CREATIV
Industry Energy Efficiency

Workpackage: D4.11: Concept, Calculation and 3D Design of R744 Oil Free Turbo Compressor Unit

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Efficient heating and cooling (SP4)

Content: page

1. Task and Background

Today's refrigeration systems have to use a defined portion of oil in order to provide lubrication of mechanical moving parts in the compressors. This oil is mainly added into the compressor but is transported with the working media into the system, so a not negligible amount is pumped around in the system. An oil free compressor (and with that oil free system) would therefore have following major advantages:

- improved heat transfer in the heat exchangers
- improved system efficiency → energy saving
- reduction of heat exchanger size
- lower system complexity (no oil return)
- increase of temperature range (today oil is limiting)
- reduction of servicing and maintenance effort

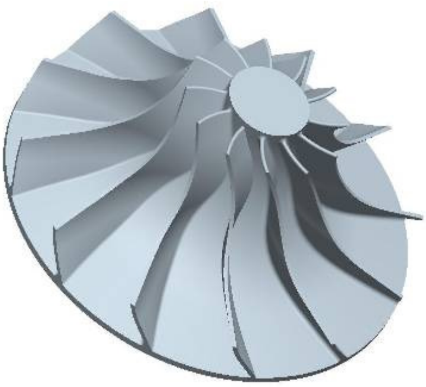
In combination with natural refrigerants an oil free refrigeration system would offer a significant step forward towards cost saving, fuel saving and environmentally friendly refrigeration systems.

In previous studies R744 (carbon dioxide) was found as a very promising solution for such a oil free turbo machine among the natural refrigerants.

Task of this study is a 3D design, calculation and definition of R744 turbo compressor unit under given specifications for a defined application in order to pre-determine required amount of stages, required dimensions and to get a first prediction of possible, reachable compressor efficiencies.

2. Target Conditions and Specifications

Application:	Fishing boat / cooling of fresh good
Media:	R744 (carbon dioxide) oil free
Inlet Conditions:	30bar, 0℃ (5K SH)
Outlet Conditions:	55 to 80 bar, best efficiency at 65 bar
Mass Flow:	6000 kg/h, +/- 5% in application
Power In:	approx. 75 kW
Speed range:	approx. 50.000 rpm
Efficiency Target:	> 75% (turbo stages only)



3. Calculation and 3D Design of R744 Turbo Compressor Unit

3.1 Pre – Calculation: Turbo-Quick-Dim Tool

In a first step a simplified excel-calculation tool is created ("Turbo-Quick-Dim"), allowing a first prediction of required stages, a fist idea of outer diameter and speed range of compression stages:

This calculation tool is based on procedure as mentioned in handbook of mechanical engineering: Dubbel-Version18-R73-Capter7.5.

1. System and Overall Inputs				
Inlet pressure	p1	30,0	[bar]	Input
Corresponding evaporating temperature	t_evap	-5,4	[°C]	Calculated
Required overheat	dt	5	[K]	Input
Inlet temperature	t1	-0,4	[°C]	Calculated
Outlet pressure	p2	65	[bar]	Input
Max. allowable speed of rotation	n_max	45000	[1/min]	Input
		750	[1/s]	Calculated
Required mass flow	m_dot	6000	[kg/h]	Input
		1,67	[kg/s]	Calculated
Assumed overall compressor efficiency	eta_overall	75	[%]	Input

Step 1: Definition of running conditions, especially inlet temperature and pressure, outlet pressure, available electrical driving power, available max. motor speed, required refrigerant mass flow.

2. Gas Data Calculations				
Inlet density	roh_1	77,7	[kg/m³]	Calculated
Inlet enthalpy	h_1	-65,4	[kJ/kg]	Calculated
Inlet entropy	s_1	-0,835	[kJ/kg]	Calculated
Inlet speed of sound	a_1	221	[m/s]	Calculated
Inlet volume flow	V_dot_1	0,021	[m³/s]	Calculated
Isentropic outlet temperature	t_2_is	58	[°C]	Calculated
Isentropic outlet enthalpy	h_2_is	-32,9	[kJ/kg]	Calculated
Isentropic delta h	dh_is	32,4	[kJ/kg]	Calculated
Real delta h - eta_overall considered	dh_real	43,2	[kJ/kg]	Calculated
Outlet enthalpy	h_2	-22,1	[kJ/kg]	Calculated
Outlet temperature	t_2	65	[°C]	Calculated
Outlet density	roh_2	135	[kg/m³]	Calculated
Outlet speed of sound	a_2	250	[m/s]	Calculated

Step 2: Calculation of real gas values out of given in and outlet conditions. For the calculation of the real outlet temperature t_2 an assumed, reachable compressor efficiency is used. The real gas calculation is based on R744 data base of Prof. Wagner, Univ. of Essen, Germany.

3. Determination of stages

Speed of sound approach	x	70,0	[%]	Input value
Circumpherencial speed at d2	u2_max	160	[m/s]	Calculated
Target value of psi (per stage)	psi_target	1,1	[-]	Input value
Volume flow coefficient	phi	0,023	[-]	Calculated
Blade outer diameter	d2	97,6	[mm]	Calculated
Specific polytropic work per stage	y_p	22,0	[kJ/kg]	Calculated
Amount of stages required	i	2,5	[-]	Calculated

Step 3: Determination of stages required. Important input value is the speed of sound approach, which determines the approach of the outer wheel velocity towards the speed of sound of the gas. Literature is showing 40 to 90%, 90% as maximum. The lower the speed of sound approach, the more stages are required.
More stages does also mean more possible leakages and complexity of the real turbo machine, so our target here was to minimise the required amount of stages.
Basic result: for the given conditions a stage amount of 2 for the R744 turbo is calculated and the blade outer diameter is below 100 mm.

4. Further output values

Compressor power consumption	P_clutch	72,07	[kW]	Calculated
Shaft driving torque	M_shaft	15,29	[Nm]	Calculated
Flow-through coefficient	phi	0,023	[-]	Calculated
Speed coefficient	sigma	0,128	[-]	Calculated
Diameter coefficient	delta	7,0	[-]	Calculated

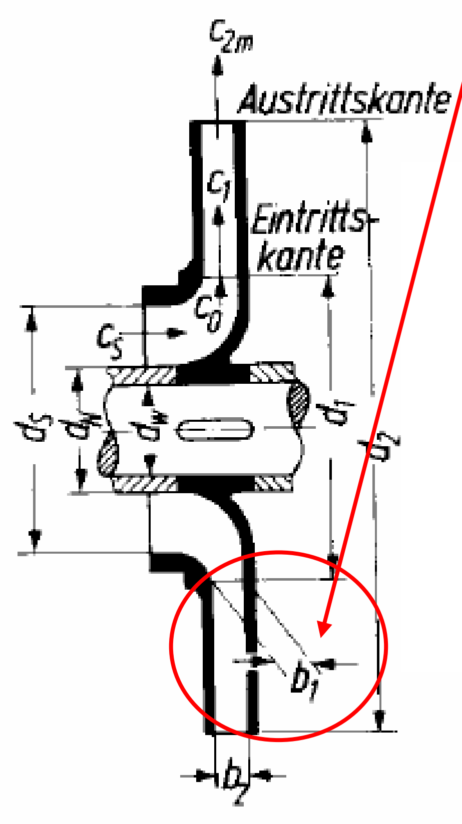
Step 4: Important for layout of the turbo e-motor is the power consumption and driving torque required at the calculated condition. This calculation is based on the estimated overall efficiency.
Basic result: E-drive running at 45.000 rpm requires 15,3 Nm driving torque at this point, resulting in total 72 kW output shaft power.

5. Pressure distribution of stages and geometry aspects

Chosen amount of stages	i_set	2,00	[-]	Rounded value
Pressure ratio per stage	pi_stage	1,47	[-]	Calculated
Medium inlet velocity	c_m	50,0	[m/s]	Input value
Inlet diamter (out of design aspects)	d_1	40,0	[mm]	Input value

A common medium inlet velocity can be found in literature from 30 to 100 m/sec. Inlet diameter is depending on the real compressor design and alignment, shaft and bearing concept.

	p_out [bar]	roh [kg/m³]	V_dot [m³/s]	A_required [mm²]	**b_1 [mm]**	**delta p [bar]**
Stage 1	44,1	106,2	0,01569	523,0	**7,5**	**14,2**
Stage 2	65,0	134,7	0,01237	412,5	**6,4**	**20,8**



Step 5: With a main input of the given stages (rounded value to full number), the expected pressure distribution is calculated moreover the width b2 (see sketch above) is determined. Main result: Pressure distribution 30 – 44 – 65 bar. Stage 1 width is 7,5 mm, Stage 2 width is 6,4 mm.

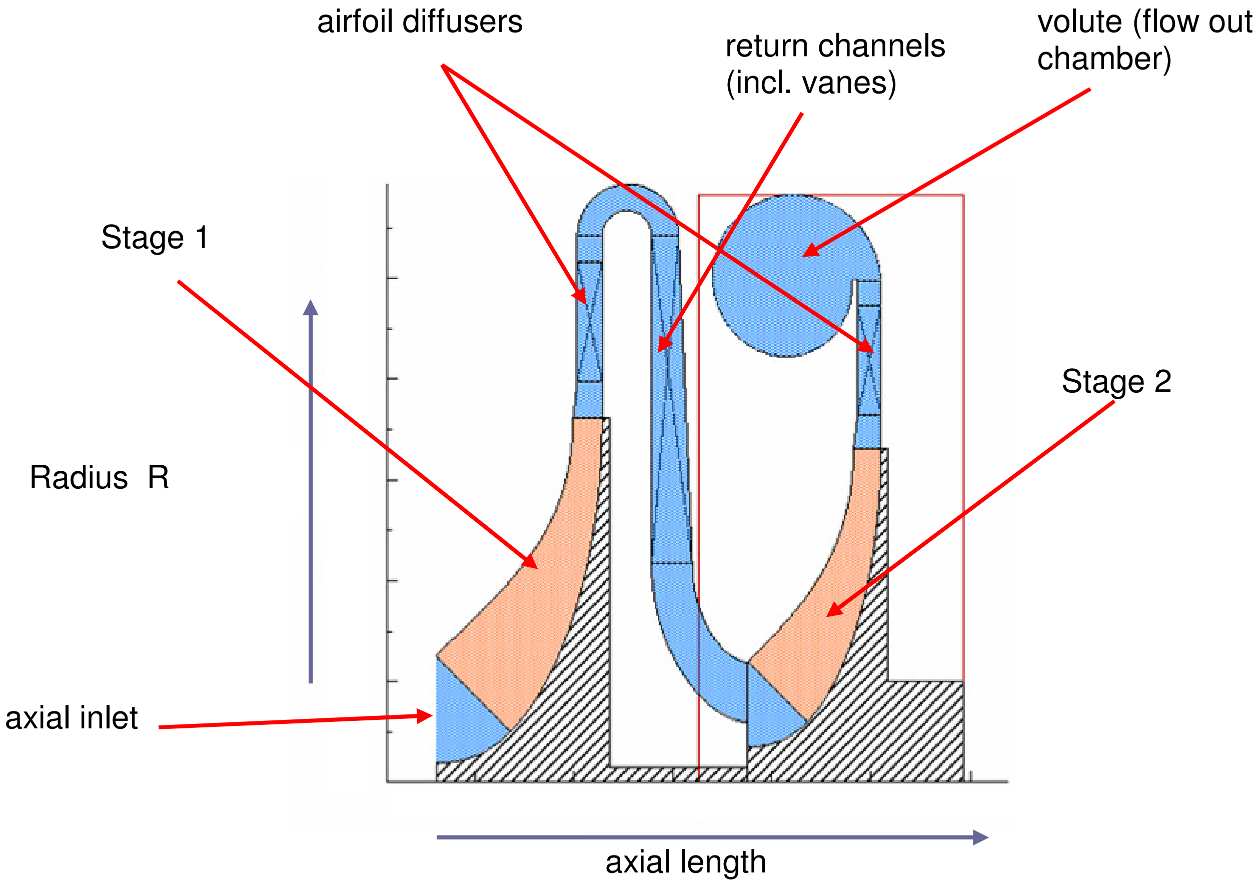
Conclusion of Quick-Dim calculation: a very compact R744 turbo compressor with 100 mm outer diameter, 2 stages and narrow width of 7,5 and 6,4 mm is pre calculated.

3.2 One–Dimensional Turbo Simulation Tool

Within in this study a common simulation tool for turbo compressors was chosen for the 1D study. A 1D study gives a very good first result on the real compressor dimensions required and efficiency prediction.
These simulation tools allow the design, changing of parameters, input of sealing concepts and prediction of expected efficiencies. The targeted main outputs are

- number of stages
- main dimensions
- impeller type open or closed
- amount of blades
- diffuser concepts
- efficiency prediction incl. leakage models

3.3 Naming and declaration of compressor components



3.4 Number of Stages / Speed Range

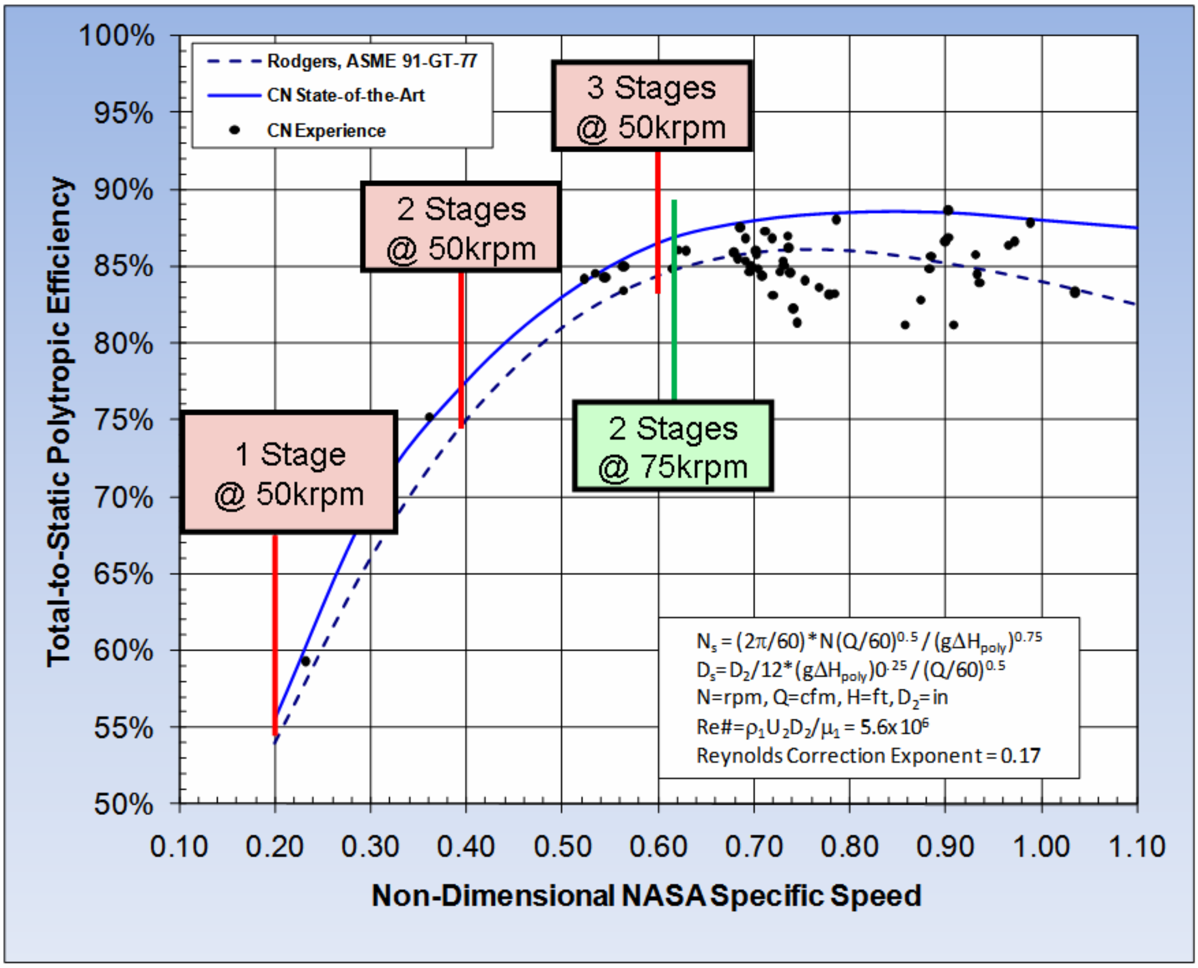


Chart above is showing variation on amount of stages and also a speed variation. Basic results are

- a one stage solution can not meet efficiency target
- a three stage solution has theoretically an advantage towards 2-stage, but in real world rotor dynamic effects and increased leakage surfaces must be considered
- therefore a 2-stage solution is best compromise for given conditions in order to meet the 75% efficiency target, whereby
- the increase of speed (and with that increase of power in and mass flow) would offer further efficiency potential, see green box 2-stages at 75krpm

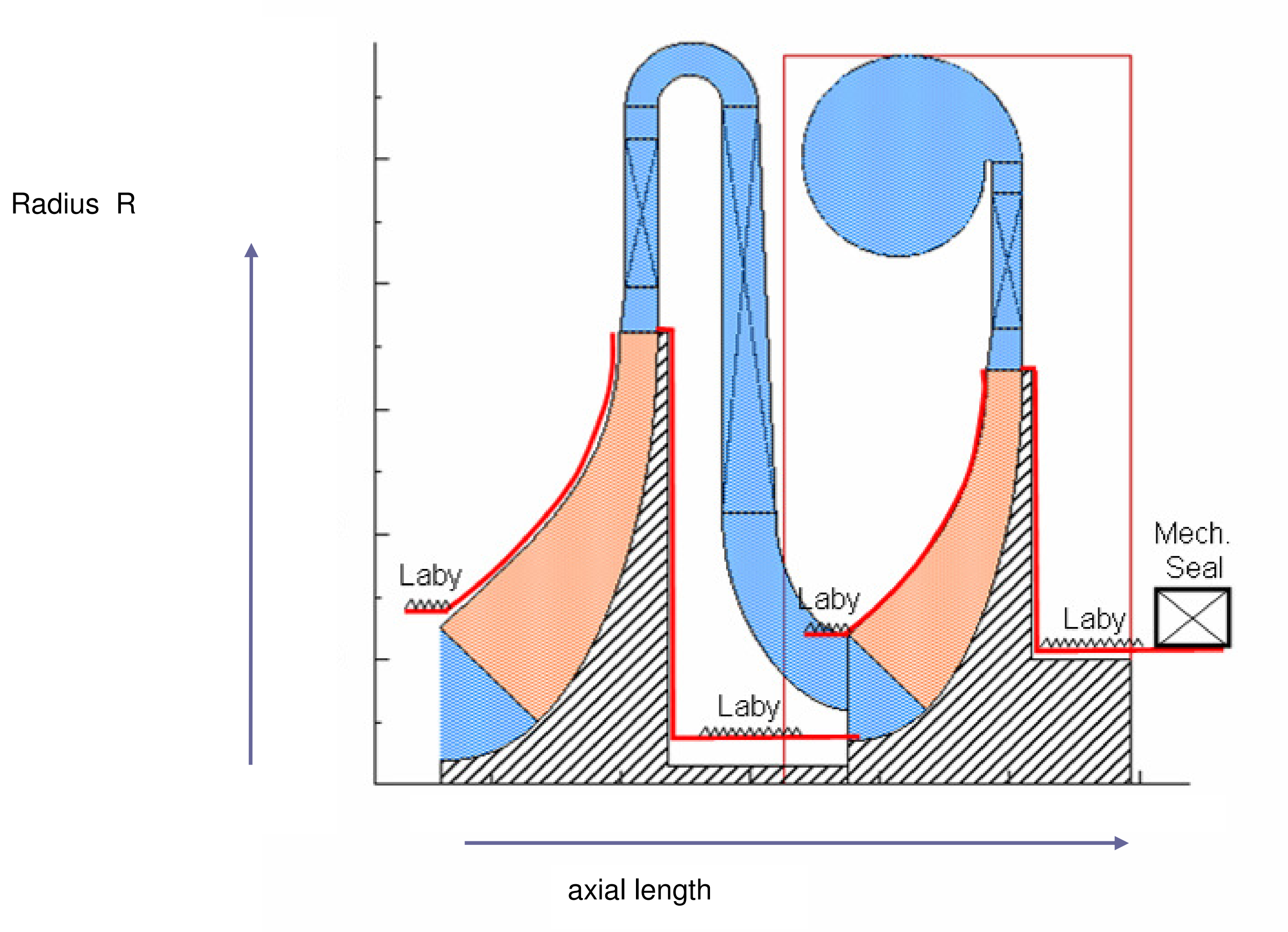
3.5 First 1-D-Simulation Results

Parameter	Units	Stage 1	Stage 2	Overall
Mass Flow Rate	Kg/s	1.667	1.66622	1.667
Shaft Speed	rpm	45000	45000	45000
Stage Power	kW	38.9	33.3	72.2
Stage PR_{TT}	-	1.55	1.40	2.17
Stage PR_{TS}	-	1.51	1.39	2.10
Stage η_{TT}	-	76.30%	76.79%	76.53%
Stage η_{TS}	-	71.56%	74.61%	72.97%
Rotor η_{TT}	-	88.46%	88.18%	88.33%
Exit Pressure	bar			65.11

efficiency goal of > 75% is achievable

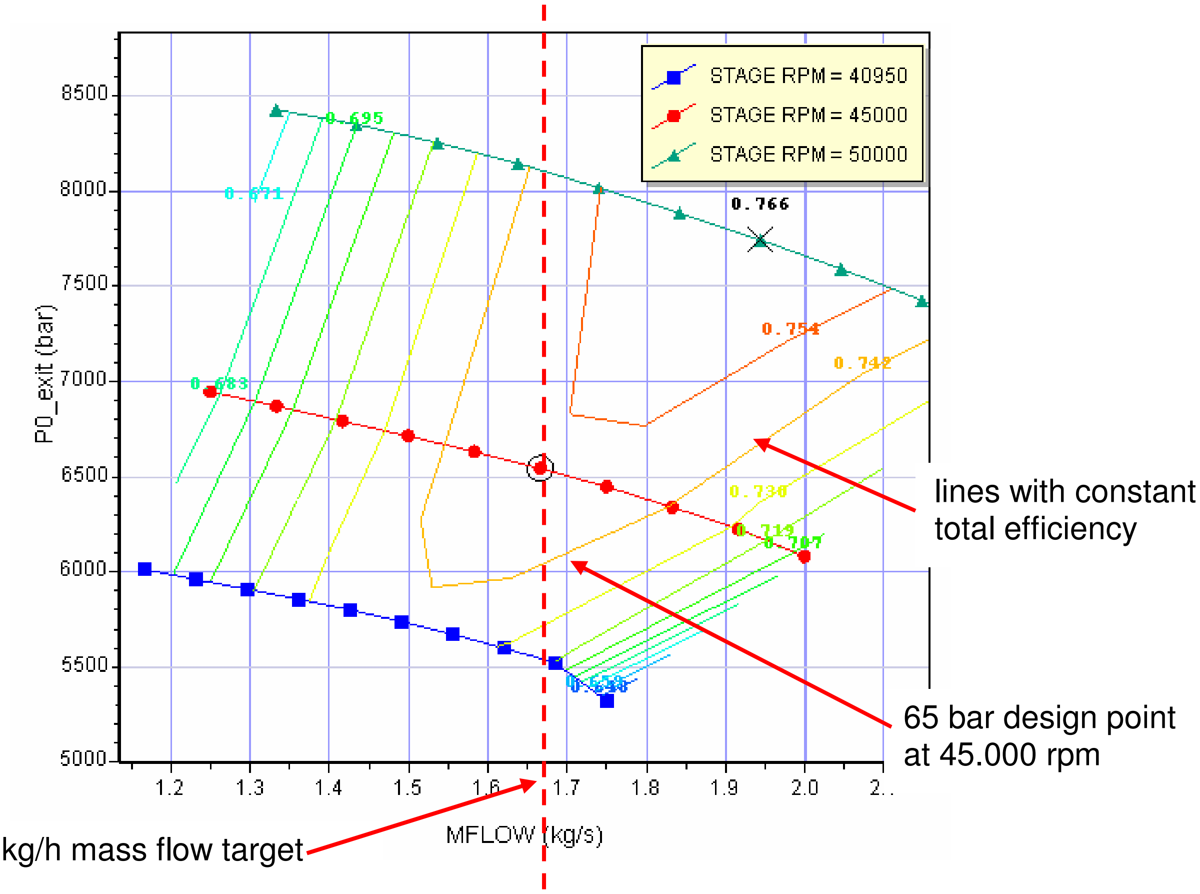
The first 1-D simulation results confirm the results of the Quick-Dim simulation tool with two stages. The outer blade diameter is even smaller than calculated in quick. The goal of achieving > 75% is achievable.

First 1-D simulation results dimensions:

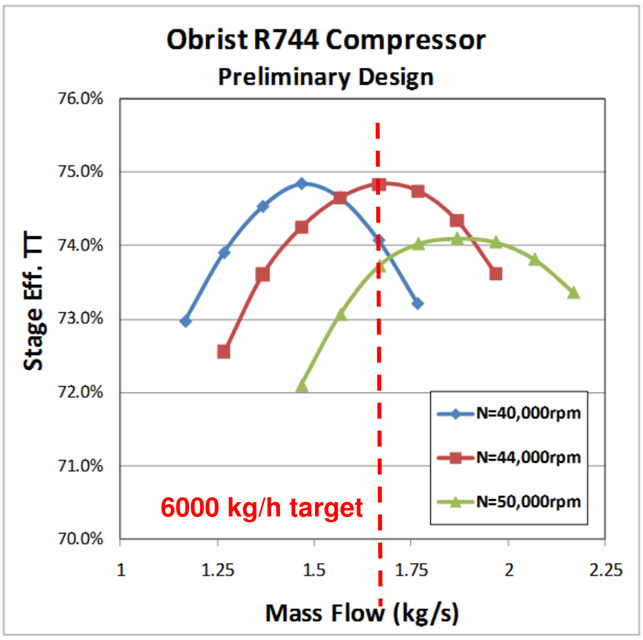


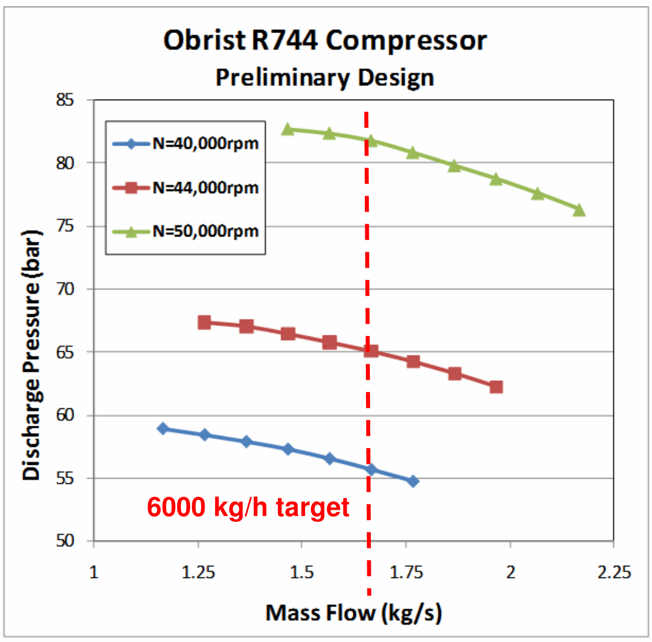
A 75kW R744 turbo stage unit with two stages can be designed very compact. The blade outer diameter is below 100 mm the total axial length is below 150 mm.

3.6 R744 Turbo Compressor Operation Map

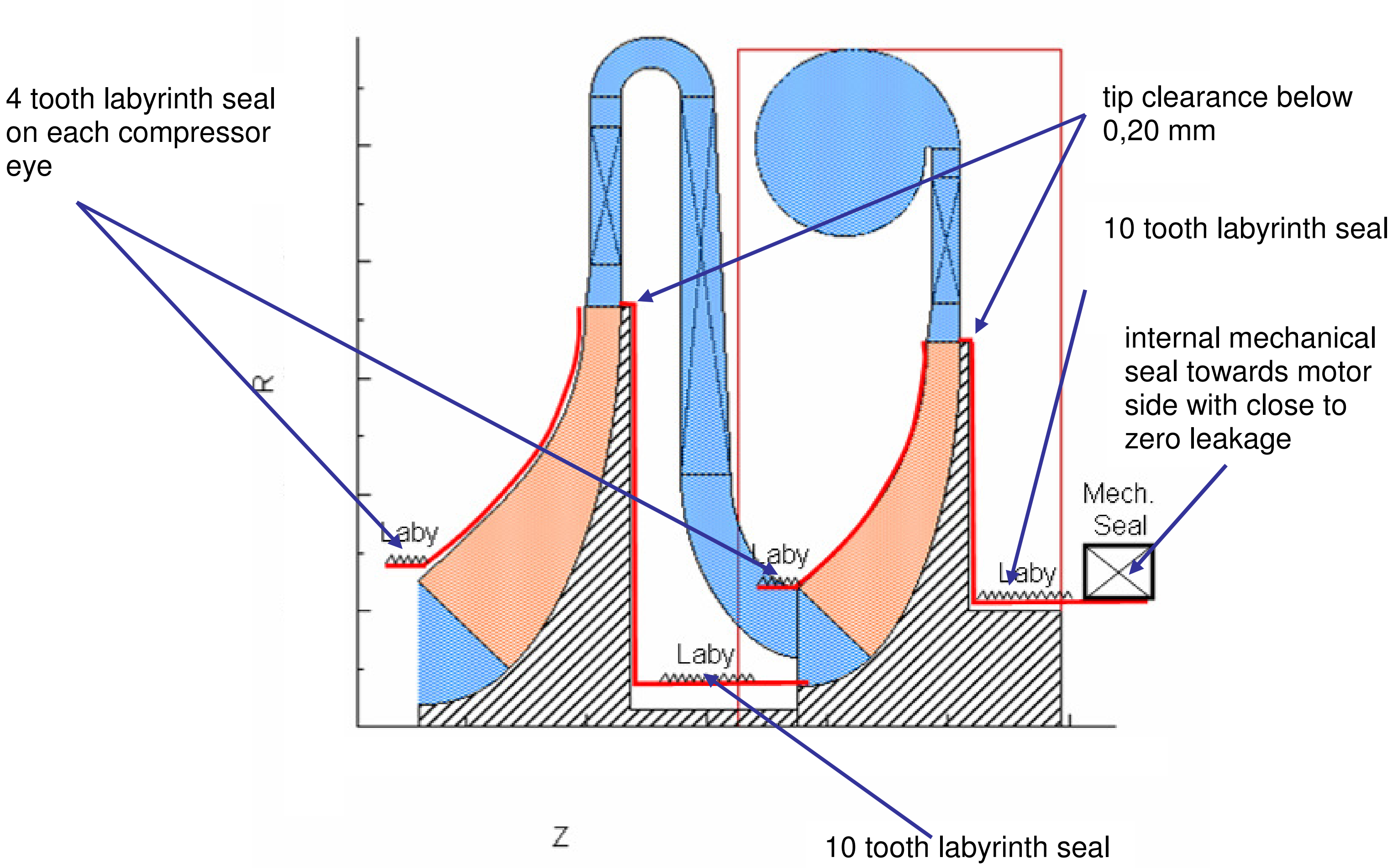


Results efficiency map: The target output pressure p2 from 55 to 80 bar is reached with a speed variation from 40 to 50 krpm. Below 55 bar outlet pressure the efficiency drops significantly.



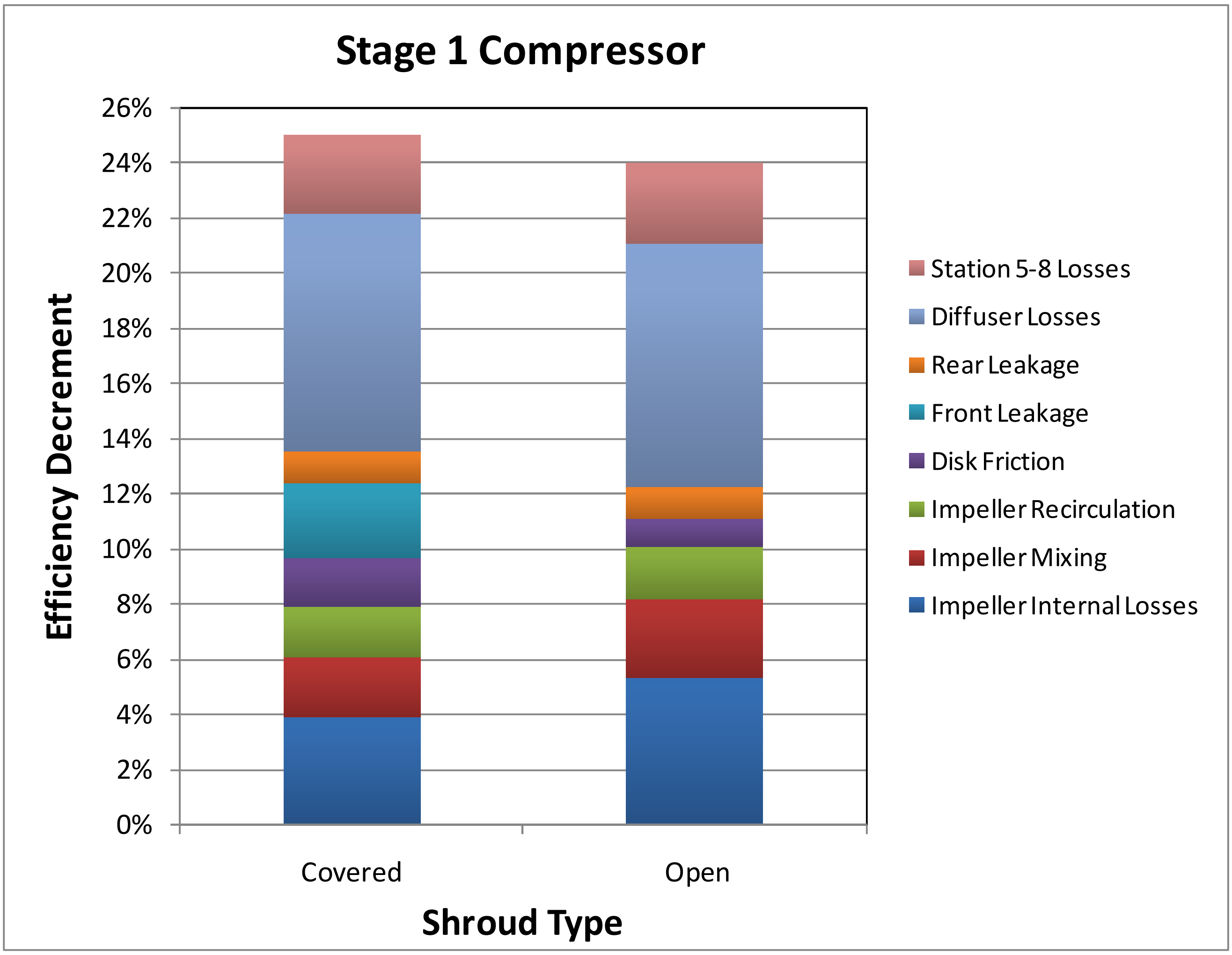


3.7 R744 2-Stage Turbo Sealing Loss Model



For the sealing loss model realistic, achievable values are assumed.

3.8 Open / Closed type wheel comparison



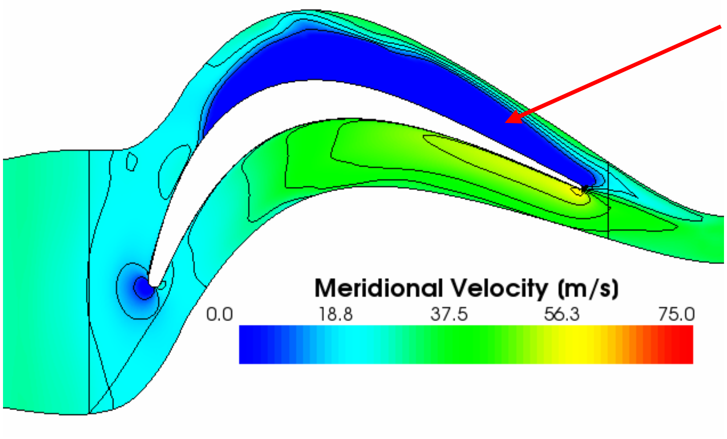
A covered wheel has internal advantages for flow guidance, but additional front side leakage losses. The open type wheel on the other hand has higher passage losses. This results in total in comparable total efficiencies of these two types. But a closed type wheel offers following advantages

- stiff and rigid blade design, reduced critical frequencies
- reduced axial bearing forces (see below)

			Covered Impellers			Open Impellers		
	Parameter	**Units**	***Stage 1***	***Stage 2***	***Overall***	***Stage 1***	***Stage 2***	***Overall***
Performance	Mass Flow Rate	Kg/s	1.667	1.62759	1.667	1.667	1.62753	1.667
	Shaft Speed	rpm	45000	45000	45000	45000	45000	45000
	Stage Power	kW	39.7	33.1	72.8	39.0	32.2	71.3
	Stage PR_{TT}	-	1.56	1.41	2.19	1.55	1.40	2.17
	Stage PR_{TS}	-	1.52	1.40	2.12	1.51	1.39	2.10
	Stage $\square_{TT}$	-	75.68%	76.93%	76.25%	75.98%	77.78%	76.79%
	Stage $\square_{TS}$	-	71.12%	74.79%		71.33%	75.59%	
	Exit Pressure	bar			65.81			65.17
	Axial Thrust	N	-167.9	-704.4	-872.4	-1512.7	-1874.8	-3387.5

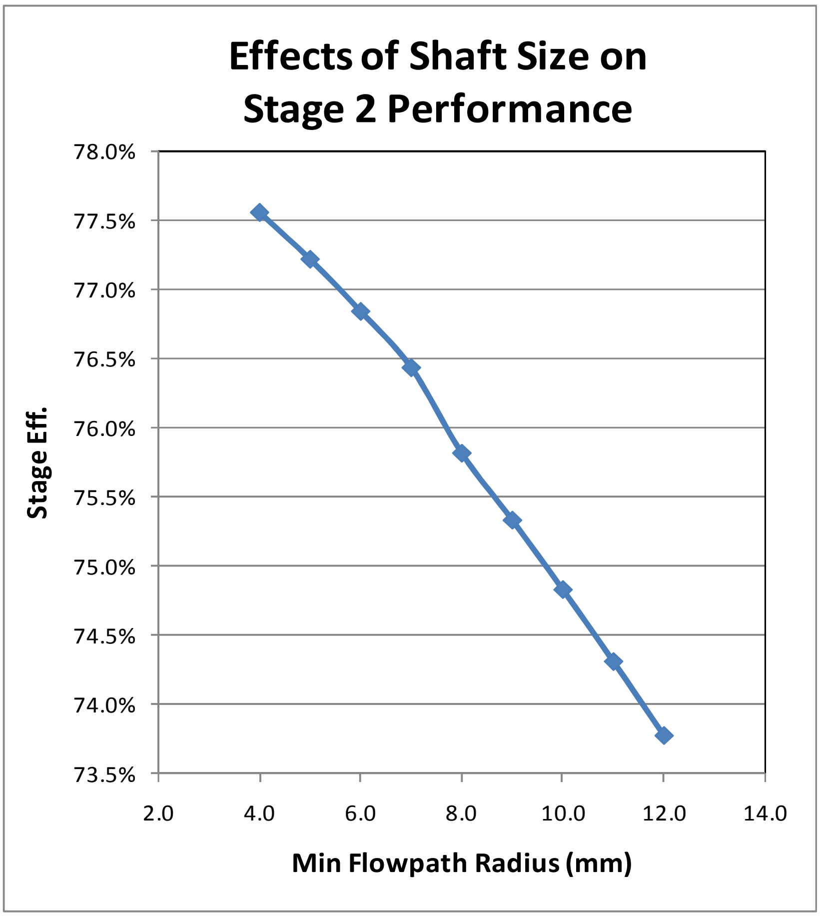
Calculation above shows significant reduction of axial thrust forces with closed impeller type. Hence the axial bearing solution for the R744 turbo unit is judged as critical, this significant reduction and the closed wheel type has to be taken into consideration.

3.09 First CFD results



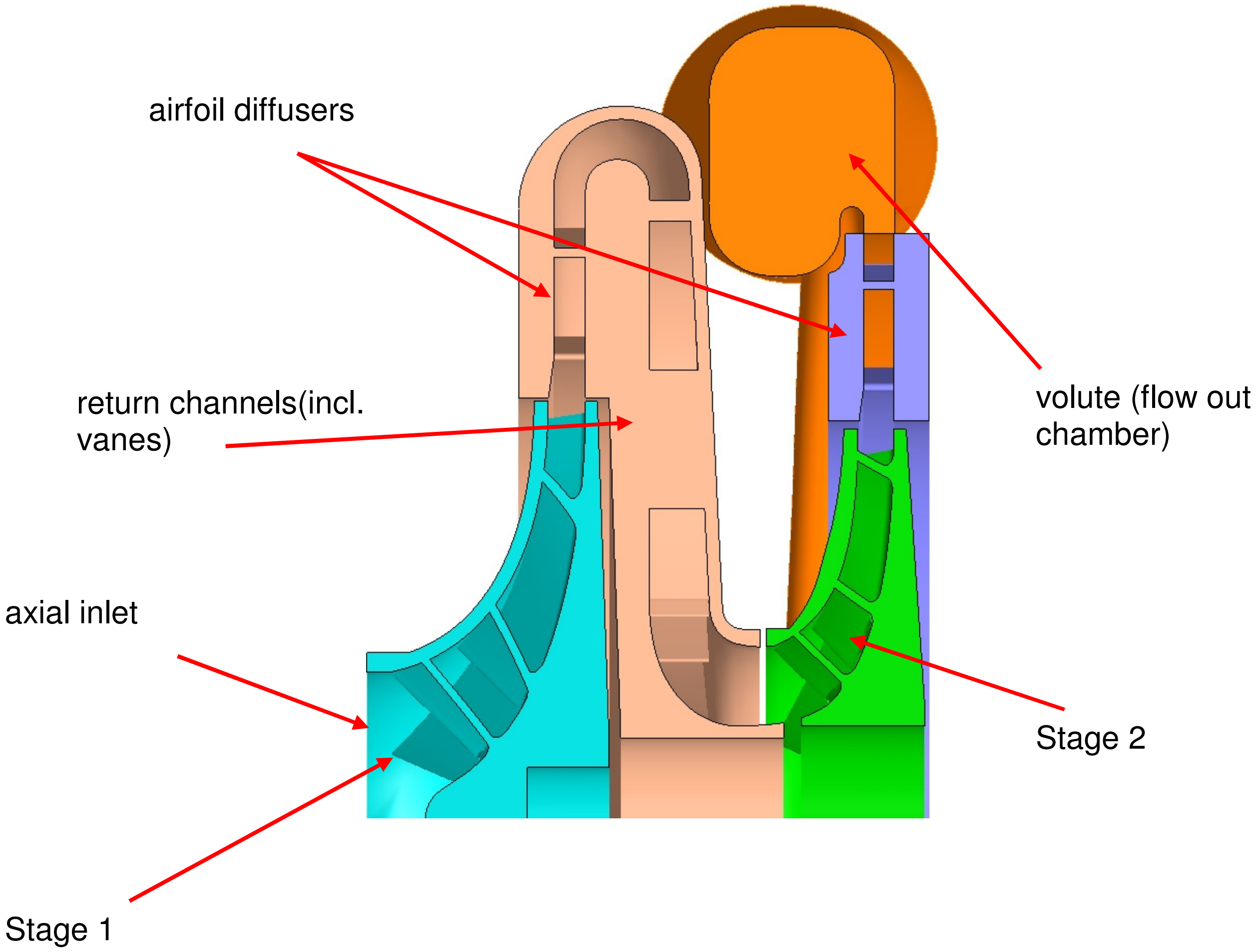
First CFD with ideal gas values shows good results, not perfect yet. The dark blue area in the return channel with zero flow velocity (dead area) has to be reduced.

3.10 Influence of shaft diameter



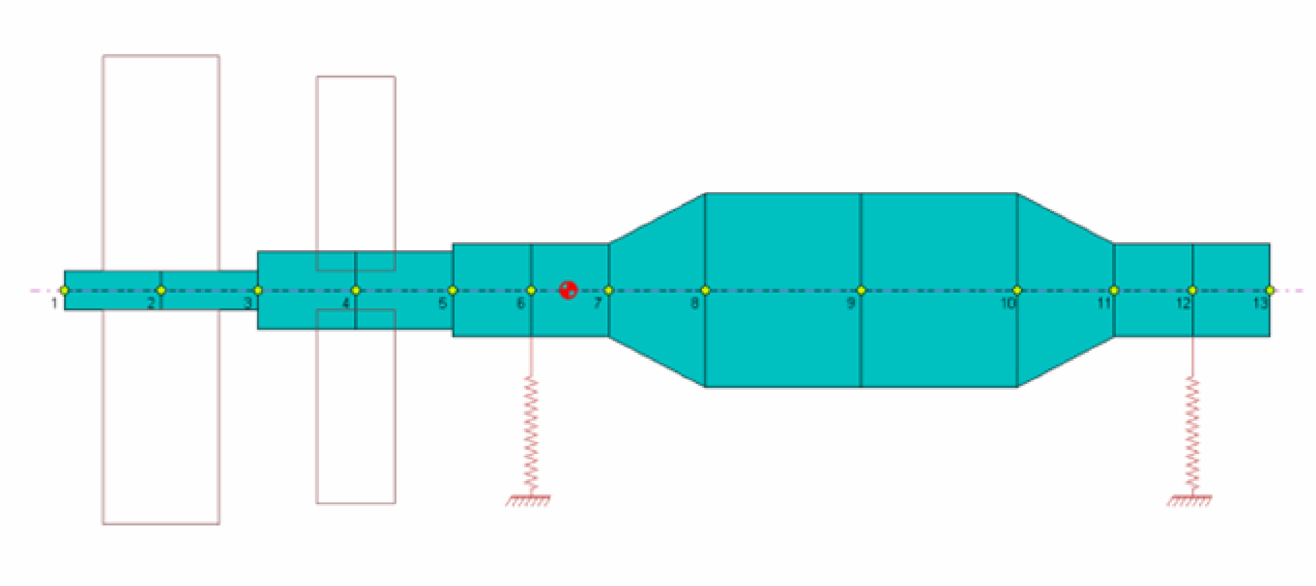
In this study and calculation a shaft radius of 10 mm was chosen, as best compromise of stiffness and efficiency loss. Further, detailed rotor dynamic calculations must confirm and verify this assumption.

4. Turbo Unit Design – 3D



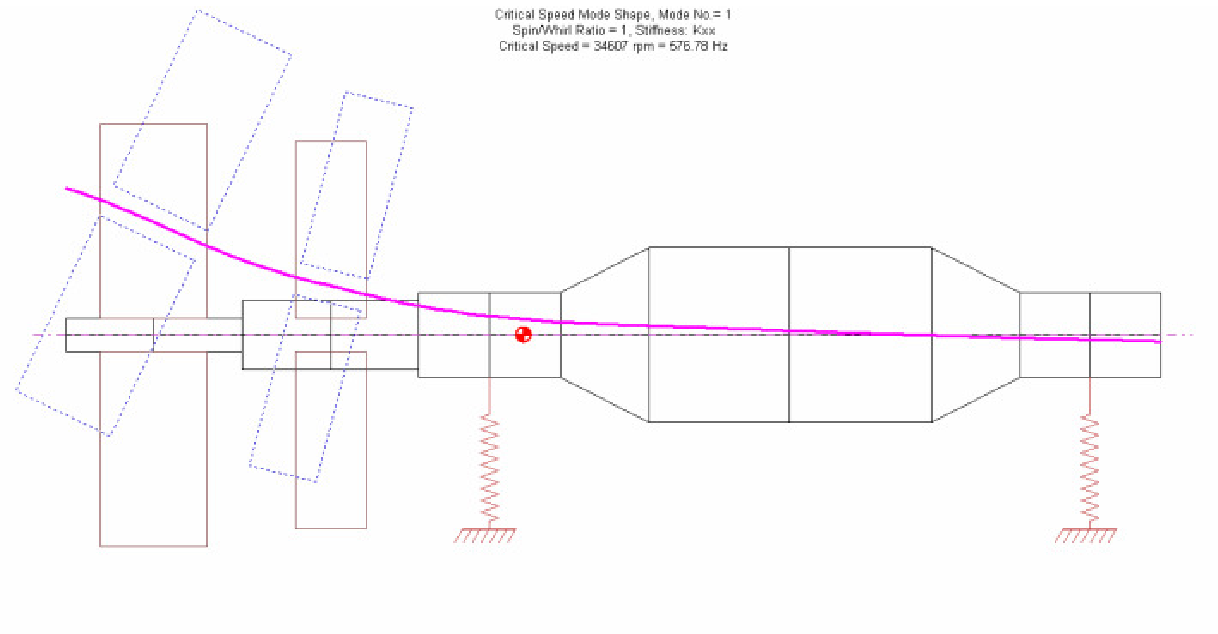
With given simulation results a 3D model of a possible R744 turbo unit, 2-stages was designed. Detailed solutions on sealing and manufacturer feedback required in next steps to further optimise design. After that housing solutions and shaft bearing solutions must follow.

5. Rotor Dynamics – First Calculation



Model assumptions:

- standard ball bearings with a bearing stiffness of 20 kN/mm
- closed impellers
- aluminium alloy for impellers
- standard motor geometry chosen
- both impellers overhung on one shaft end



Results:

- first undamped critical speed predicted to occur at 35.000 rpm
- second undamped critical speed > 100.000 rpm
- with a shaft diameter of 14 mm below the 2nd stage impeller, the localised shaft bending is in acceptable limits
- design speed is above the 1st critical speed → damping required

6. Summary / Conclusions

With the given conditions of the application a R744 oil free turbo compressor unit was calculated and a 3D design of the stages, diffusers, return channels was generated. A two stage compressor is the best compromise for the given condition, the focus for the detailed design must be on leakage reduction. A possible technology is shown. The target of >75% of the compressor unit only (without e-motor or bearing friction losses) are achievable.
The high density of the R744 results in a very compact design. First CFD and rotor dynamic results are presented, but both have to be detailed in next design and development round, when housing, bearing and further compressor parts will be determined. Expected dimensions around 105 mm blade outer diameter and below 130mm total axial length.