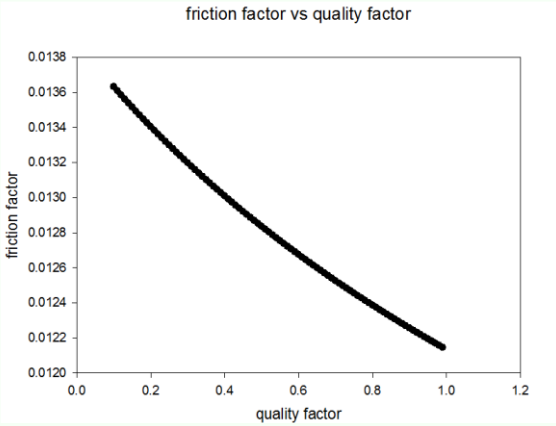
$$\dot{m} = vC_v \Delta P^{\frac{1}{1.8}}$$

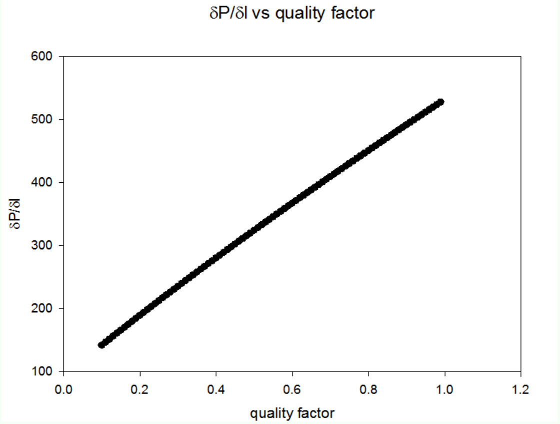
Pressure drop versus mass attenuation factor For $\dot{m} = 29g/s$



Output of the Homogeneous flow model with varying quality factor







The frictional losses in the distribution line are very high if the quality factor *x* of two phase produced after second Joule Thomson is high. Hence TCF50S has double JT system so we can always get low quality factor.

The homogeneous flow model reproduces fairly well, the observed values during the actual operation of the LINAC

Heat load calculation:

In our system there are two major causes of heat load:

- Conduction due to supports and radiation from surroundings at higher temperatures.
- Due to the flow, there is a continuous frictional drag which results in heat dissipation. And given by $\dot{m}\left[U_R - U_S\right] = \frac{dQ}{dt} + \left[P_S \frac{dV_S}{dt} - P_R \frac{dV_R}{dt}\right]; \frac{dQ}{dt} = \dot{m}\left[H_R - H_S\right]$; Supply point is S return point is R

Calculating the upper limit of the heat load by assuming the supply pressure 1.6 bara and return pressure 1.2 bara, we get available refrigeration capacity

Summary of total standalone Heat Load in Cryogenic System

	Estimated Heat Load (W)		Heat Load (W)	Heat Load (W)
	Phase I	Phase II	Old Plant	Upgraded
The mass flow from compressor			**62 g/s**	**79 g/s**
The mass flow through distribution system			**20 g/s**	**29 g/s**
The deviation of operating point from the rated value	----	----	10W	16 W
Frictional drag losses	----	----	32W	55 W
Heat Load in He Dewar	6 W		6 W	6 W
Distribution box, main box and trunk line	16 W	16 W	32W	32 W
Transfer tube and cryostat, 12W each	4X12W=48W	4X12W=48W	96W	96 W
QWR @6W each and Superbuncher @4W	12x6W=72W 1 x 4W =4W	16x6W=96W	172W	172 W
Total (Phase I + II)	306 W		348W	377 W

Upgradation of TCF50S plant

The upgradation of TCF50S was carried out at TIFR site in December 2010.

The compressor which delivers 80g/s was installed Associated ORS system was replaced. The valve seats of the control valves are changed to accommodate the higher mass flow rates. Due to a higher mass flow first expansion turbine X3130 in the cold box was replaced with a larger unit.


Turbine TED 16

internal view of TCF 50S

Conclusions

- The Available refrigeration capacity is 410W hence with the heat load of 377W LINAC operations is possible without the Nitrogen Precooling. The measured refrigeration capacity using built-in heater in DEWAR is ~410 W. We still left with 33 W which could be utilized to run the RF cavities at higher dissipating power of ~7 W each. This helps to get higher accelerating field in the resonators and ultimately the higher energy gain.
- The Dynamic Simulations was done with the coldbox as liquefier/refrigerator mode. The real data will be obtained shortly for comparison. The coldbox is still conditioning and different parameters for controllers and valve settings needed tuning. The data from Dynamic simulation is helpful for generating control strategies and optimizing the system.

Conclusions

- The characteristic curves for turbines must be considered when refrigerator simulation is modeled. This will accurately calculate the efficiency and head with varying speeds of turbine and hence producing better simulation results.
- The controller settings and performance was simulated. The PID controllers are much stable feedback controllers than PI or P-only controllers.
- We upgraded the existing plant with higher mass flow rate (~79 g/s).The existing TCF 50S refrigeration plant is modified to get the higher refrigeration capacity. The new turbine and the control valve seats are changed to accommodate the higher mass flow rate. After upgradation the new calculated capacities are; without LN_2: Refrigeration: 410 W at 4.5 K, Liquefaction: 80 l/hr.

THANK YOU